REFRIGERATION FUNDAMENTALS

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Refrigerating da

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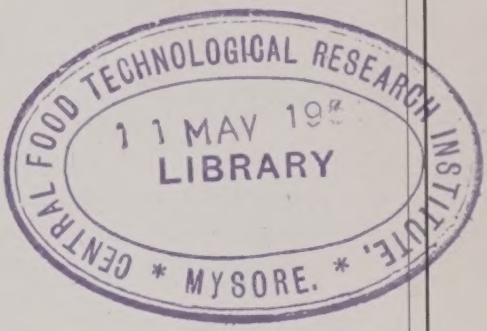
THE REFRIGERATING DATA BOOK

THE REFRIGERATING DATA BOOK

BASIC VOLUME

Sixth Edition



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PREFACE

Former editions of the REFRIGERATING DATA BOOK appeared in 1932, 1934, 1936, 1938, 1940, 1943 and 1946. The editions of 1940 and 1946 were the only DATA BOOKS which did not attempt to include the whole field of refrigeration in a single volume. Data and information fundamental to refrigeration were included in the edition of 1943 but without emphasis on applications, whereas the edition of 1946 was primarily an applications volume.

The present edition, which has been reset and largely rewritten again covers basic data, fundamental principles and practices of refrigeration. This basic-volume REFRIGERATING DATA BOOK is the sixth of a series and the Society proposes in the future to have revisions of it appear biennially in odd numbered years as 1949, 1951, 1953, etc. The applications volume REFRIGERATING DATA BOOK is in its second edition and revisions of this will appear in even numbered years as 1950, 1952, 1954, etc. Under this program the DATA BOOKS can be kept actively abreast of current progress in the field of Refrigeration.

This edition has been prepared under the guidance of six associate editors and some fifty authors, many of whose names appear for the first time in this edition. This volume incorporates the new ASA abbreviations which use less space through the elimination of periods and other unnecessary marks, and in general improve ease of reading.

In a book as comprehensive as this, there is necessarily some overlapping of subject matter, but it is hoped that this has been kept to a minimum. An effort has been made to increase the amount of tabular material in the book. This is particularly true in the case of tables of refrigerant properties, of psychometric data and insulation characteristics.

Suggestions by readers as to items requiring clarification or inconsistencies will be welcomed by the Society. This work represents the unselfish efforts of a large number of cooperating individuals and all of the authors merit the thanks of the Society and of the refrigerating industry for the generous contribution of time and ability which they put into making this book possible.

B. H. JENNINGS, Editor in Chief

Associate Editors:
A. B. STICKNEY
R. J. PANLENER
C. M. ASHLEY
H. B. POWNALL
J. W. CRAIG
C. N. DEVERALL

CONTRIBUTORS

Carl T. Ashby, Unit Engineer, Servel, Incorporated, Evansville, Ind. (*Co-author, Chapter 3*)
Carlyle M. Ashley, Mechanical Engineer, Director of Developments, Carrier Corporation, Syracuse, N. Y. (*Associate Editor, Part III, Author Part 2, Chapter 22*)
Ernst S. H. Baars, Milwaukee School of Engineering, Milwaukee, Wisc. (*Author, Chapter 14*)
J. S. Beamensderfer, Director of Engineering Laboratory, York Corporation, York, Pa. (*Author, Chapter 28*)
George K. Bently, Refrigeration Engineer, McCray Refrigeration Company, Kendallville, Ind. (*Author, Chapter 33*)
W. H. Bickford, Associate Application Engineer, York Corporation, York, Pa. (*Co-author, Chapter 27*)
W. S. Bodinus, District Manager, Carrier Corporation, Chicago, Ill. (*Author Part 2, Chapter 23*)
J. Mason Brown, Chief Engineer, Airco, Limited, Kidderpore, Calcutta, India. (*Co-author, Chapter 12*)
Leon Buehler, Jr., Chief Refrigeration Engineer, The Creamery Package Manufacturing Company, Chicago, Ill. (*Author, Chapter 23*)
R. M. Buffington, Research Engineer, Servel, Incorporated, Evansville, Ind. (*Co-author, Chapter 3*)
Howard F. Burpee, Sales Engineer, Worthington Pump and Machinery Corporation, Harrison, N. J. (*Co-author, Chapter 12*)
Alfred Chadburn, Advertising Manager, The American Society of Refrigerating Engineers. (*Editor, Classified Section*)
John W. Craig, General Works Manager, The Crosley Corporation, Richmond, Ind. (*Associate Editor, Part V*)
Charles N. Deverall, Engineer, Niagara Blower Company, Buffalo, N. Y. (*Associate Editor, Part VI*)
H. W. Eagles, Refrigeration Cabinet Design and Testing Engineer, General Electric Company, Erie, Pa. (*Author, Chapter 30*)
Harry D. Edwards, Works Engineer, Linde Air Products, New York, N. Y. (*Author, Chapter 7*)
C. E. Ehrenhardt, Air Conditioning Department, General Electric Company, Bloomfield, N. J. (*Author, Chapter 19*)
Harold M. Hendrickson, District Engineer, Ralph E. Manns Company, Wilmington, Calif. (*Author, Chapter 37*)
S. R. Hirsch, Executive Engineer, Worthington Pump and Machinery Corporation, Holyoke, Mass. (*Author, Chapter 31*)
B. H. Jennings, Professor and Chairman, Department of Mechanical Engineering, The Technological Institute, Northwestern University, Evanston, Ill. (*Editor-in-Chief, Author, Chapter 41*)
R. L. Jones, Airtemp Division, Chrysler Corporation, Dayton, Ohio. (*Author, Chapter 11*)
Milton Kalischer, Section Refrigeration Engineer, Westinghouse Corporation, Springfield, Mass. (*Author, Chapter 29*)
Joseph F. Kern, Jr., Sales Agent, New York Territory, Pittsburgh Lectrodryer Corporation, New York, N. Y. (*Author, Chapter 34*)
L. C. Leslie, deceased. (*Co-author, Chapter 15*)
C. Osborn Mackey, Professor, Heat Power Engineering, Cornell University, Ithaca, N. Y. (*Author, Chapters 10, 35*)
L. J. Murphy, District Consulting and Application Engineer, Westinghouse Electric Corporation, Buffalo, N. Y. (*Author, Chapter 20*)
Justin Neuhoff, Engineer in Charge of Air Conditioning Production Design, Worthington Pump and Machinery Corporation, Holyoke, Mass. (*Author, Chapter 5*)
Alwin B. Newton, Director of Sales, Acme Industries, Incorporated, Jackson, Mich. (*Author, Chapters 22, 40*)
H. B. Nottage, Research Associate, American Society of Heating and Ventilating Engineers Research Laboratory, Cleveland, Ohio. (*Author, Chapter 21*)
Helen P. Oakley, Tenafly, N. J. (*Editorial Assistant*)
Fred Ophuls, President, Fred Ophuls Associates, Incorporated, New York, N. Y. (*Author, Chapter 2*)
Ralph G. Owens, Professor, Mechanical Engineering Department, Illinois Institute of Technology, Chicago, Ill. (*Author, Chapter 1*)
Everett P. Palmatier, Administrative Engineer, Carrier Corporation, Syracuse, N. Y. (*Co-author, Chapter 4*)
Richard J. Panlener, Director of Industrial Relations, College of Engineering, Marquette University, Milwaukee, Wisc. (*Associate Editor, Part II*)

D. E. Perham, Construction Engineer, Vilter Sales and Construction Company, Chicago, Ill. (*Co-author, Chapter 9*)

H. B. Pownall, Chief Industrial Application Engineer, York Corporation, York, Pa. (*Associate Editor, Part IV*)

Gayle B. Priester, Air Conditioning Engineer, Consolidated Gas Electric Light and Power Company, Baltimore, Md. (*Author, Chapter 32*)

Paul B. Reed, Educational Director Refrigeration Service Engineers Society, Milwaukee, Wisc. (*Author, Chapter 8*)

William R. Rinelli, Director of Customer Relations Department, Ansul Chemical Company, Marinette, Wisc. (*Author, Chapter 13*)

Lucien R. St. Onge, Partner, Florida Hill York Corporation, Miami, Fla. (*Author, Chapter 25*)

S. Charles Segal, Chief Engineer, Kramer-Trenton Company, Trenton, N. J. (*Author, Chapter 39*)

S. P. Soling, Chief Industrial Application Engineer, York Corporation, York, Pa. (*Co-author, Chapter 27*)

Charles L. Spoonhour, Manager, Lewis Refrigeration and Supply Company, Seattle, Wash. (*Author, Chapter 24*)

A. B. Stickney, Engineer, The Rust Engineering Company, Pittsburgh, Pa. (*Associate Editor, Part I, Author, Chapter 16*)

John F. Stone, Staff Manager Insulation Department, Johns-Manville Sales Corporation, New York, N. Y. (*Co-author, Chapter 15*)

Julia I. Szabo, The American Society of Refrigerating Engineers, New York, N. Y. (*Managing Editor*)

Michael F. Tokach, Manager, Application Engineering Division, Vilter Manufacturing Company, Milwaukee, Wisc. (*Co-author, Chapter 9*)

Walter O. Walker, Research Director, Ansul Chemical Company, Marinette, Wisc. (*Author, Chapter 18*)

C. E. Waring, Technical Assistant to the President, The Davison Chemical Corporation, Baltimore 3, Md. (*Author, Chapter 17*)

Daniel D. Wile, Chief Engineer, Refrigerating Engineering, Incorporated, Los Angeles, Calif. (*Co-author, Chapter 4*)

Joseph R. Zwickl, Consulting Engineer, Carbondale Division, Worthington Pump and Machinery Corporation, Harrison, N. J. (*Author, Chapter 6*)

PART I

THEORY

Associate Editor, A. B. Stickney, Engineer,
The Rust Engineering Company, Pittsburgh, Pa.

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1. BASIC THERMODYNAMICS

Definitions of Terms

1. Refrigeration is the science of producing and maintaining temperature below that of the surrounding atmosphere. Low temperatures may be produced in a number of ways, the one most frequently used being that of allowing an expansion of a fluid to take place in such a manner as to extract heat. Consideration of the simplified schematic diagram of a compression refrigerating system shown in **Fig. 1** will lead to an understanding of what thermodynamic changes are involved. The system may be considered as having a low-pressure and a high-pressure side; this separation is shown by the broken line. Low-pressure vapor (refrigerant) from line (1) enters the compressor (A) and is compressed to a high pressure. Passing through line (2), the high-pressure vapor enters the condenser (B), where it is condensed under constant pressure by some medium (usually air or water). The high-pressure liquid in line (3) then expands through the throttling valve (C) to a lower pressure in the evaporator (D). The latent heat necessary to evaporate the refrigerant is taken from brine being circulated through coils in the evaporator, thus cooling the brine. The low-pressure vapor is then ready to be compressed and repeat the cycle, which is indicated on the skeleton Ph chart, **Fig. 2.**

The change 1–2 is an isentropic (s const) expansion and 2–3 a constant pressure contraction, the superheat being removed from the refrigerant (2–2′) and condensation taking place (2′–3) while the latent heat of vaporization is being removed. The change 3–4 represents throttling (h const) of the refrigerant through the expansion valve and 4–1 absorption of heat by the refrigerant in the evaporator.

The absorption-type refrigerating system relies upon the expansion of a high-pressure liquid and its subsequent evaporation at low pressure to produce cold in exactly the same manner as the mechanical compression system. These systems differ, however, in the manner in which the liquid is put under high pressure.

2. Thermodynamics in a broad sense is the science which treats of energy. In a more restricted sense thermodynamics is the science which treats of heat energy and its transformations into other forms of energy. Thermodynamics develops the laws governing these transformations, and studies the properties of the various media by which the transformations are effected. Finally, the principles thus developed are applied to the various heat machines.

3. A body or system or bodies is said to possess **energy** when capable of doing mechanical work while undergoing a change of state.

Kinetic energy is the energy a body possesses by virtue of its motion, the amount of which is determined by

$$\tfrac{1}{2}\, m\ \mathrm{vel}^2$$

where m is the mass and vel the velocity of the body. For a system of bodies, the total kinetic energy is found by adding the separate kinetic energies of all the bodies making up the system.

Potential energy of a system is due to its position or configuration. The work the

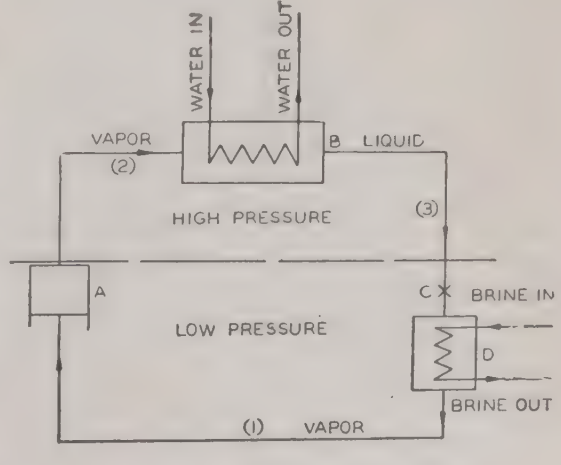

Fig. 1. Scheme of Compression Cycle

system can do in passing from its given position or configuration to a standard reference position or configuration is the measure of its potential energy. The hammer of a pile-driver, when raised, possesses potential energy due to its position, while a stretched rubber band possesses potential energy due to its configuration.

4. Work is accomplished when a force acting on a body displaces it. The amount of work done by the force is determined by the product of the magnitude of the force and the distance through which the body is moved. Thus, if a force of 10 lb displaces a body 5 ft, the amount of work done is $10 \times 5 = 50$ ft lb. The work done by an expanding fluid is given by, work $= \int_{v_1}^{v_2} P dv$. This integral can be evaluated only if the pressure, P, is known as a function of the volume, v. v_1 and v_2 are the initial and final volumes, respectively, of the fluid.

5. Heat, from a strictly thermodynamic standpoint, is **thermal energy** in transition. When heat flows into a body, it is stored as thermal energy in the body. The unit used to measure heat and thermal energy is the British thermal unit. The mean Btu is defined as 1/180 of the amount of heat necessary to raise the temperature of one pound of water from 32 F to 212 F under a pressure of one standard atmosphere.

The **specific heat** of a substance is the amount of heat, expressed in Btu, required to raise the temperature of one pound of the substance one degree Fahrenheit.

The **normal latent heat of vaporization** of a substance is the amount of heat, expressed in Btu, required to change one pound of the liquid to vapor at standard atmospheric pressure. The temperature, being the saturation temperature corresponding to the particular pressure of the substance during vaporization, remains constant during the change. The latent heat of vaporization will vary with pressure. Since the change takes place at constant pressure, the latent heat of vaporization is equal to the change in enthalpy.

The **latent heat of sublimation** of a substance is the heat necessary to change one pound of the solid substance to vapor without passing through the liquid state. The latent heat of sublimation of CO_2 is 248

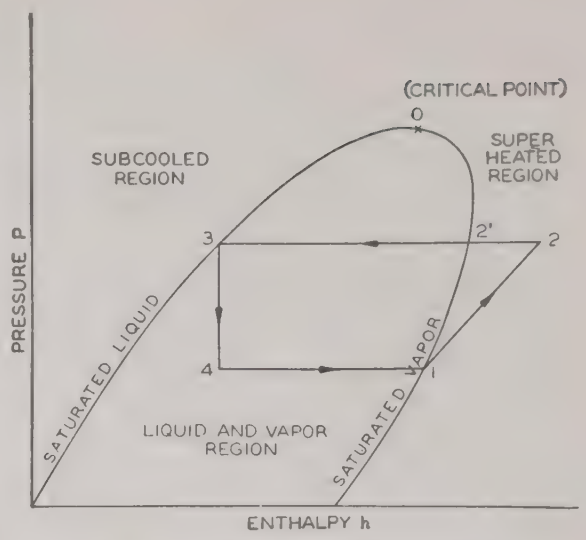

Fig. 2. Pressure-Enthalpy Diagram

Btu per lb. This, together with the approximately 27 Btu which is absorbed by the vapor in going from −109 F, the subliming temperature of CO_2 at atmospheric pressure, to 32 F, makes a total heat-absorbing capacity of 275 Btu per lb.

The **latent heat of fusion** of a substance is the amount of heat necessary to change one pound of the solid to liquid, the temperature and pressure remaining constant during the change.

Thermal energy may be transferred as heat by any one of three methods or combinations thereof, namely, **radiation, conduction,** or **convection.**

A body may dissipate internal kinetic energy by emitting electromagnetic radiation in the form of waves. This process of transferring internal energy from one body to another is called **radiation.** These electromagnetic waves are identical in character with light and radio waves, differing from them only in wave length.

Conduction in solids is probably effected by a transfer of kinetic energy between adjacent vibrating molecules.

Convection is the transfer of thermal energy by mass movements. The movement of warm air through ducts, whether forced by a fan or free due to density variations, is by convection. Cooling a condenser by air is partially a convection process. Heat flows by conduction from the condenser walls through a thin stagnant film of air, and is finally convected away

by the moving air which is in contact with the stagnant film.

6. The **working substance (refrigerant)** is the medium by which the necessary transformations are carried out during the refrigerating cycle. From a thermodynamic standpoint, a number of characteristics are desirable in the working substance. The saturation pressure corresponding to the condenser temperature should be low, while the saturation pressure corresponding to the desired evaporator temperature should be high, since the smaller the pressure ratio the less the compression work. The vapor should have a high specific heat, while that of the liquid should be low. The specific volume-specific enthalpy relationship should be such that the required piston displacement for a given load will be small. The latent heat should be large. The coefficient of performance should be as large as possible and the saturated vapor line should approach as nearly as possible an isentropic line.

Refrigerant Properties

7. **Properties** of the refrigerant which are important in engineering thermodynamics are six in number. Three of these, **pressure, volume** and **temperature,** are directly measurable. The other three, **internal energy, enthalpy** and **entropy,** are functions of the first three.

Pressure, P, is expressed in lb per sq ft or psi.

$$P_{abs} = P_{gage} + 14.7 \text{ psi}$$

Specific volume, v (per lb), the volume occupied by one pound of the substance, is expressed in cu ft per lb.

Temperature, T, is expressed in degrees Fahrenheit absolute.

$$T_{abs} = t_{Fah} + 460$$

From the Carnot theory, the temperature of a body is said to be zero absolute when all of the thermal energy has been removed from the body. This definition of absolute zero does not presuppose any theory of heat. From the standpoint of kinetic theory, absolute zero is reached when all molecular motion ceases, which, of course, presupposes a theory of heat.

Internal energy, u, is expressed in Btu per lb. Kinetic theory attributes the internal energy of a substance to the motion and configuration of the particles (molecules, atoms, and sub-atomic particles) which go to make up that substance. That part of the internal energy due to the motion of the particles is termed sensible thermal energy, while that part due to the configuration of the particles is termed latent energy. Only the sensible thermal energy can be detected by a thermometer.

Enthalpy, h, is defined by the expression $h = u + APv$, where $A = 1/778$. In a flow system Pv is called flow work. When steady continuous flow exists, the product Pv represents the energy necessary to inject the fluid into the pipe entrance. Although in the case of a non-flow system no physical meaning can be attached to this term, the expression for h is still useful. For a non-flow system the change in h is equal to Q under a constant pressure change.

Entropy, s, is expressed in Btu per deg F·lb. For an orderly process (reversible), the change in specific entropy is defined by the expression $ds = dq/T$. The use of this term will be made clear in the discussion of the Carnot cycle (p. 9).

8. The **graphical representation** of properties is frequently used in the solution of refrigerating problems. Any two properties may be used as rectangular coordinates for a chart.

A skeleton Ph chart of the type most frequently used in refrigerating work is shown in Fig. 2. The saturated liquid line is the locus of the state points which represent liquid having a temperature corresponding to the saturation pressure. The saturated vapor line is the locus of the state points which represent vapor having a temperature corresponding to the saturation pressure. The two lines meet at 0, the critical point. The critical temperature is that below which a gas must be cooled before it can be liquefied by pressure alone, regardless of the amount of pressure applied. In the area to the left of the saturated liquid line, called the subcooled region, the temperature of the liquid is less than the saturation temperature corresponding to the pressure of the liquid. In the area between the saturated liquid and saturated vapor lines a mixture of vapor and liquid exists. In the area to the right

of the saturated vapor line, known as the superheated region, the temperature of the vapor is greater than the saturation temperature corresponding to the pressure of the vapor. A state point is determined by any two properties on the chart except in the wet region, where the quality must be given in addition to the two properties.

The Pv, Ts and hs charts are also used in refrigerating work. The Pv and Ts charts have the advantage that areas on these charts have physical meanings. Since the work done by the expanding fluid is given by work $= \int_{v_1}^{v_2} P dv$, the area under the curve on the Pv diagram represents work. From the definition of change in entropy, $dS = dQ/T$, it is seen that areas on the Ts diagram represent heat.

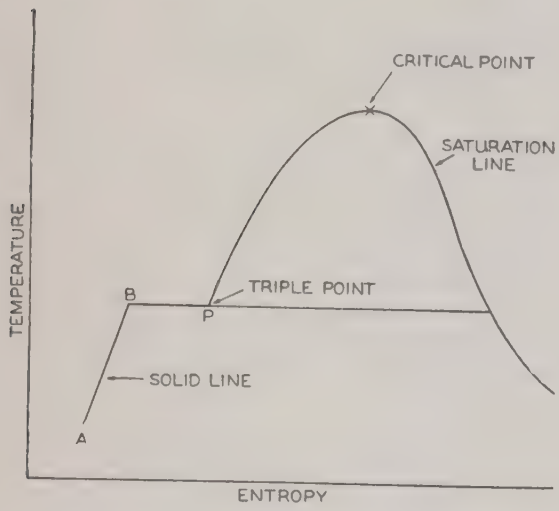

Fig. 3. Temperature-Entropy Diagram

The properties of certain refrigerants, especially carbon dioxide, must be considered in a region which lies outside that already discussed. **Fig. 3** is a Ts diagram for carbon dioxide which includes the triple point. At temperatures above that at P, the triple point, CO_2 can exist as a liquid, a vapor or a mixture of liquid and vapor, as previously discussed. At the triple point, CO_2 can exist as liquid, vapor and solid, and if either of the two properties, pressure or temperature, is varied, one of the phases will disappear. At any temperature below the critical, CO_2 will exist as a solid to the left of line AB, as a mixture of solid and vapor between the solid and saturation lines, and as a vapor to the right of the saturation line.

First Law of Thermodynamics

9. The **first law of thermodynamics** is a restricted statement of the law of conservation of energy. It is restricted in that it is limited in its application to heat and mechanical energy. The first law of thermodynamics states that heat and mechanical energy are mutually convertible and in a definite quantitative relationship, which is expressed in equation form as $E = JQ$. E is mechanical energy, J Joule's mechanical equivalent of heat (778 ft lb per Btu) and Q is heat.

It should be understood that the first law does not give an answer to the question of how easily such transformations can be carried out. While any mechanical energy may be converted one hundred per cent into heat, which eventually must necessarily take place, the conversion from heat to mechanical energy is not so easily accomplished. The first law may be extended and written as an energy balance for both static (non-flow) and flow systems. For a static system the energy balance is

$$q = u_2 - u_1 + A \times \text{work}$$

In other words, the law states that any heat which is added to a system increases the internal energy of the system, leaves the system as work, or is divided between the two. If negative signs are given to any of the quantities, a reverse action is indicated, in that heat may flow from the system, the internal energy may decrease, and work may be done on the system.

For a flow system the energy balance is

$$h_1 + AZ_1 + \frac{A\,\text{vel}_1^2}{2g} \pm q \pm A \cdot \text{work} = h_2 + AZ_2 + \frac{A\,\text{vel}_2^2}{2g}$$

enthalpy	heat equivalent of potential energy	heat equivalent of kinetic energy	heat added or removed	heat equivalent of work added or removed	enthalpy	heat equivalent of potential energy	heat equivalent of kinetic energy

where Z represents the static head above some convenient level. The subscripts 1 and 2 refer to any two cross-sections in the path of the fluid. The plus sign before q

represents heat added and the minus sign represents heat withdrawn from the fluid between the sections 1 and 2, the direction from 1 to 2 being taken as that of the motion of the fluid. The plus and minus signs before work have a similar meaning.

Applying the flow equation to a complete compression system, and assuming no heat leakage, $Q_{evap} - Q_{cond} + A \times$ work $= 0$, or $Q_{evap} + A \times$ work $= Q_{cond}$. The heat absorbed by the refrigerant in the evaporator, plus the heat equivalent of the work done by the compressor, is given out to the cooling fluid in the condenser.

The flow equation may be applied to each unit separately. Assuming the static head through each unit to be negligible, the heat leakage small and no appreciable increase in kinetic energy of the fluid through the compressor, the following equations derived from the flow equation are applicable (subscripts refer to Fig. 2):

$\text{work}_{comp} = h_2 - h_1$	compressor
$q_{cond} = -(h_2 - h_3)$	condenser
$h_3 = h_4$	expansion valve
$q_{evap} = h_4 - h_1$	evaporator

10. The use of **tables and charts** may be shown by the solution of a problem.

Consider an ammonia refrigerating system which operates on the ideal cycle shown in Fig. 2. The suction pressure is 20 psia and the discharge pressure 150 psia. The load is 50 tons. Find (a) the weight rate of flow of ammonia, (b) horsepower of the compressor, (c) heat given up to the brine in the condenser, (d) heat absorbed by the refrigerant in the evaporator, and (e) the coefficient of performance.

The values of h for the various places in the cycle may be taken from the Ph chart for ammonia or from the tables. From the Ph chart h_1 is found at the intersection of the line $P = 20$ psia and the saturated vapor line. This is Chart 2 of Chap. 7.

$$h_1 = 606 \text{ Btu}$$

Following an isentropic line (constant entropy) from this point to the intersection of the line $P = 150$ psia, we find

$$h_2 = 734 \text{ Btu}$$

Proceeding from point (2) along the constant pressure line, $P = 150$ psia, to the intersection with the saturated vapor line

$$h_2' = 630 \text{ Btu}$$

The constant pressure line, $P = 150$ psia, is followed further to the left until it intersects the saturated liquid line and we find

$$h_3 = 130 \text{ Btu}$$

Point 4 in the cycle is located by proceeding from (3) along the line of constant enthalpy (vertical on the Ph chart), until intersection with the horizontal line, $P = 20$ psia. Since this change is one of constant enthalpy

$$h_4 = 130 \text{ Btu}$$

(a) weight rate $= \dfrac{200 \times \text{tons}}{(h_1 - h_4)}$ lb per min

$= \dfrac{200 \times 50}{(606 - 130)}$

$= 21$ lb per min

(b) hp $= \dfrac{21\ (h_2 - h_1)}{42.42}$

$= \dfrac{21\ (734 - 606)}{42.42}$

$= 63.4$

(c) q_{cond} = weight rate $(h_2 - h_3)$ Btu per min

$= 21\ (734 - 130)$

$= 12{,}700$ Btu per min

(d) q_{evap} = weight rate $(h_1 - h_4)$ Btu per min

$= 21\ (606 - 130)$

$= 10{,}000$ Btu per min

(e) cop $= \dfrac{\text{heat absorbed in evaporator}}{\text{energy supplied to compressor}}$

$= \dfrac{h_1 - h_4}{h_2 - h_1}$

$= \dfrac{606 - 130}{734 - 606}$

$= 3.72$

11. Thermodynamic changes from one state point to another may follow any number of paths, depending upon the manner in which the change is carried out.

An **adiabatic** change is one in which no heat flows into or out of the system. If the medium is such that it obeys the perfect gas laws in the region in question, the functional relationship between pressure and volume is $Pv^k = \text{const}$, where $k = c_p/c_v$. c_p and c_v are the specific heats at constant pressure and constant volume, respectively.

An **isothermal** change is one in which the temperature of the working substance remains constant.

An **isentropic** change requires that the entropy of the substance remain constant in addition to being adiabatic.

A **polytropic** change for a gas follows the path represented by $Pv^n = \text{const}$, where $(1 < n < k)$.

12. Work of the compressor is found by use of an expression which is easily developed. **Fig. 4** shows an ideal indicator diagram A-B-C-D for a compressor without

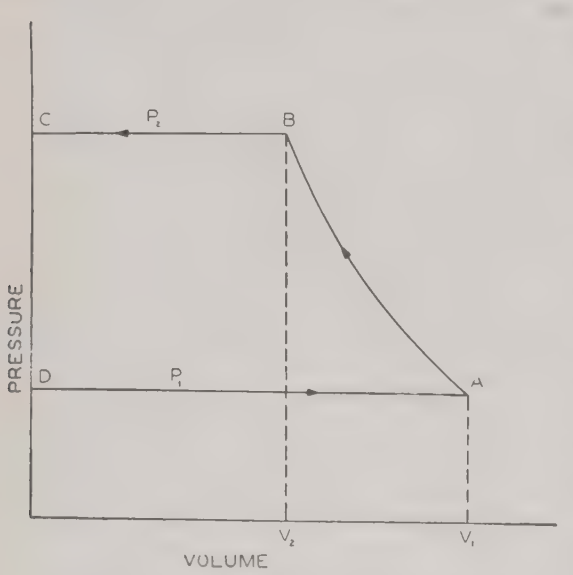

Fig. 4. Ideal Indicator Diagram without Clearance

clearance. The change A-B is a polytropic compression following the law $Pv^n = c$ from a pressure P_1 and volume V_1 to a pressure P_2 and volume V_2. B-C represents expulsion of the refrigerant vapor at constant pressure P_2 into the high-pressure side of the refrigerating system. D-A represents the suction stroke when the low-pressure refrigerant enters the compressor. Remembering that the area under a curve on the Pv diagram represents work, the area A-B-C-D-A may be seen to represent the work done by the compressor during a cycle.

work during suction stroke	$\text{D-A} = P_1V_1$
work during compression	$\text{A-B} = \dfrac{(P_1V_1 - P_2V_2)}{(n-1)}$
work during expulsion	$\text{B-C} = -P_2V_2$

The net work for the cycle W_c is found by adding algebraically the separate expressions

$$W_c = \frac{P_1V_1 - P_2V_2}{n-1} + P_1V_1 - P_2V_2$$

$$W_c = \frac{n}{n-1}(P_1V_1 - P_2V_2)$$

This equation may be transformed into the form

$$W_c = \frac{n}{n-1} P_1V_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{(n-1)/n}\right]$$

which shows that the work of compression depends upon the pressure ratio.

The **effect of clearance** may be considered by referring to **Fig. 5.** The changes 3-4 and 5-2 are polytropic. For mechanical reasons it is necessary to have some clearance, which means that some gas volume will remain in the cylinder at the end of the discharge stroke. This high-pressure gas expands behind the cylinder, and the intake valve does not open until the gas pressure drops to that of the intake. The volume V_C is known as the "clearance volume," and the ratio of this volume to the displacement volume, V_C/V_D, is known as the "clearance."

The area 2-3-4-5, which represents the work of the cycle, is the difference between the two areas 1-3-4-6 and 1-2-5-6, each of which represents a no-clearance type cycle. The expression for this type of cycle has been developed.

Subtracting the expressions for the work for these two cycles, work for cycle

$$\text{2-3-4-5} = \frac{n}{n-1} P_1V_1 \left[1 - \left(\frac{P_2}{P_1}\right)^{(n-1)/n}\right]$$

Thus clearance has no effect on the work, provided the actual volume of gas taken in is used in place of the displacement volume.

13. An expression for the **theoretical volumetric efficiency,** which is the ratio of the theoretical intake volume to the displacement volume, may be easily developed.

$$\eta = \frac{V_I}{V_D} = \frac{V_3 - V_2}{V_3 - V_4}$$

$$V_2 = V_3 \left(\frac{P_2}{P_1}\right)^{1/n} \text{ polytropic change}$$

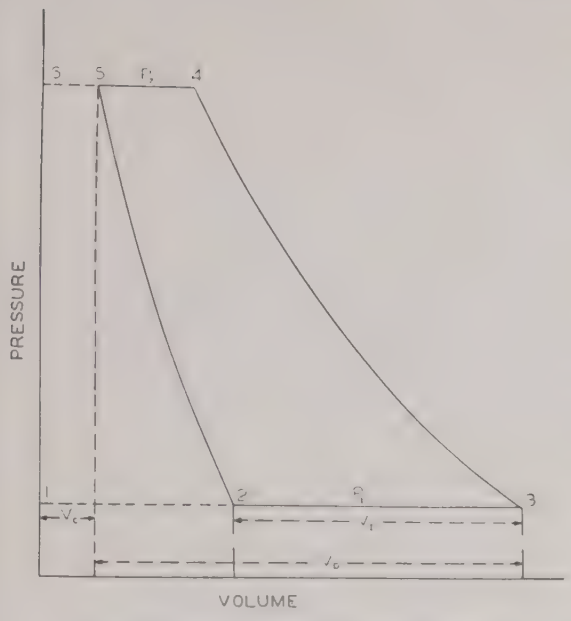

Fig. 5. Ideal Indicator Diagram with Clearance

$$V_3 - V_2 = V_3 - V_5 \left(\frac{P_2}{P_1}\right)^{1/n}$$

we have $$\eta = \frac{V_3}{V_3 - V_5} - \frac{V_5}{V_3 - V_5}\left(\frac{P_2}{P_1}\right)^{1/n}$$

the clearance $$C = \frac{V_5}{V_3 - V_5} = \frac{V_C}{V_D}$$

$$1 + C = \frac{V_3}{V_3 - V_5}$$

thus $$\eta = 1 + C - C\left(\frac{P_2}{P_1}\right)^{1/n}$$

14. The **second law of thermodynamics,** which limits the conversion of heat energy into mechanical energy, may be stated in several different ways. Two statements of the law are: heat of itself cannot flow up a temperature gradient, and no self-acting machine can completely and continuously transform all of the heat supplied to it into mechanical energy.

15. The **Carnot cycle** is a theoretical, highly idealized cycle which permits the determination of the maximum possible amount of mechanical energy which may be derived from a given amount of heat energy. The Carnot cycle on a Ts diagram is shown in **Fig. 6.** The cycle is made up of two isentropic and two isothermal changes. All changes are considered to be reversible. The area under B-C to the s axis represents heat supplied to a Carnot engine at a high temperature, T_1, and the area under A-D to the s axis represents heat given out by the Carnot engine at the low temperature, T_0. The area A-B-C-D represents the heat equivalent of the work done during the cycle by the Carnot engine.

$$\text{Efficiency} = \frac{\text{output}}{\text{input}} = \frac{Q_1 - Q_0}{Q_1} = \frac{T_1 - T_0}{T_1}$$

Note that the Carnot efficiency increases as T_0 decreases. The practical limit of T_0 is the temperature of the atmosphere. A lower temperature could be maintained, but would require the expenditure of energy as great as that which would be gained by the engine.

A reversible engine working between given temperature limits is the most efficient one which can be operated between the same temperature limits.

Available energy is defined as the maximum conceivable amount of a given amount of heat energy which may be transformed into mechanical energy.

$$Q_{\text{avail}} = Q_{\text{sup}}\left(1 - \frac{T_0}{T_1}\right)$$

The available energy is always less than the energy supplied because it is impossible to expand to absolute zero.

Unavailable energy is the minimum amount of energy which must be rejected from a Carnot engine at the lowest available temperature.

$$Q_{\text{unavail}} = Q_{\text{sup}}\frac{T_0}{T_1} = T_0 \Delta S$$

This equation states that if Q units of heat are added to a body at a temperature T_1, the unavailable energy of the heat added is given by the product T_0, the lowest available temperature, and the increase in entropy (ΔS) of the body due to the heat added.

If the entropy within a system increases due to friction, the unavailable energy of the system will increase. The increase in unavailable energy accompanying any self-proceeding change in nature is equal to the product of the increase in entropy and the lowest available temperature. Entropy is, therefore, a measure of availability.

Example. 100,000 Btu are supplied to a large body at constant temperature, 212 F.

If the temperature of the atmosphere is 40 F, which may be taken as the lowest available temperature, how many Btu are available for doing mechanical work?

Solution: The increase in entropy of the body due to the heat added is

$$\Delta S = \frac{Q}{T} = \frac{100{,}000}{(460+212)}$$

$$Q_{\text{unavail}} = T_0 \Delta S = 500 \times \frac{100{,}000}{672} = 74{,}405 \text{ Btu}$$

Since the Carnot cycle is a reversible one, heat can be absorbed from the cold

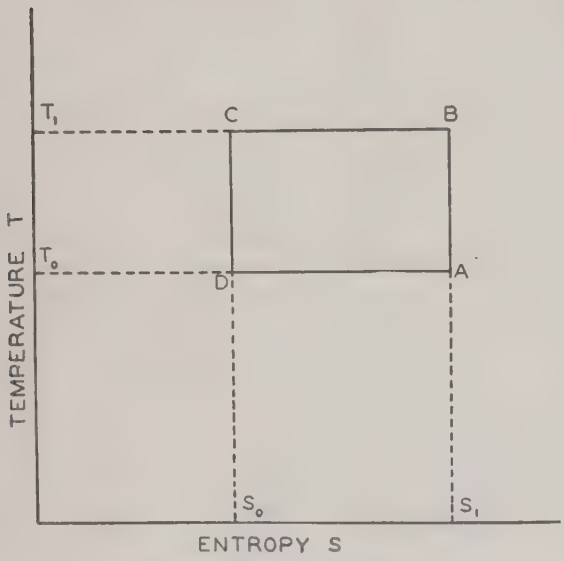

Fig. 6. Carnot Cycle

body, energy supplied to the Carnot engine from without the system and heat expelled at the high temperature, T_1. The reversed Carnot cycle gives the optimum of performance for any refrigerating machine. It is clear that it would be impossible to attain it in practice.

16. The **coefficient of performance** (cop) of a refrigerating system is defined as

$$\frac{\text{heat absorbed at low temperature}}{\text{heat equivalant of work done}}.$$

For the Carnot cycle,

$$\text{cop} = \frac{Q_0}{Q_1 - Q_0} = \frac{T_0}{T_1 - T_0}$$

Example. If 6 hp are supplied to a refrigerating machine operating on the Carnot cycle with a coefficient of performance of 4, (a) how many Btu per min are removed from the cold body; and (b) how many Btu per min are rejected at the high temperature?

Solution: (a) 6 hp is equivalent to $6 \times 33{,}000 = 198{,}000$ ft lb per min

$$Q_1 - Q_0 = \frac{198{,}000}{778} \text{ Btu}$$

$$\text{cop} = \frac{Q_0}{Q_1 - Q_0} = 4$$

$$Q_0 = 4(Q_1 - Q_0) = 4 \times \frac{198{,}000}{778} = 1018 \text{ Btu per min}$$

(b) $$Q_1 = 5/4 Q_0 = \frac{5 \times 1018}{4} = 1272 \text{ Btu per min}$$

17. One **ton** of refrigeration is defined as 200 Btu absorbed per min or 12,000 Btu per hr. A **ton-day** is equal to 288,000 Btu, which is equivalent to the latent heat of a ton of ice.

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2. COMPRESSION CYCLE OF RECIPROCATING COMPRESSORS

1. The compression cycle starts where the suction vapor enters the suction stop valve and ends where the compressed vapor leaves the discharge stop valve on the way to the condenser. An analysis of the compression cycles of the two most commonly used refrigerants, anhydrous ammonia and Freon-12, is given here.

Since the reciprocating compressor is the most important element of this cycle, its behavior under varying operating conditions will be studied first.

In Chapter 23 the design and handling of industrial compressors are described. It is the purpose here to point out how these two factors affect the operating results of these machines under varying pressure and temperature conditions.

Standard Rating Cycle

2. Some years ago The American Society of Refrigerating Engineers proposed and adopted certain **standard operating conditions** for rating refrigerant compressors. These standard rating conditions are:

Saturated evaporation temperature, 5 F
Suction gas to compressor, 9° superheat, 14 F
Saturated condensing temperature, 86 F
Liquid at expansion valve, 9° supercooled, 77 F

The corresponding ammonia pressures are 169.2 and 34.27 psia, or a compression ratio of 4.937. For Freon-12 these pressures are 107.9 and 26.51, or a compression ratio of 4.066.

On **Fig. 1** are shown the $T-s$, $P-h$, and indicator card ($P-v$) diagrams of the standard rating cycle, with reference points given in **Table 1** and operating data in **Table 2.** They are based entirely on ideal or theoretical conditions, from which actual machines depart. **Fig. 2** shows the

Table 1. Reference Points for Standard Rating Cycle Analysis

Refrigerant	Point (Fig. 1)	p psia	v cu ft/lb	t F	s Btu/lb F	h Btu/lb
Ammonia	1	169.2	0.02691	86	0.2875	138.9
Freon-12		107.9	0.0124	86	0.05708	27.72
Ammonia	2	169.2	0.02657	77	0.2685	128.4
Freon-12		107.9	0.0122	77	0.05315	25.56
Ammonia	3	34.27	1.176	5	0.2820	128.4
Freon-12		26.51	0.3557	5	0.05589	25.56
Ammonia	4	34.27	8.15	5	1.3253	613.3
Freon-12		26.51	1.485	5	0.17052	78.79
Ammonia	5	34.27	8.241	14	1.3369	618.7
Freon-12		26.51	1.516	14	0.17317	80.04
Ammonia	6	169.2	2.387	222.8	1.3369	720.6
Freon-12		107.9	0.416	109.5	0.17317	91.13
Ammonia	7	169.2	1.772	86	1.1904	631.5
Freon-12		107.9	0.389	86	0.16640	87.37

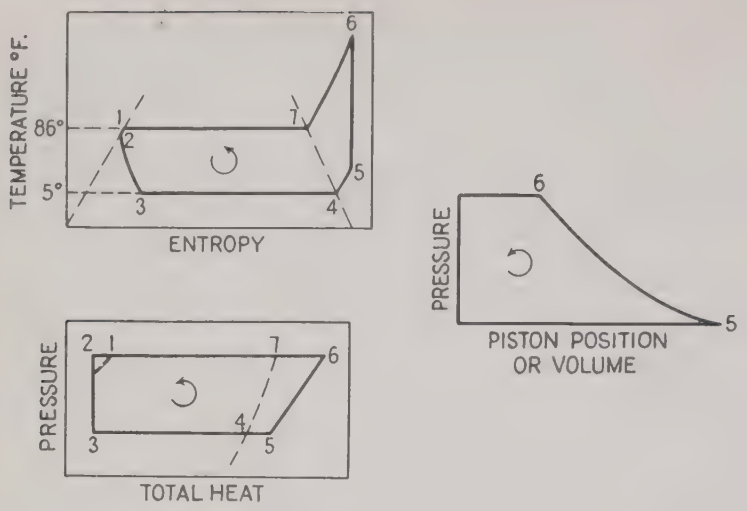


Fig. 1. Rating Cycle (see Table 1)

cfm of ammonia or Freon-12 per ton for various other operating conditions. The three discharge pressures plotted are nearly the same in temperature for the two refrigerants, so that the curves give a close indication of comparative compressor sizes. **Fig. 3** gives adiabatic discharge temperatures which are considerably lower for Freon-12 than for ammonia.

On **Fig. 4** the theoretical adiabatic power requirements for anhydrous ammonia compression at various suction pressures are plotted for three different condenser pressures. The solid curves give the values for single-stage compression and the broken curves for compound compression. The theoretical equivalent values for Freon-12 are about 98% of those for ammonia.

Theoretical Compression Cycle

3. For a clear understanding of the theoretical cycle three diagrams are useful: first, the indicator diagram, **Fig. 5**; second, the temperature-entropy diagram, **Fig. 6**; and third, a plot of temperature against crank position, which is called the time-temperature diagram, **Fig. 7**. The cycle diagram, **Fig. 8**, shows the essential elements of the single-stage compression refrigeration system with lettering to correspond to the points in Fig. 6. Figs. 5 and 7 refer to the compressor alone and Figs. 6 and 8 to the entire system. Compression cycle points are denoted in capital letters and system points in small letters.

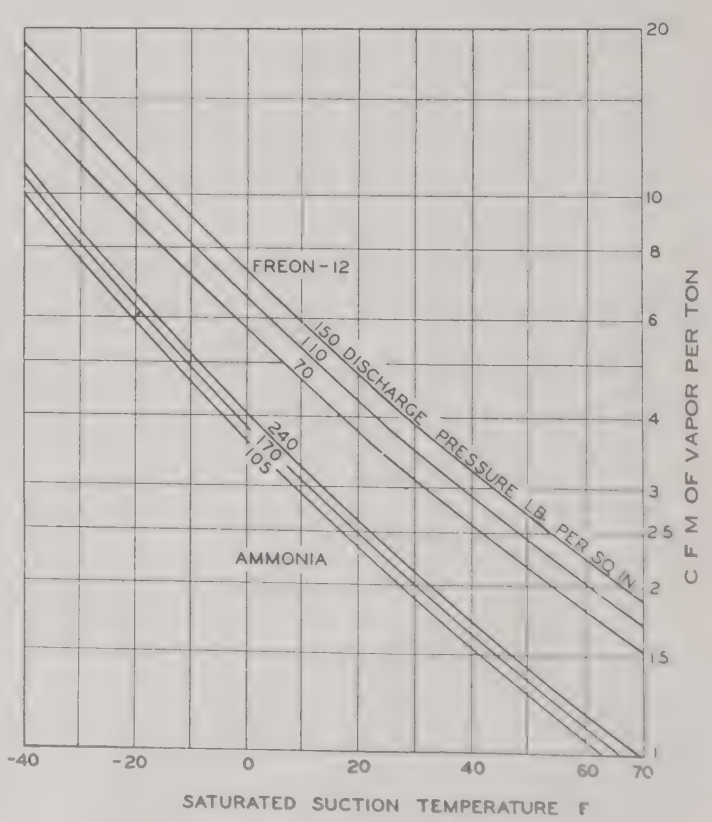


Fig. 2. Cfm Freon-12 or Ammonia per Ton

(Saturated suction and liquid expansion at saturation temperature corresponding to discharge pressure.)

Table 2. Standard Rating Cycle Data

	Ammonia	Freon-12
Refrigerating effect, Btu per lb	490.3	54.48
Cfm per ton	3.36	5.56
Adiabatic hp per ton	0.979	0.959

In Figs. 5 and 7, AB represents the compression, BC the discharge, CD the re-expansion, DA the suction. In Figs. 6 and 8, *a* is the suction from D to A (Figs. 5 and 7), *b* is the discharge from B to C, *c* is the liquid leaving the condenser, e the mixture of liquid and vapor leaving the expansion valve. After passing through the evaporator the cycle is completed at the suction *a*. The line CD is not shown in Figs. 6 and 8. It represents the re-expansion of the vapor trapped in the clearance space of the compressor. Area *a-b-c-d-a* represents the work of adiabatic compression from *a* to *b*, and area *e-f-g-a-e* the refrigerating effect, if distance *e-f*, which is foreshortened in the figure, represents the absolute temperature of the evaporating liquid. In Fig. 7 level DA represents the temperature of the incoming suction gas, and level BC the theoretical discharge temperature, which is also represented by point b of Fig. 6 and developed further in Fig. 3.

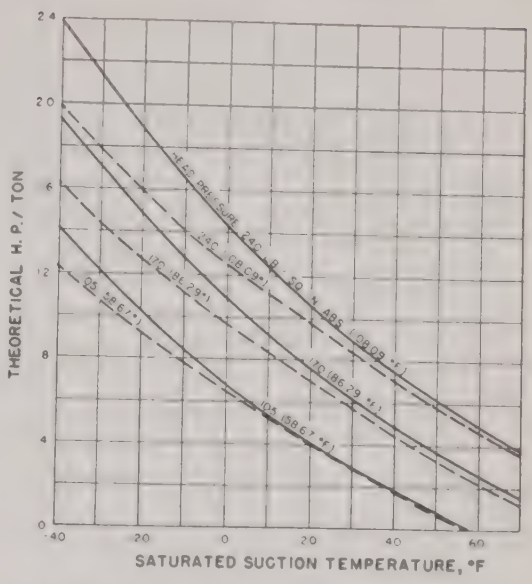


Fig. 4. Theoretical hp/ton for Ammonia Compressor

Values for Freon-12 approximately 98% for same suction and condensing temperatures; broken line curves apply to compound compression with $P_{int} = 1.1\sqrt{P_{suct} \times P_{disch}}$ 1st stage; discharge water cooled to 108 F and cooled by ammonia to saturation temperature. Liquid cooled by ammonia to saturated intermediate temperature.

Figs. 5, 6, and 7 are constructed for **adiabatic compression,** since it is customary to use adiabatic compression as the standard of comparison on refrigeration duty. Air compressors are frequently com-

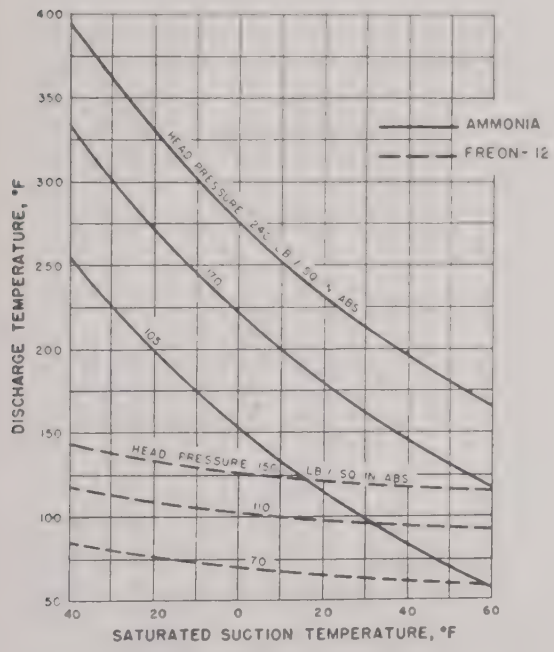


Fig. 3. Adiabatic Discharge Temperatures

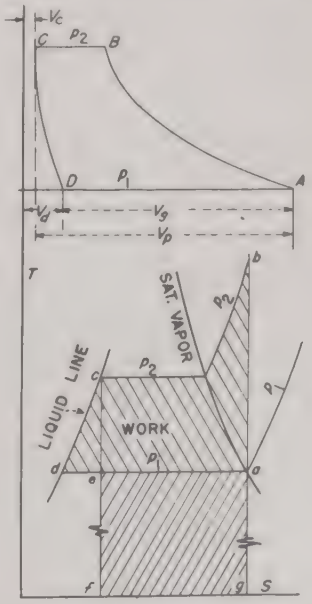


Fig. 5 (above). Indicator Diagram
Fig. 6. Temperature-Entropy Diagram

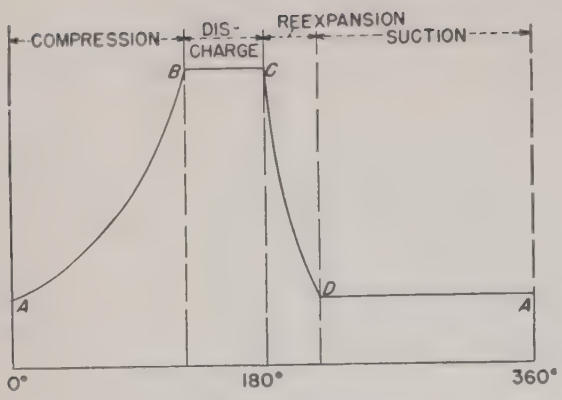


Fig. 7. Theoretical Time-Temperature Diagram

pared with the isothermal, and on indicator cards from refrigerating machines the isothermal compression curve (compression exponent of 1) is frequently drawn in for the purpose of studying valve action in the compressor. Actually, in refrigeration, the isothermal curve has no physical meaning (unless there is extremely high suction superheat), since isothermal compression would result in the discharge of subcooled liquid refrigerant instead of vapor and, due to the change of state, the volume would be very much reduced below the curve with the exponent 1. Furthermore, if a cooling fluid were available at a temperature low enough to equal the suction temperature (again excepting the case of high suction superheat), there would be no need for the refrigerating plant, as the cooling fluid itself could be used directly.

Clearance Volume

4. A reciprocating compressor cannot be designed so that all the vapor contained in it is forced out by the piston at the end of the discharge stroke. There must be some clearance between the piston and cylinder heads at that point of the stroke. The design and location of the compressor valves often leave pockets that the piston cannot fill completely. This total space is called the clearance volume.

The vapor confined in the clearance volume has been compressed to the condenser pressure, and when the suction stroke begins this vapor will re-expand and will not allow the suction valves to open and admit additional vapor until its pressure has been reduced to the suction pressure (Fig. 5). The following is a discussion of the effect the clearance volume has on the theoretical vapor pumping capacity of the compressor.

Nomenclature and Equations (see Fig. 5)

V_p = piston displacement
V_c = clearance volume
V_d = re-expansion volume
$V_g = V_p + V_c - V_d$

$$m = \text{percent clearance} = \frac{100 V_c}{V_p}$$

n = compression exponent
$V_d = V_c (P_2/P_1)^{1/n}$
$V_g = V_p - V_c[(P_2/P_1)^{1/n} - 1]$
$V_c = mV_p/100$
R = compression ratio
E_1 = volumetric efficiency of theoretical cycle, which is

$$E_1 = \frac{V_g}{V_p} = \frac{V_p - \frac{mV_p}{100}\left[\left(\frac{P_2}{P_1}\right)^{1/n} - 1\right]}{V_p}$$

$$= 1 - \frac{m}{100}\left[\left(\frac{P_2}{P_1}\right)^{1/n} - 1\right]$$

$$= 1 - \frac{m}{100}\,(R^{1/n} - 1).$$

Tables 3 and **4** respectively list the values of $(R^{1/n} - 1)/100$ for ammonia and for Freon-12. Then the **theoretical volumetric efficiency** = 1 − (percentage of clearance × tabular value).

Applying the clearance factors to the standard rating cycle, the theoretical volumetric efficiency of the compressors operating under these conditions for varying clearance volumes for ammonia is

$$1 - \frac{m}{100}\left[\left(\frac{169.2}{34.27}\right)^{1/1.3} - 1\right]$$

$$= 1 - 0.02415m.$$

Theoretical volumetric efficiency of Freon-12 compressor is

$$1 - \frac{m}{100}\left[\left(\frac{107.9}{26.51}\right)^{1/1.138} - 1\right]$$

$$= 1 - 0.02433m$$

These equations show conclusively that for the same percentage of clearance its effect on the theoretical displacement is

practically the same for both ammonia and Freon-12. As a matter of fact, the theoretical volumetric efficiency of compressors having the same clearance volume is less for the Freon-12 than for the ammonia compressor, when operated at their respective compression ratios corresponding to the same heat levels for suction and discharge.

For the analysis of the actual compression cycle, the effect clearance has on the weight of refrigerant handled by the compressor in the theoretical cycle must be known.

Referring again to the indicator diagram shown in Fig. 5 it appears that, if:

w_c = weight of the recirculated gas, in lb per min
w_s = weight of suction gas, in lb per min
V_s = specific volume of the suction gas in cu ft per lb,

then

$$w_c = \frac{V_d}{V_s}; \qquad V_d = V_c R^{1/n} = \frac{m}{100} V_p R^{1/n}$$

$$w_c = \frac{\frac{m}{100} V_p R^{1/n}}{V_s}; \qquad V_p = \frac{V_g}{E_1} = \frac{w_s V_s}{E_1}$$

$$w_c = \frac{\frac{m}{100} R^{1/n} w_s}{E_1}$$

This shows that both the weight of the refrigerant flowing through the compressor and that which recirculates continuously can be calculated for the theoretical cycle. Tables 3 and 4 can be used to obtain $R^{1/n}$ by adding 0.01 to the values given in them for the R under consideration.

The effect of clearance on the theoretical cycle when the ratio of compression is changed is illustrated in **Fig. 9.** In this figure, the theoretical cards of two machines having different clearances are shown. Card AB_1C_1DA is the record of the machine having the larger clearance, V_{C1}; and AB_2C_2DA is the card of the machine having the smaller clearance, V_{C2}, when operating between pressures p_1 and p_2. These machines both handle the same volume of suction gas, V_G, but the larger-clearance machine must have a larger displacement, V_{p1}, than the smaller-clearance machine whose displacement is V_{p2}. The

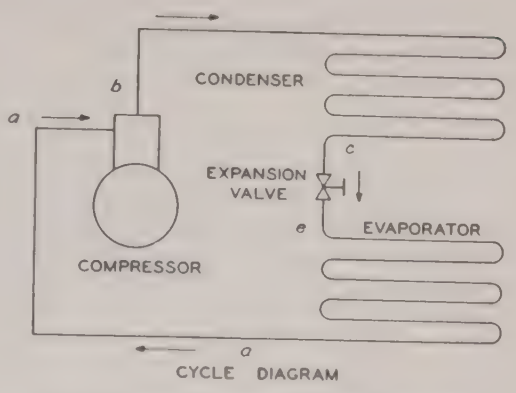


Fig. 8. System Related to Cycle Diagram

theoretical power consumption of both compressors is the same when operating under the same suction and discharge pressures, p_1 and p_2.

If, now, the suction pressure is raised to p_1', without changing the discharge pressure, the card for machine 1 becomes $A'B_1'C_1D_1'A'$, and of machine 2 becomes $A'B_2'C_2D_2'A'$. In this case, the two machines which formerly had the same capacity, V_G, now have different capacities, V_{G1} and V_{G2}, the larger-clearance machine having increased in capacity more than the smaller-clearance machine. Conversely, if the ratio of compression had been increased, the capacity of machine 1 would have decreased more than that of machine 2. On the other hand, of two compressors having the same piston displacement the one having the smaller volumetric clearance will have the larger capacity irrespective of the compression ratio under which they are compared.

These conclusions are true for the theoretical compression cycles of all refriger-

Table 3. Clearance Factors for Ammonia, $(R^{1/n}-1)/100$ when $n=1.3$

R	.0	.2	.4	.6	.8
1	.0000	.0015	.0030	.0044	.0057
2	.0070	.0083	.0096	.0109	.0121
3	.0133	.0145	.0156	.0168	.0179
4	.0190	.0201	.0213	.0223	.0234
5	.0245	.0255	.0266	.0276	.0287
6	.0296	.0307	.0317	.0327	.0337
7	.0347	.0357	.0366	.0376	.0386
8	.0395	.0405	.0414	.0423	.0423
9	.0442	.0451	.0460	.0470	.0478
10	.0488	—	—	—	—

Table 4. Clearance Factors for Freon-12, $(R^{1/n}-1)/100$ when $n=1.138$

R	.0	.2	.4	.6	.8
1	.0000	.0018	.0034	.0051	.0068
2	.0084	.0100	.0116	.0132	.0147
3	.0163	.0178	.0193	.0208	.0223
4	.0238	.0253	0268	.0282	.0297
5	.0312	.0326	.0340	.0355	.0371
6	.0383	.0398	.0411	.0425	.0440
7	.0456	.0468	.0480	.0495	.0510
8	.0523	.0535	.0549	.0562	.0576
9	.0589	.0603	.0617	.0633	.0645
10	.0657	—	—	—	—

ants. The analysis of the actual compression cycle appearing in this chapter, under the heading "Real Volumetric Efficiency" (page 26), proves that these conclusions are not true for those refrigerants having a high specific weight, such as for instance Freon-12.

Compressor Performance

5. The performance of a compressor departs to a marked degree from the theoretical. To determine the required piston displacement of a compressor or check the size of a compressor for a given refrigerating duty under given temperature and pressure conditions, it is necessary to evaluate as closely as possible the effect which the design of the compressor and the materials from which it is constructed have on the refrigerant gas flowing through it.

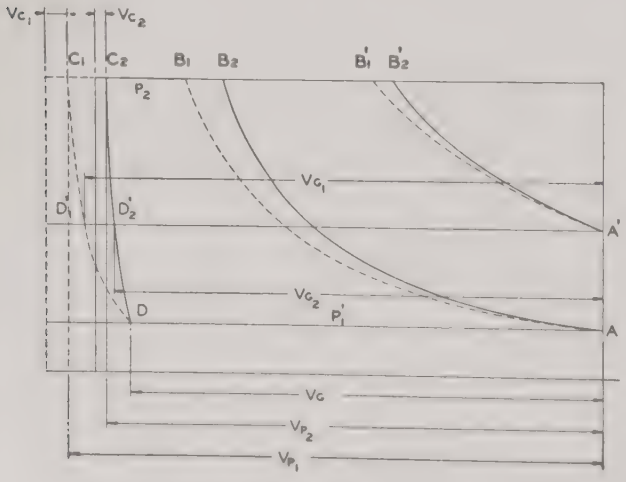


Fig. 9. Effect of Clearance

These two practical limitations demand that the piston displacement of the actual compressor be greater than the volume of the suction vapor drawn to the suction stop valve. The ratio of the volume of the suction vapor and the corresponding piston displacement is called the **total volumetric efficiency** (also real volumetric efficiency and charge volumetric efficiency).

Since the real volumetric efficiency of the actual compression cycle is always less than that of the theoretical cycle, it follows that the power consumption per unit weight of gas handled of the former is

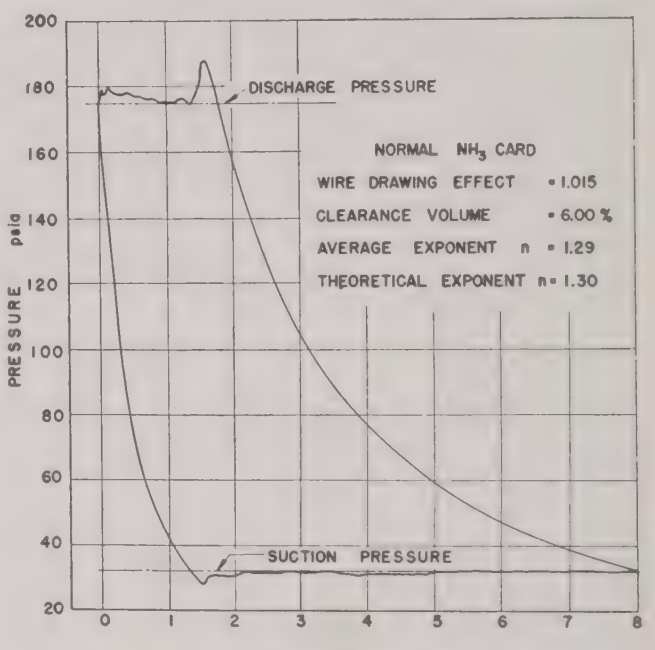


Fig. 10. Normal NH_3 Card

greater than that of the latter.

These departures from theoretical calculations are caused by the following factors:

A. Wiredrawing
B. Cylinder heating
C. Suction superheat
D. Water jacketing of compressor cylinders or cooling by radiation
E. Wet compression
F. Speed of rotation.

A. Wiredrawing. A compressor is expected to produce its rated capacity at specified conditions, usually measured at its suction and

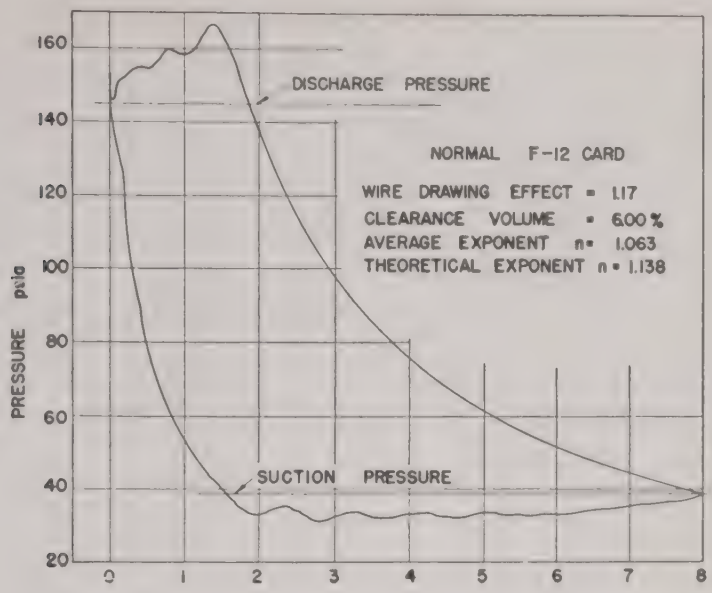


Fig. 11. Normal F-12 Card

discharge stop valves. For flow to take place, there must be pressure losses through the stop valves, compressor passages and compressor valves, including eddy currents within the gas. There must also be sufficient pressure differences across the valves to overcome spring and valve weight and inertia. The power expended on the gas to overcome these losses, occasioned by the design of the compressor, is called wiredrawing. It has the effect on the card (Fig. 5) of bowing the suction line down and the discharge line up, and increasing the indicated work. At the end of each stroke the piston velocity becomes zero, so the lines tend to return to points A and C. The wiredrawing losses add to the heating of the gas during its flow through the compressor.

Figs. 10 and **11** give samples of normal compressor indicator cards of well-designed ammonia and Freon-12 compressors, respectively. The wiredrawing effect is the work equivalent of the area of the indicator card above the discharge pressure line plus the area below the suction pressure line. This effect is generally expressed in terms of the ratio of the total indicator card area and that part lying between the actual suction and discharge pressure lines.

This ratio for the card of the ammonia compressor (Fig. 10) is 1.015, and of the Freon-12 compressor (Fig. 11), 1.17.

Since both these indicator cards are labeled "normal," an explanation of the great difference between the wiredrawing effects of the two must be given.

While the resistance offered to the flow of the refrigerant through the suction and discharge ports and valve housings adds to the wiredrawing, a considerable percentage of this work in a modern well-designed compressor is used up for accelerating the gas through the suction and discharge valves.

In a compressor so designed that the suction gas passes through suction valves in the piston, any initial velocity this gas may have had in the suction main is arrested and turned into heat before the suction stroke begins. The same is true of the gas before it flows through the discharge valves. Therefore the gas must be accelerated from zero velocity. In addition, basing the design of the valves entirely on the ratio of the piston displacement and the effective area of the valve opening is misleading, because the tension of the valve springs decreases this area, particularly at the lower speeds of the piston travel. Therefore a careful study of the loading of the valves, both from the point of view of their correct functioning and also from its effect on the work of wiredrawing, must be made to obtain all-round satisfactory compressor performance.

Again referring to the standard rating cycle, the data in Table 2 show that the weight of refrigerant circulated becomes 0.4079 and 3.671 lb per min per ton for

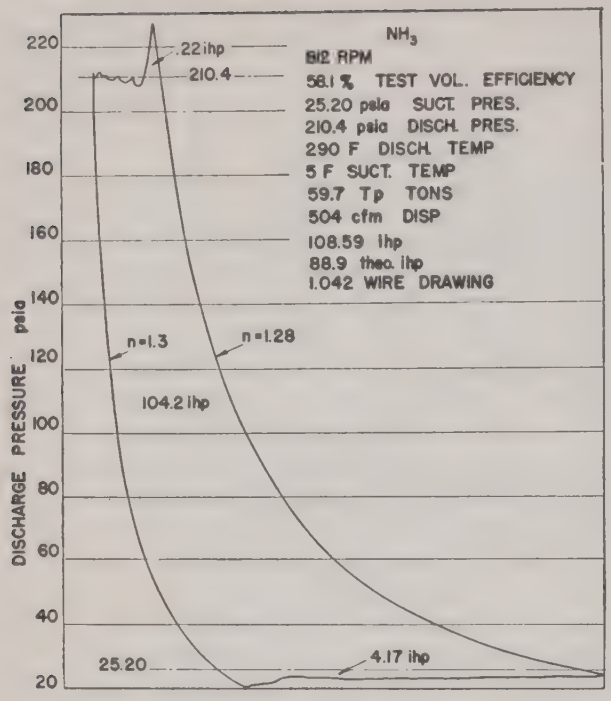


Fig. 12. Wiredrawing Effect—Normal NH_3 Card

ammonia and Freon-12, respectively. In this case about nine times as much Freon-12 by weight must be circulated. If the acceleration of the Freon-12 gas passing through the suction and discharge valves is the same as for the ammonia compressor, the wiredrawing effect of the former will be nine times that of the latter. Since the work of acceleration is generally more, the wiredrawing effect is further increased.

A comparison of the four indicator diagrams shown in **Figs. 12, 13, 14,** and **15** leads to a further explanation of the apparent excessive wiredrawing recorded on indicator cards of Freon-12 compressors.

All the indicator diagrams record the pressure and volume changes occurring during the compression cycle in a compressor of the same size operating at somewhat varying speeds and differing pressure ratios. Figs. 12 and 13 show these changes for ammonia gas and Figs. 14 and 15 for Freon-12 for high and low compression ratios, respectively. In **Table 5** appear some data obtained from the analysis of these diagrams.

The high wiredrawing effect shown on cards of Figs. 13, 14, and 15 cannot be accounted for by velocity changes alone. Furthermore, the apparent compression exponents for these three cards is too low. For Freon-12 it is practically isothermal. In addition, the theoretical ihp for the Freon-12 cards is in excess of the actual ihp between the suction and discharge pressure lines. The conclusion must be that

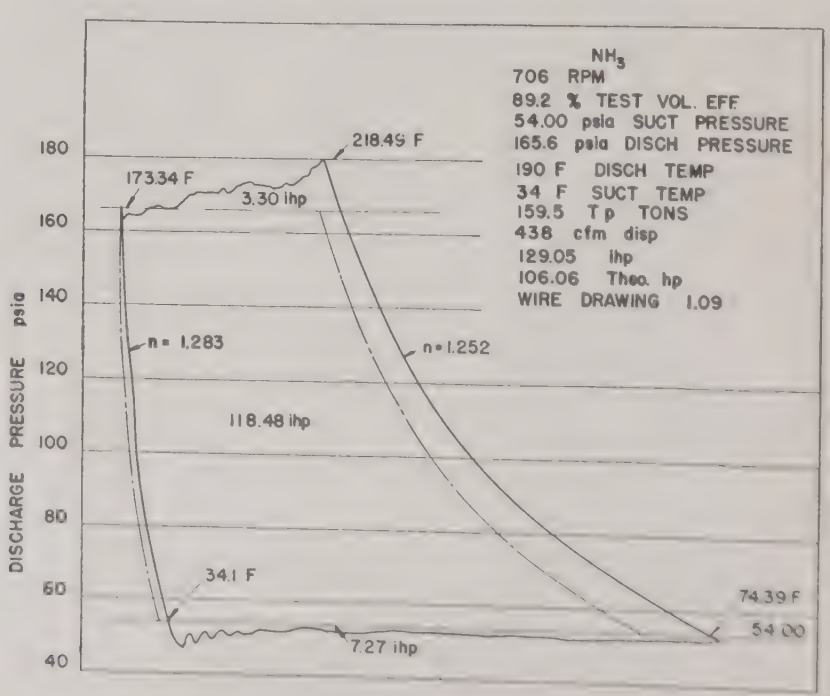


Fig. 13. Wiredrawing Effect—Normal NH_3 Card

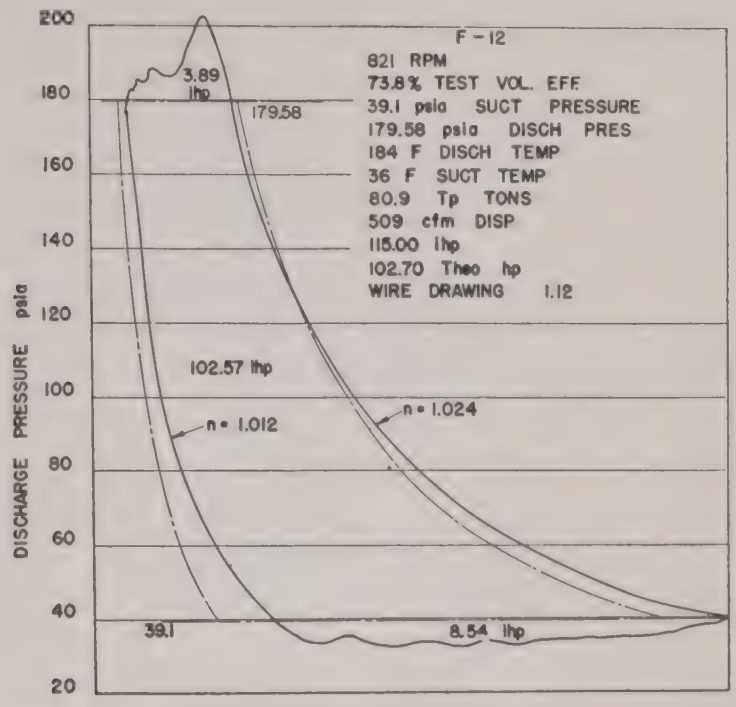


Fig. 14. Wiredrawing Effect—Normal F-12 Card

these indicator cards do not show the actual pressures in the compressor cylinder at the respective locations of the piston. Also, that this lagging action must be the result of the time necessary to overcome the inertia of the refrigerant entering and leaving the instrument. This action is particularly noticeable with Freon-12 on account of its high specific weight. The instrument itself may be partly responsible for these distortions.

Due to the fact that the piston comes to rest and reverses its motion at the end of the stroke, the inertia effect of the gas balances out so that the power measured on the indicator card will be reasonably close to the actual power developed in the compressor.

B. Cylinder heating. Another principal cause for deviation between actual and theoretical performance is the exchange of heat between gas and cylinder walls. Since the cylinder walls (including cylinder head and piston head) are in contact with alternately hot and cold gas (Fig. 7), they will tend to take up a mean temperature somewhere between the two extremes, such that the heat transfer to the walls during the time the gas is hotter than the walls will be exactly balanced by the heat transfer to the gas when it is cooler than the walls, plus the heat leakage outward to the atmosphere surrounding the cylinder, or, if jacketed, to the jacket water. The inner wall surface temperature will then fluctuate about this mean temperature during

Table 5. Comparisons of Ammonia and Freon-12 Indicator Diagrams

Fig. No.	12	13	14	15
Refrigerant	NH_3	NH_3	Freon-12	Freon-12
Compression ratio	8.36	3.0667	4.59	2.646
Compression ratio based on Freon-12	6.86	2.68	4.59	2.646
Wiredrawing	1.042	1.09	1.12	1.29
Apparent exponent $-n$				
for compression curve	1.28	1.252	1.024	1.012
for expansion curve	1.30	1.283	1.012	0.998
Actual ihp between pressure lines	104.2	118.5	102.6	107.4
Total ihp	108.6	129.0	115.0	138.4
Theo. ihp	88.9	106.1	102.7	108.8

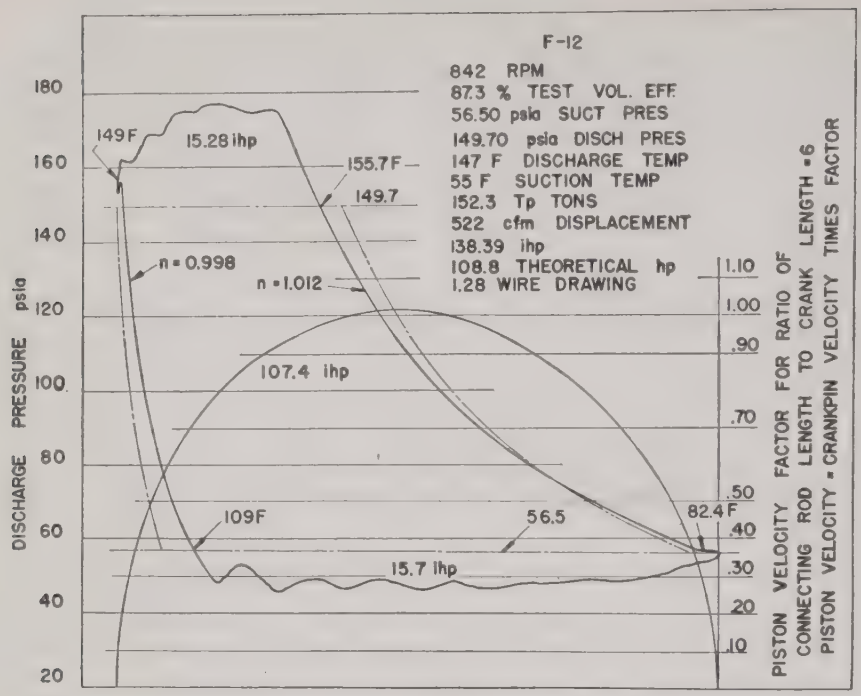


Fig. 15. Wiredrawing Effect—Normal F-12 Card

each revolution by such an amount as is called for by the heat transferred. Various parts of the wall are exposed for various periods during the revolution, so that the mean temperature taken up, and the amount of heat transferred per unit of wall area, will vary with the position of the piston. Wirth has measured the wall temperatures at various points in a compressor.

Fig. 16, which is schematic only, is adapted from Wirth. The curve marked t_g corresponds to Fig. 7 for an actual machine; t_w represents (mean) wall temperature; st_w exposed surface multiplied by mean wall temperature of that surface; and st_g the exposed surface multiplied by gas temperature. The vertical distance between the latter two is then proportional to the heat transfer, which is from the upper curve to the lower one. The dotted line at the beginning of the suction stroke is the temperature of the suction gas. The point where it crosses the dividing line between expansion and suction represents the temperature in the suction pipe. As it mixes with the re-expanded gas at higher temperature, and absorbs heat from the cylinder walls, it rises in temperature.

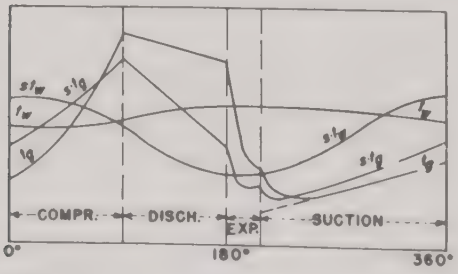


Fig. 16. Time-Temperature Diagram for Actual Case

The theoretical volumetric efficiency is based on having suction gas at the start of the compression stroke at the same temperature as in the suction pipe. The real volumetric efficiency takes into account the temperature of the gas at the end of the suction stroke, and the increased specific volume at that temperature. It has been shown by calculation that the ratio of indicated work to heat absorbed during the suction stroke in a single-stage ammonia compressor varies from about 2.6 to 3.8 with suction pressures between 35 and 5 psig, and with discharge pressures varying from 145 to 205 psig. The discharge pressure makes little difference in this ratio, although much in the actual temperature rise. The heat absorbed per lb of NH_3 varies between 29.7 and 54.6 Btu, corresponding to temperature rises of roughly 55 and 100 F.

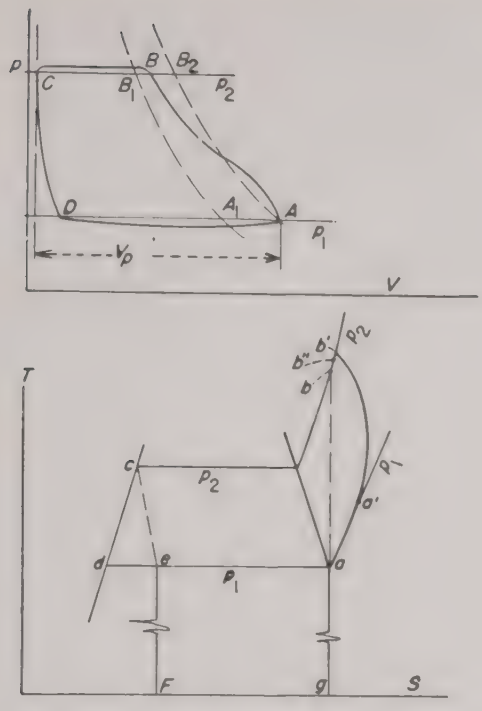


Fig. 17a. Effect of Heat Absorption in Cylinder Walls (P-V)
Fig. 17b. Effect of Heat Absorption in Cylinder Walls (T-S)

During the first part of the compression stroke, more heat is absorbed, but once the gas temperature exceeds the wall temperature, the transfer is reversed, and for the remainder of the compression stroke, the discharge stroke, and the start of the expansion stroke, the gas is giving up heat to the wall. The quantity of heat absorbed by the walls must be equal to the heat previously given up to the gas, plus the heat withdrawn through the walls. Since the time during which this takes place is considerably less than that during which the gas is being heated, the temperature differences must be greater, and the mean temperature of the wall must be closer to the suction stroke gas temperature than to the discharge stroke gas temperature.

Considering the effects on the indicator card and the temperature entropy diagram, we have **Figs. 17a** and **17b.** In Fig. 17a, exaggerated for clarity, DA represents approximately the volume of suction gas at the end of the suction stroke, and DA_1 the volume of the same gas in the suction pipe. The total volumetric efficiency is then DA_1/V_p and the ratio DA_1/DA is the volumetric efficiency due to cylinder heating. Dashed lines A_1B_1 and AB_2 are lines of constant entropy, put in for comparison with the compression line AB. The actual mean discharge temperature is somewhat less than is indicated by point B, since the gas remaining in the cylinder continues giving up heat to the cylinder walls throughout the discharge stroke. The area above line p_2 and below line p_1 represents the wiredrawing loss.

In Fig. 17b, point a represents the condition in the suction pipe, a′ the end of the suction stroke, a′b′ the compression stroke, and b′b″ the discharge stroke, the drop being the cooling by the cylinder walls. The mean temperature of the discharge gas will fall between b′ and b″. Area aa′b′cda does not represent the work of compression but the area e-f-g-a-e, which is fore-shortened, represents the refrigerating effect. The excess of b' over b is the increase in work over that of adiabatic compression, in addition to the wiredrawing as measured on the card.

During re-expansion, the gas is first cooled by the cylinder walls, and then heated. It follows a curve of the general shape shown for expansion in Fig. 17.

C. Suction superheat. Superheat increases the specific volume of the suction vapor. It also increases the total heat of the vapor and hence the refrigerating effect per pound of refrigerant flowing through the cycle, provided the heat absorbed by the vapor in being superheated represents useful refrigeration. Cooling the liquid refrigerant, on its way from the receiver to the evaporator, in countercurrent flow with the suction gas before it reaches the compressor, produces a useful refrigerating effect.

The data presented in **Tables 6** and **7,** also in **Fig. 18,** demonstrate that the refrigerating capacity of Freon-12 compressors increases and the power per ton decreases as the suction gas superheat increases. The reasons for these facts appear clearly from an analysis of the test data in Table 6, the results of which are recorded in the curves shown in **Fig. 19.**

To make this analysis, the equation given in section 4 of this chapter (page 15), showing the relation in the theoretical cycle of the weight of gas, w_s, flowing

Table 6. Test Results on a 15-hp, 1,150 rpm, Completely Enclosed Freon-12 Compressor with a Water-Cooled Motor at 130 psi Discharge Pressure and 37 psi Suction Pressure

(Calculations based on constant volumetric efficiency and constant entropy during compression)

Superheat entering compressor	9.5 F		25.5 F		39.7 F		54 F	
	Test	Calculated	Test	Calculated	Test	Calculated	Test	Calculated
Capacity	100%	100%	104.0	100.5	106.6	101.0	110.0	101.5
Motor input	100%	100%	98.6	100.3	97.5	100.6	96.7	100.9
Total volumetric efficiency	74.5	74.5	77.2	74.5	78.8	74.5	81.0	74.5
Btu per hr per watt	100%	100%	105.5	100.3	109.3	100.5	113.7	100.6
Lb of refrig circulated per min	100%		100		98.6		98.05	

through it and the weight of gas, w_c, recirculated in the cycle, is used.

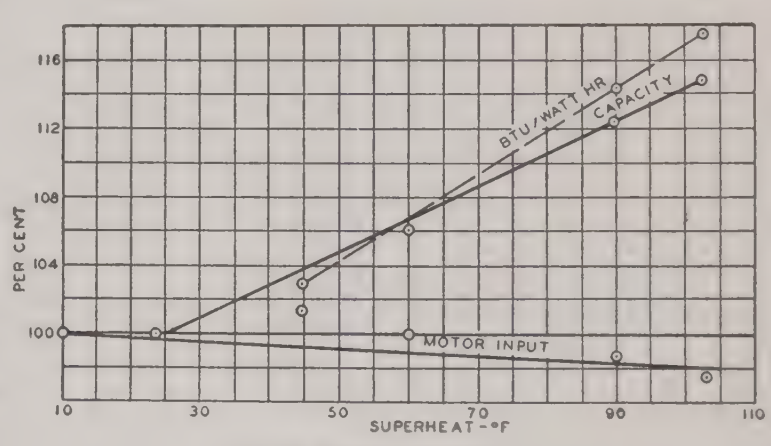
The equation is:

$$w_s = \frac{\frac{m}{100} R^{1/n} w_s}{E_1}$$

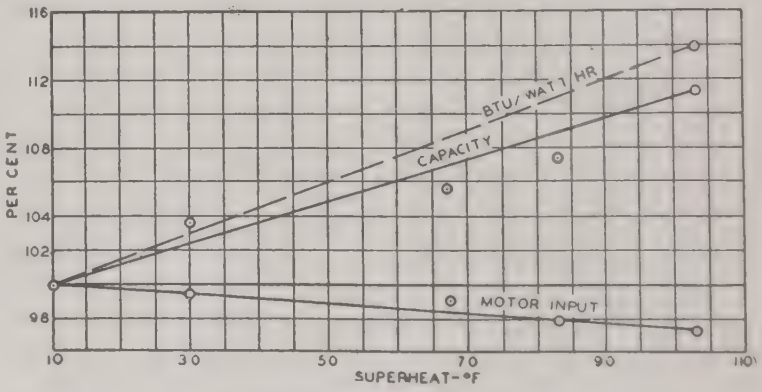
E_1 is the volumetric efficiency of the theoretical cycle. It is equally true of the actual compression cycle if the real volumetric efficiency, E, is substituted for E_1, provided the compressor tested is operated at a speed of rotation which will allow a reasonably accurate functioning of the suction and discharge valves, and has been operated long enough so that the leakage through suction and discharge valves and past the piston is reduced to a minimum consistent with its size and design.

$$E = \frac{v_s}{v_d}$$

where v_s is the volume of suction gas drawn into the compressor and v_d is the compressor displacement.



36 psi suction pressure
110 psi condensing pressure



14.6 psi suction pressure
110 psi condensing pressure

Fig. 18. Test Results of ¾-hp Air-Cooled Condensing Unit at Different Suction Pressures

The compressor displacement combines in it the effect that compressor design and operation have on the refrigerant flowing through it.

The equations used in this analysis are therefore the following:

$$w_{c1} = \frac{\frac{m}{100} R^{1/n} w_s}{E_1} = \frac{\frac{m}{100} R^{1/n} w_{c1}}{V_s},$$

used to find the end points of the curves on Fig. 19.

$$w_c = \frac{\frac{m}{100} R^{1/n} w_s}{E} = \frac{\frac{m}{100} R^{1/n} v_d}{V_s}$$

$$V_r = \frac{\left(1 + \frac{m}{100}\right) V_s}{E + \frac{m}{100} R^{1/n}}$$

w_{c1} = the weight of recirculated gas in the theoretical compressor
v_{d1} = the displacement of the theoretical compressor
V_s = the specific volume of the suction gas
V_r = the specific volume of the gas at the end of the suction stroke
m is taken at 6%

Assuming that the capacity of the compressor was 10 tons of refrigeration under the test results and conditions given in column 1 of Table 6, the compressor displacement is found to be 58.38 cfm. The cylinder heating effect for the four tests, when plotted against the volume of suction gas flowing through the compressor in cfm, gives curve I, whose end point is zero cylinder heating for a theoretical volume of suction gas flowing through the compressor of $58.38 \times 0.879 = 51.315$, where $0.879 = E_1$ is the theoretical volumetric efficiency of the compression cycle.

Curve II shows the plot of the real volumetric efficiencies recorded in the test results also plotted against the volume of suction gas. The end point of this curve is at the theoretical volume of 51.315 cfm and $E_1 = 0.879$.

In curve III the suction gas temperature is plotted against the volume of suction gas. The end point of this curve, which shows a suction gas temperature of 142.2 F at a suction gas volume of 51.315 cfm, is found as follows:

The saturated suction gas temperature at a pressure of 37 psia is 21.8 F. The maximum theoretical liquid Freon-12 cooling will be obtained when the liquid is reduced in temperature from the saturation point at the condenser pressure to the saturation point at the suction pressure. This reduction in heat of the liquid must be gained by the suction gas. In this case the corresponding theoretical suction gas temperature is 142.2 F.

Table 7. Test Results on a 20-hp, 600-rpm, Four Cylinder V-Type Open Belted F-12 Compressor with Water Jackets

(124 psi discharge pressure, 37 psi suction pressure)

Superheat	10 F	40 F	55 F	85 F
Capacity, %	100	108.5	112.1	117.7
Motor input, %	100	101.9	101.9	100.4
Btu per hr per watt	100	106.5	110	117.2

Refrigerating capacity for the four tests is plotted against suction gas temperature in curve IV. The end point is found at 142.2 F and 12.36 tons of refrigeration. Knowing the thermodynamic properties of Freon-12 at 37 psia and 142.2 F and that the volumetric efficiency at this point is $E_1 = 0.879$, the refrigerating tonnage, 12.36, is readily determined.

Curve V is found by plotting the weight of Freon-12 flowing through the compressor against the suction gas temperature. The end point of this curve is found at a gas temperature of 142.2 F and a weight of 36.5 lb per min. This weight has been calculated, using the above formulas and the end point data found.

The above analysis can be applied to determine whether or not the capacity of an ammonia compressor increases or decreases as the suction gas superheat increases. Having available one test point on the capacity curve, IV, for the ammonia compressor, the end point on it can be calculated.

For example, a 10×10 2-cylinder VSA ammonia compressor, operating at 360 rpm with a suction gas temperature of 15 F, a suction pressure of 20 psig and a discharge pressure of 155 psig, produces a refrigerating effect of 80.6 tons. The theoretical end point for its capacity curve will be reached when the suction gas has a temperature of 168.14 F. The corresponding refrigerating capacity is 79.2 tons. The reduction in capacity from the given test point to the final end point is 1.74%.

For any normal increase in suction gas superheat of an ammonia compressor, its

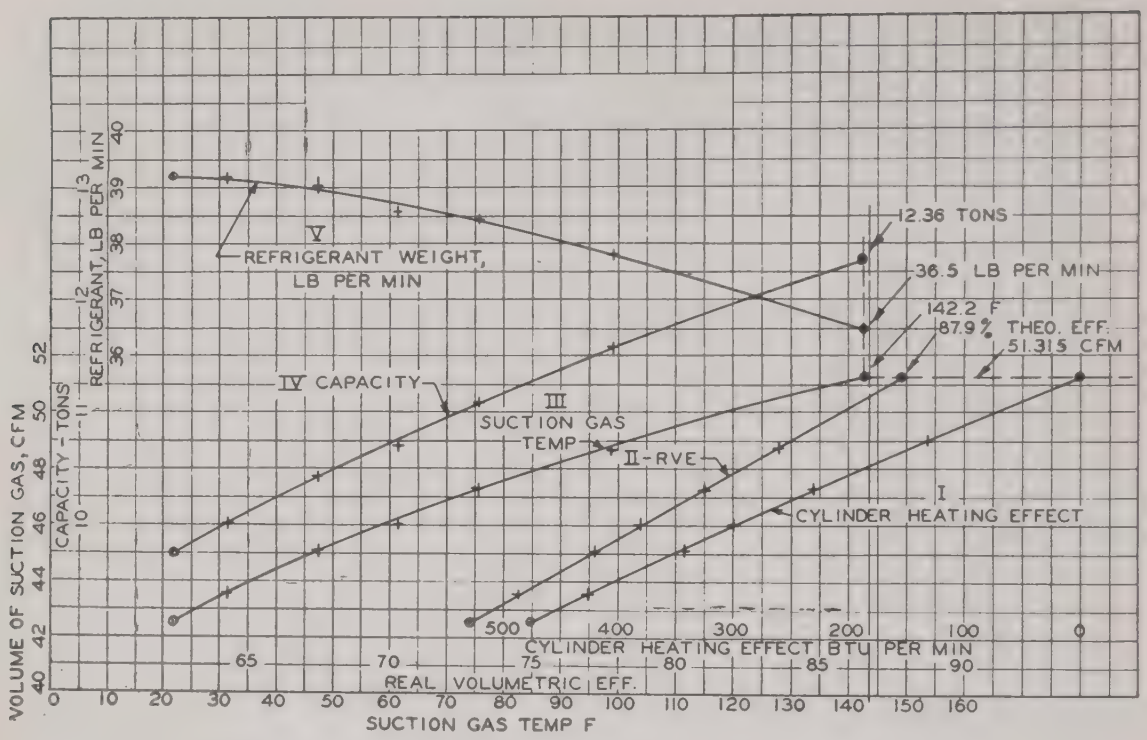


Fig. 19. Effect of Suction Gas Superheat-Analysis of Test Data in Table 6

refrigerating capacity decreases very little as long as the superheat is gained by the suction gas doing useful work such as cooling the liquid ammonia before it reaches the evaporator.

The curves drawn on Fig. 19 establish the fact that when the suction gas superheat increases the cylinder heating effect decreases until it becomes zero at the end point (see curve I). The end point is reached when the volume of suction gas handled becomes equal to the compressor displacement multiplied by the theoretical volumetric efficiency of the compression cycle. At this point the heat added to the suction gas, from its saturation temperature on, is equal to the heat removed from the liquid refrigerant when cooled from its saturation temperature to the saturation temperature of the suction gas.

In practical compressor operation, the maximum useful refrigeration obtainable by the exchange of heat between the suction gas and the liquid is reached when the temperature of the former is raised to the temperature of the liquid as it flows from the condenser.

The plots in Fig. 18 give the increase in capacity and Btu per whr, also the corresponding decrease in motor input, of a small Freon-12 compressor when operated with increasing suction superheat.

D. Water jacketing of compressor cylinders. If the temperature of the jacket water is below the mean wall temperature, the effect of jacketing is to decrease the wall temperature, and therefore decrease the amount of heating during the suction stroke, and also the temperature of the discharge gas. The total volumetric efficiency is increased, and since the entire line ab′ in Fig. 17b is shifted to the left, the work per pound is decreased. Since the machine handles more gas, the power input is not decreased, although this is not true of the work per pound of refrigerant, or the power per ton. Since the wall temperature varies with location, those parts having the highest temperature are most benefited. Obviously, it is not beneficial to jacket walls whose temperature is as low as, or lower than, that of the jacket water. This is reflected in single-acting practice, where the suction valve is in the piston, and only the upper part of the compressor cylinder and the lower part of the discharge chamber are jacketed. Wirth shows that where the mean wall temperature is lower than

the condensation temperature corresponding to the discharge pressure, losses similar to those with wet compression occur. Fischer shows improvement with a slow-speed horizontal compressor by jacketing the entire cylinder when compressing superheated vapor, the contrary with wet suction.

Jacketing shows the greatest advantage when high discharge temperatures are encountered, hence ammonia compressors are provided with jackets. Freon-12 compressors are rarely jacketed, usually being provided with radiating fins, or cooling devices are entirely dispensed with, since, due to low adiabatic constant, discharge temperatures are low (Fig. 3).

It is customary to operate ammonia compressors with suction gas at least 10 degrees above the saturated vapor temperature corresponding to the suction pressure. The purpose is to bring to the compressor dry suction gas. Experience indicates that even at this elevated temperature the suction gas is not always dry.

The analysis of the effect of suction gas superheat on the capacity of the ammonia compressor recorded in the preceding section, C, taken together with the heat absorbed by the jacket water, permits the conclusion that even a higher degree of useful suction superheat will not impair the capacity of the compressor. The greater superheat will insure dry suction gas. The higher temperature during the compression cycle will increase the real volumetric efficiency, since the jacket water will remove more of the heat absorbed by the cylinder walls.

E. Wet compression. Theoretically, wet compression is more efficient than dry compression. Referring to Fig. 6, if the line on which points g, a, and b lie were moved to the left until point b were on the saturated vapor line, point a would represent wet suction vapor, and the ratio of the work to refrigerating effect would be a minimum. As compression takes place the contained liquid would be evaporated.

In practice, this is not the case, because of the time element involved in the evaporation. The liquid exists not in molecular suspension but in the form of droplets, and on compression the vapor superheats while the droplets remain in existence. Some evaporation takes place by heat transfer from the gas, but at the end of the discharge stroke a considerable amount of the liquid may remain and continue to evaporate during the expansion stroke. Furthermore, this liquid is now at the higher temperature corresponding to the discharge pressure, and a considerable portion

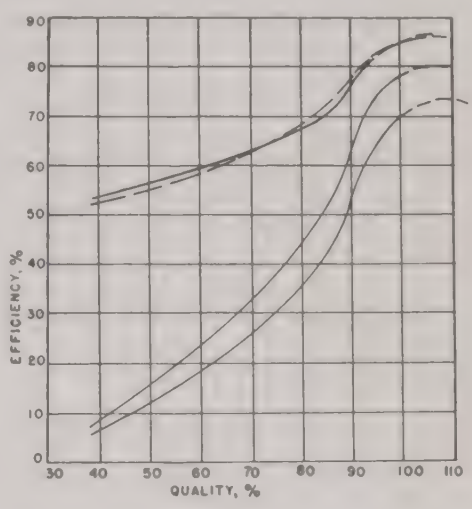


Fig. 20. Effect of Suction Gas Quality on Volumetric Efficiency

Top lines, apparent volumetric efficiency; lower lines, compression efficiency.

of it evaporates almost instantaneously because the heat necessary is contained in the droplet, and need not be absorbed by heat transfer. The action is similar to flashing on passing through an expansion valve. The expansion stroke with wet compression therefore involves an increase in volume of the refrigerant remaining in the clearance space, reducing the volumetric efficiency without a proportional decrease in power.

Fig. 20 shows the effect of varying the quality of the suction gas, as determined by Fischer.

With ammonia, wet compression is particularly objectionable as the liquid tends to wash oil from the cylinder walls. In enclosed compressors the liquid will run into the crankcase, causing oil foaming and excessive oil pumping.

F. Speed of rotation. For the theoretical compression cycle the refrigerating capacity, the bhp and the volumetric efficiency remain constant per cubic foot of compres-

sor displacement for all speeds of rotation. This is not true for the actual compression cycle.

How these factors vary with speed for a Freon-12 compressor the curves of **Figs. 21** and **22** illustrate. The upper curves show the increase in bhp and refrigerating capacity as the speed increases. The test results are recalculated for a speed of 1,300 rpm and those at 1,300 rpm made equal to 100%. The lower curves show the variation with speed of the bhp per ton and the real volumetric efficiency, respectively.

The bhp and bhp per ton and capacity increase with speed, whereas the real volumetric efficiency increases until the speed of 1,300 rpm is reached and after that decreases.

These variations from the theoretical are brought about mainly by two characteristics inherent in the operation of reciprocating compressors. The one is back leakage of gas through the suction and discharge valves and the piston. This leakage is not proportional to the speed of rotation but depends on the mean effective pressures existing in the compressor cylinder. Its amount increases for the compressor results in Fig. 22 from 300 to 1,700 rpm about 25%, whereas the volume of suction gas handled for the same range of speed increases about 580%.

Considering this characteristic alone the real volumetric efficiency should continue to increase as the speed increases. But the wiredrawing effect interferes. The wiredrawing effect is a function of the volume of gas flowing through the compressor. It is the ratio of the total ihp and the ihp between the actual suction and discharge pressures. When the wiredrawing effect increases the cylinder heating effect also increases.

At the low speed the leakage per cubic foot of gas handled is high. Its effect is to add heat and gas volume during the suction stroke, reducing the volume of suction gas drawn into the compressor. At the same time the effect of wiredrawing on cylinder heating is small, but the action of the sum of the two in reducing the volume of gas drawn in per revolution is greater at that speed than at any intermediate speed up to 1,300 rpm. As the speed of rotation increases, the adverse effect of leakage decreases more rapidly than the increase in cylinder heating due to wiredrawing. At or near 1,300 rpm, the effect of these two adverse conditions is a minimum per revolution and therefore the real volumetric efficiency a maximum.

The real volumetric efficiencies at 1,300 rpm are only 3.9% and about 1% greater than those at 300 and 900 rpm, respectively, for the two tests cited.

Some manufacturers of Freon-12 equipment start the rating of their compressors at this critical speed and limit the maximum speed to about 133⅓% of this speed, as, for instance, for the compressor described in Figs. 21 and 22 from 1,200 to 1,750 rpm. For a larger compressor it may be from 660 to 880 rpm.

The Freon-12 compressor, of which the test results are shown in Figs. 21 and 22, has four cylinders, 2¼-in. bore and 1 11/16-in. stroke; cylinder and cylinder heads are finned. The clearance volume is 9% and the maximum operating speed 1,750 rpm.

The curves plotted in **Fig. 23** are obtained as follows:

I—from test results
II—by calculation
III, IV, and V—by use of the weight formulas given in section 5C of this chapter (page 23).

The prolongation of curve III to zero rpm appears to indicate that while maintaining 51.7 psi suction pressure before the suction stop valve and 140.9 psi discharge pressure after the discharge stop valve the total gas leakage will be equivalent to a heat flow of 24 Btu per min at that point.

Real Volumetric Efficiency

6. The effect volumetric clearance, cylinder superheating, leakage, wiredrawing and all other compressor design and operating features have on the value of the real volumetric efficiency varies with type of refrigerant used.

The refrigerants having high specific weights and low enthalpies show the greater variation from the theoretical compression cycle.

Two cases are given here. The first is the relation between the absolute pressure ratio and the real volumetric efficiency and

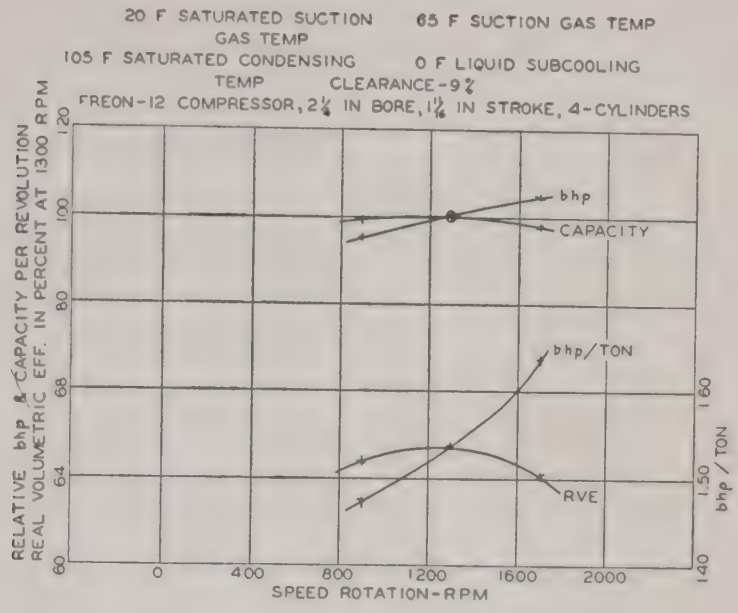
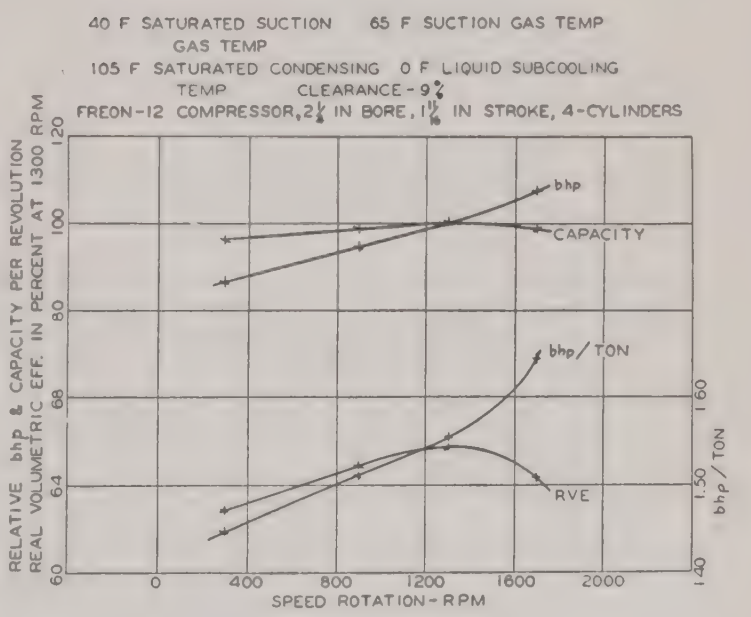


Fig. 21. Effect of Varying Speed of Rotation on Capacity, bhp and RVE

Fig. 22. Effect of Varying Speed of Rotation on Capacity, bhp and RVE



the second the effect of clearance volume on the refrigerating capacity.

(1). **Relation** between the **absolute pressure ratio** and the **real volumetric efficiency** of Freon-12 compressors.

Figs. 24 and 25 present various curves derived from tests on a Freon-12 compressor. The real volumetric efficiency curve plotted on Fig. 24 is obtained by varying the suction and discharge pressures. The test points marked are all within the range of allowable error so that the path of the curve accurately pictures the efficiency for the compressor tested for a considerable variation in these pressures. For comparison the curve of the theoretical volumetric efficiency is added. The lefthand dotted extension, to the line $p_2/p_1 = 1$, indicates that the real volumetric efficiency at this point is approximately 94%.

The curves on Fig. 25 show the real volumetric efficiency and the bhp per ton for the same compressor plotted against the saturated suction temperature.

The data presented on Figs. 21 and 22 and Figs. 24 and 25 were obtained from tests made on the same size Freon-12 compressor. In the former the clearance volume was 9% and in the latter $4\frac{1}{2}$%. The short curves on Fig. 25 show the effect of this difference in clearance volume on the bhp per ton and the real volumetric efficiency at a saturated condensing temperature of 115 F.

As a comparison the curves on **Fig. 26** give values of the real volumetric efficiencies of three two-cylinder, vertical, single-acting ammonia compressors plotted against the absolute pressure ratio. Here

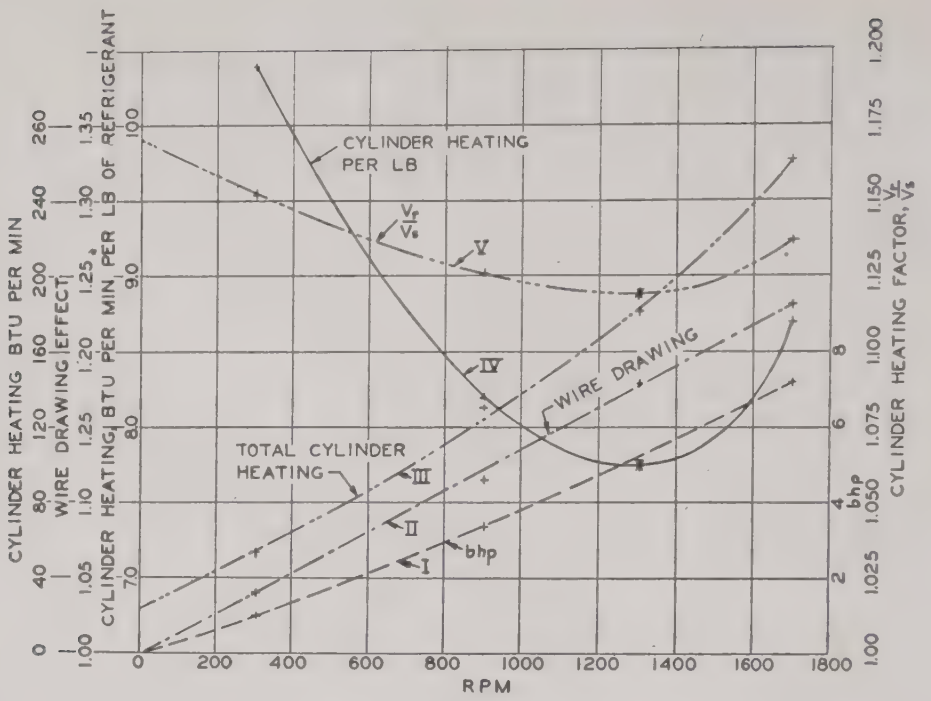


Fig. 23. Analysis of Test Results Shown in Fig. 22

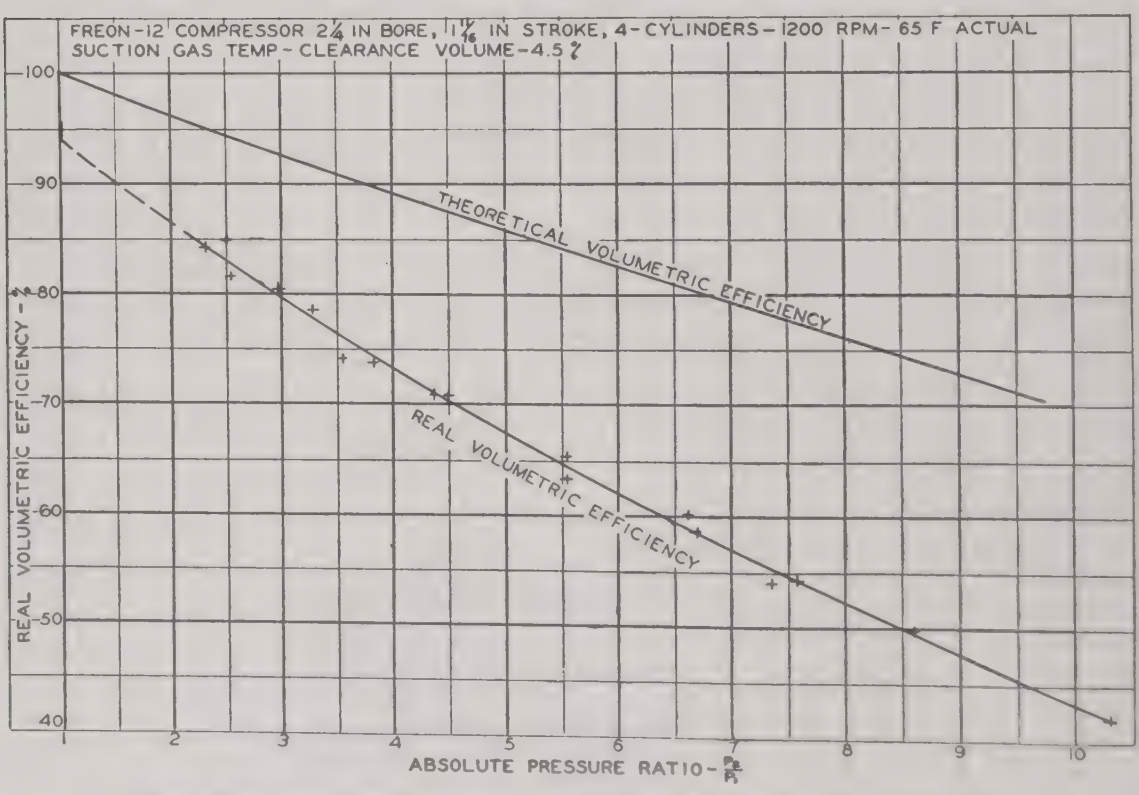


Fig. 24. Variation of Real Volumetric Efficiency with the Absolute Pressure Ratio

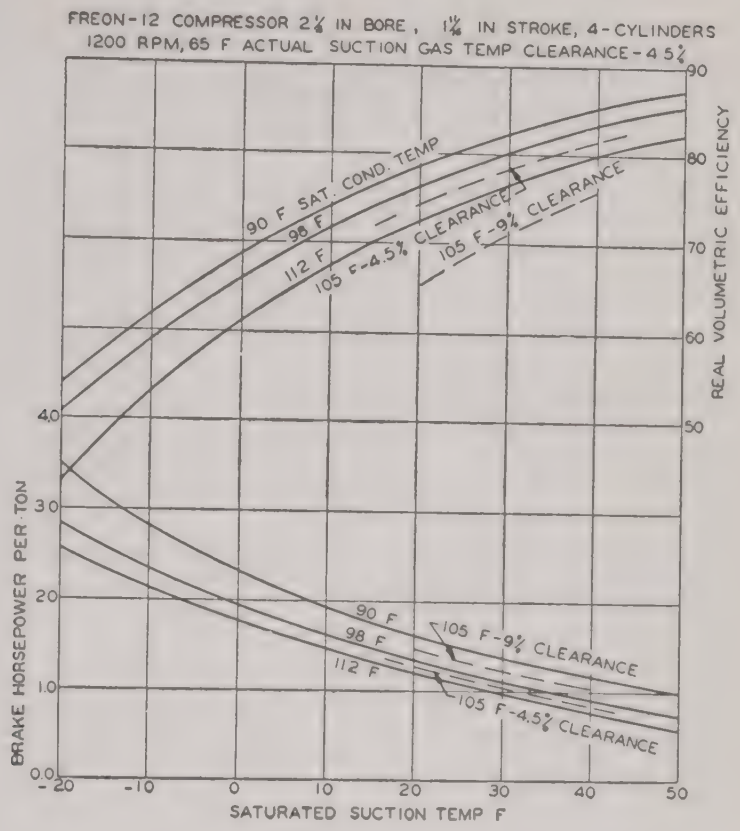


Fig. 25. Variation of Real Volumetric Efficiency with Suction and Discharge Pressures

the test points, particularly for low compression ratios, are located some distance away from a common path drawn through them. For these compressors no single curve will correctly represent the relation between the real volumetric efficiency and the absolute compression ratio for all suction pressures.

The thermodynamic properties of the two refrigerants account for this difference in the behavior of the two gases. As the temperature of the suction gas increases the specific volume of Freon-12 increases less than its enthalpy, whereas for ammonia the reverse is true. As the suction pressure rises the Freon-12 gas can absorb more of the cylinder heating effect than the ammonia gas, due to the increase in wiredrawing for each degree rise in temperature.

(2) The **effect** of **clearance volume** on the **refrigerating capacity** of Freon-12 compressors.

Fig. 27 presents tests of a 6-in. × 5-in. 4-cylinder Freon-12 compressor operating at 500 rpm. The solid and dashed line curves show the capacity and bhp test results plotted against the saturated suction gas temperature for this compressor with a volumetric clearance of 4.7 and 7.1%, respectively. Tests were run with saturated condenser temperatures of 82, 98 and 112 F.

The two capacity curves of different volumetric clearance intersect at points A, B, and C. An analysis of these test results with the aid of the weight formulas given in section 5C of this chapter (page 23) points to an explanation of this behavior of the compressor. For the theoretical cycle, the greater clearance compressor will always have a lesser refrigerating capacity than the smaller clearance compressor when operated under the same pressure conditions.

The result of the analysis appears in **Fig. 28**. The four curves shown give the change in the cylinder superheat factor,

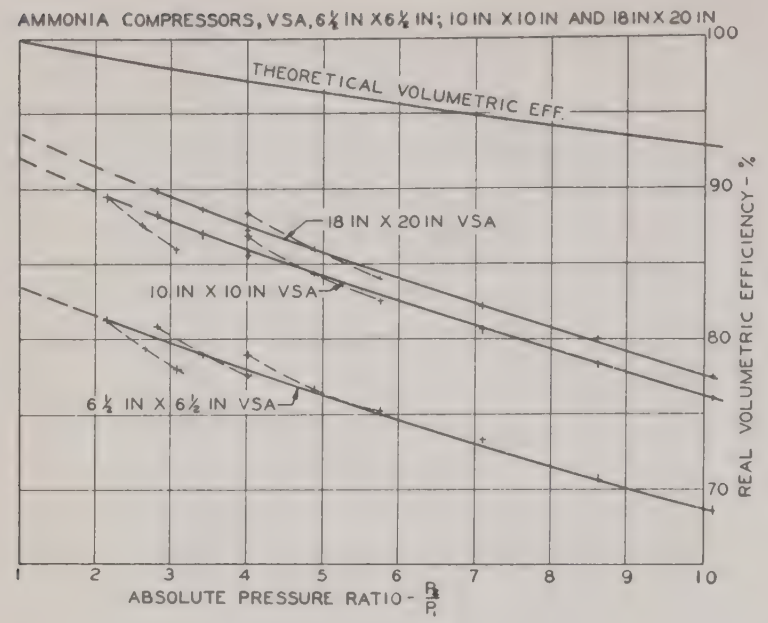


Fig. 26. Variation of Real Volumetric Efficiency with the Absolute Pressure Ratio

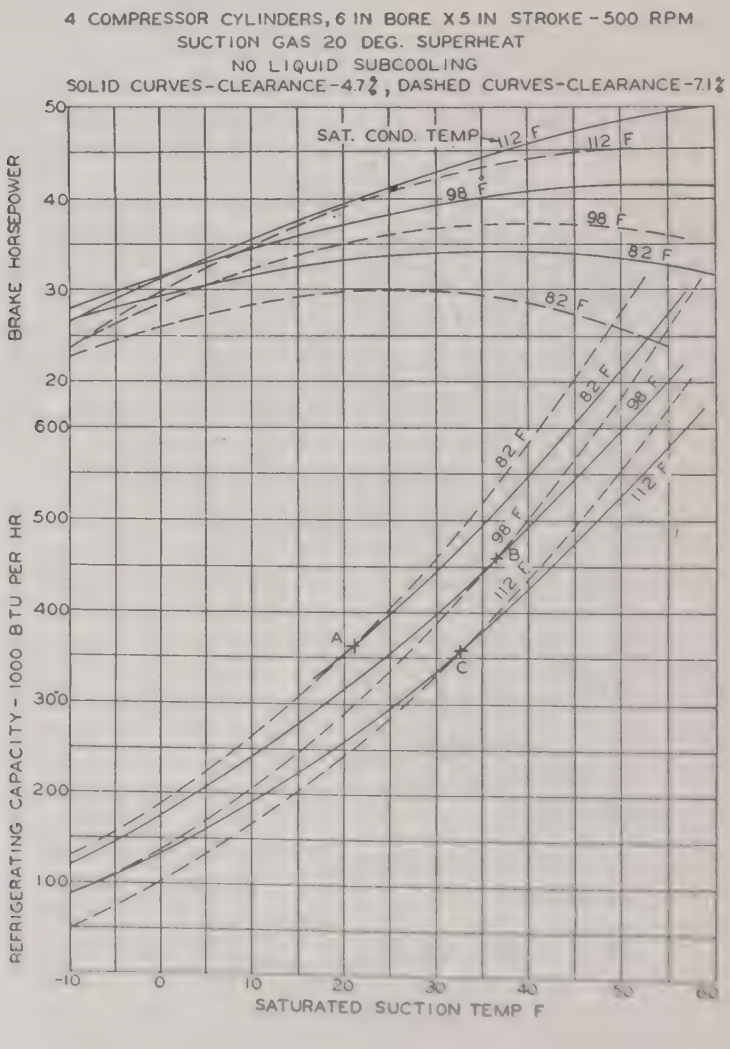


Fig. 27. Effect of Varying Clearance Volume of Freon-12 Compressors on Capacity and Bhp

V_r/V_s, plotted against the saturated suction gas temperature for the 98 F and 112 F saturated condenser temperatures.

The reason for this behavior of Freon-12 compressors is explained by the fact that the weight of recirculated gas plus the weight of the gas discharged by the compressor is greater for the compressor having larger clearance. For the same wiredrawing and somewhat less ihp at the intersecting points, it is evident that the adverse effect of the larger clearance volume is nullified by a lower temperature of the gas at the end of the suction stroke. The result is that at these intersection points the real volumetric efficiency is the same for both. For a further increase in suction pressure, the refrigerating capacity of the larger clearance compressor exceeds that of the smaller clearance compressor. To illustrate, for intersection point A the analysis presents the following results:

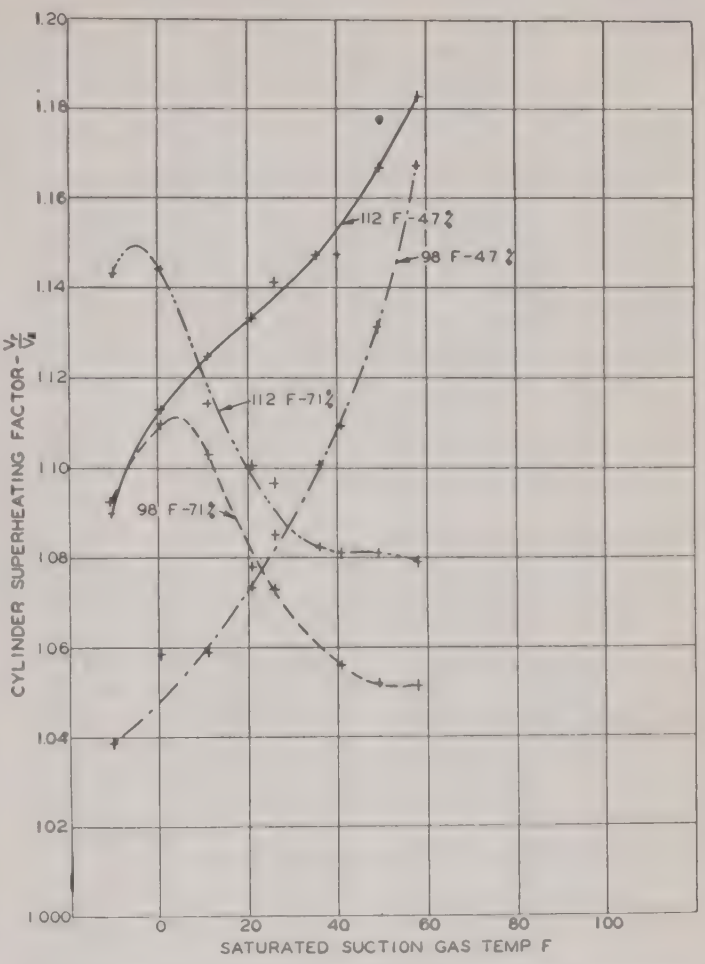


Fig. 28. Effect of Clearance Volume on Cylinder Superheating in Freon-12 Compressors

Clearance volume, %	4.7	7.1
Suction pressure, psia	47.7	47.7
Condenser pressure, psia	128.0	128.0
Refrigerating tonnage	41.75	41.75
Real volumetric efficiency	0.842	0.842
Weight of gas discharged from compressor, w_s, lb per min	153.49	153.49
Weight of recirculated gas, w_c, lb per min	20.36	30.75
$w_c/w_s \times 100$	13.3	20.0
$w_s + w_c$	173.85	184.24
Cylinder heating during suction stroke, Btu per min	1,080	693.5

These calculations affirm that the cylinder heating effect is less for the large-clearance than for the small-clearance compressor in sufficient amount to counteract the greater re-expansion volume of the former.

No single family of curves, based on actual results of tests made on one compressor using any one of the refrigerants, can accurately predict the capacity, volumetric efficiency and other properties of an operating compressor. Effective valve area, clearance, internal cylinder surface in comparison to the gas handled, and many other features of design and operation conspire to make this impossible. There is, however, one basic property of compressors which allows the prediction of capacity, real volumetric efficiency, etc. If a group of compressors varying in size from the smallest to the largest are all based on the same design features, have the same clearance volume, effective valve area, and a somewhat fixed relation of piston diameter to stroke, etc., these predictions are possible. In other words, there is a fixed relation between the various sizes of these

properties. To illustrate this fact, the calculations in **Table 8** are presented.

Three sizes in a line of vertical single-acting compressors, ranging from a 3-in. × 3-in. to an 18-in. × 20-in. are chosen. They are the $6\frac{1}{2} \times 6\frac{1}{2}$, the 10×10 and the 18×20. Assuming that a complete set of tests at suction pressures ranging in close steps from 0 to 45 psig and condenser pressures ranging from 85 to 245 psig have been run on a 10×10 compressor and one test each, say at 20 psig and 155 psig of all the other sizes of the group, complete tables of the properties of all of them can be compiled. Table 8 shows that a constant ratio exists between the real volumetric efficiencies at all suction and discharge pressures between the 18×20 and the 10×10 compressors, between the 10×10 and the $6\frac{1}{2} \times 6\frac{1}{2}$ compressors, as well as between the 18×20 and the $6\frac{1}{2} \times 6\frac{1}{2}$ compressors. The same is also true of the cylinder heating factor during the suction stroke and the refrigerating capacity.

The same relation may exist between groups of Freon-12 compressors of similar design.

Compound Compression

7. Referring to **Fig. 29**, a-b-c-d is a card for simple compression. In compounding, due to the lower compression ratio, point d would be at d′ and a at a′, for the same gas volume. That is to say, a smaller cylinder can be used on account of higher volumetric efficiency. The card for the low stage then is a′b′c′d′. The low-stage discharge is cooled, reducing its volume to point e. The high-stage card is then e-f-c-g. The area of the compound cards is less than that of the simple card, showing a reduction in indicated horsepower. These cards are theoretical; there are losses in compound compression not found in the simple machine. There are two sets of wire-drawing losses, plus pressure drops through the intercooler and lines, all of which reduce the theoretical savings.

On refrigeration service, compounding offers economies apart from the compression efficiency by cooling liquid refrigerant with the high-stage suction, and by cooling the first-stage discharge, after it leaves the water-cooled intercooler, to a point close to saturation, also by using high-stage suction for direct useful refrigeration. **Fig. 4** shows theoretical power requirements with compound compression. **Fig. 30** is a schematic diagram of a compound system. **Fig. 31** gives tonnage and power of low-stage or booster compressors. These are not test readings, but represent ratings which have

Table 8. Relation between the Real Volumetric Efficiencies and the Cylinder Heating Factors of VSA Ammonia Compressors

Suction press, psig		5			20			35			50		
Discharge press psig		125	155	185	125	155	185	125	155	185	125	155	185
RVE	#1—18 × 20	0.820	0.798	0.774	0.883	0.858	0.838	0.900	0.886	0.870			
	#2—10 × 10	0.805	0.782	0.760	0.867	0.842	0.824	0.883	0.870	0.855	0.895	0.875	0.859
	Ratio $\frac{\#1}{\#2}$	1.019	1.021	1.019	1.019	1.019	1.018	1.019	1.019	1.019			
	#3—$6\frac{1}{2} \times 6\frac{1}{2}$	0.732	0.7115	0.686	0.789	0.766	0.751	0.808	0.790	0.776	0.813	0.795	0.780
	Ratio $\frac{\#2}{\#3}$	1.100	1.100	1.109	1.099	1.100	1.098	1.092	1.102	1.102	1.101	1.101	1.101
	Ratio $\frac{\#1}{\#3}$	1.120	1.121	1.128	1.120	1.120	1.116	1.113	1.122	1.122			
$\frac{V_r}{V_s}$	#1—18 × 20	1.13	1.158	1.176	1.096	1.116	1.134	1.09	1.099	1.112			
	#2—10 × 10	1.163	1.179	1.196	1.115	1.139	1.150	1.107	1.118	1.128	1.100	1.107	1.135
	Ratio $\frac{\#1}{\#2}$	0.97	0.983	0.983	0.983	0.98	0.986	0.983	0.982	0.985			
	#3—$6\frac{1}{2} \times 6\frac{1}{2}$	1.270	1.284	1.310	1.219	1.242	1.253	1.206	1.226	1.239	1.208	1.227	1.243
	Ratio $\frac{\#2}{\#3}$	0.915	0.918	0.913	0.915	0.918	0.918	0.918	0.913	0.911	0.911	0.903	0.914
	Ratio $\frac{\#1}{3}$	0.889	0.901	0.898	0.899	0.898	0.906	0.903	0.896	0.897			

proved out in practice. Complete test results on horizontal double-acting compound compressors have been published.

Dual or Multiple-Effect Compression

8. In the dual-effect machine, gas at two suction pressures is handled in the same cylinder. The gas of lower pressure is drawn in on the regular suction stroke, and at the end of the stroke a port is uncovered through which the high-pressure gas is admitted. The theoretical indicator card for such a machine is shown in **Fig. 32** by the solid line. V_G is the volume of low-pressure gas drawn in, $V_{G'}$ the volume of high-pressure gas, and V_X is the displacement volume lost due to uncovering the port at the end of the stroke, and having to recover it before compression starts.

Mathematical analyses of the multiple-effect compressor have been made. The multiple-effect principle should not be confused with compound compression, as it lacks the possibility of intercooling, with its attendant economies. It is more nearly the equivalent of two compressors, one at low suction pressure and the other at high suction pressure, although slightly less efficient. For any given condition of suction pressures, the capacity of the compressor is fixed, and the design of machine cannot be changed to give some different ratio between the two capacities. However, there are certain applications where the high suction pressure can balance out, as for instance in an ice plant. The high suction pressure serves to cool liquid ammonia and precool water for ice-making.

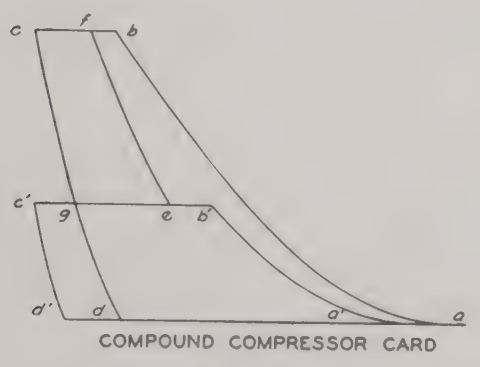


Fig. 29

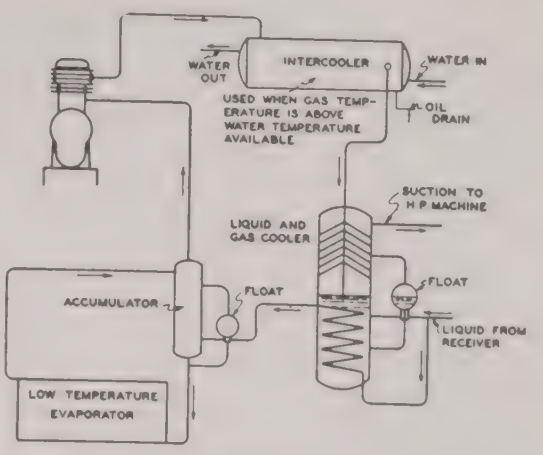


Fig. 30. Schematic Diagram of Compound Compressor

Many ice plants have been equipped with dual-effect machines with considerable reduction in power costs.

If a dual-effect compressor is provided with clearance pockets for capacity control, opening the pockets will reduce the low-pressure capacity and increase the high.

Data comparing a dual-effect CO_2 compressor, cooling the liquid at high suction pressure, with single-stage operation, appear in **Table 9.**

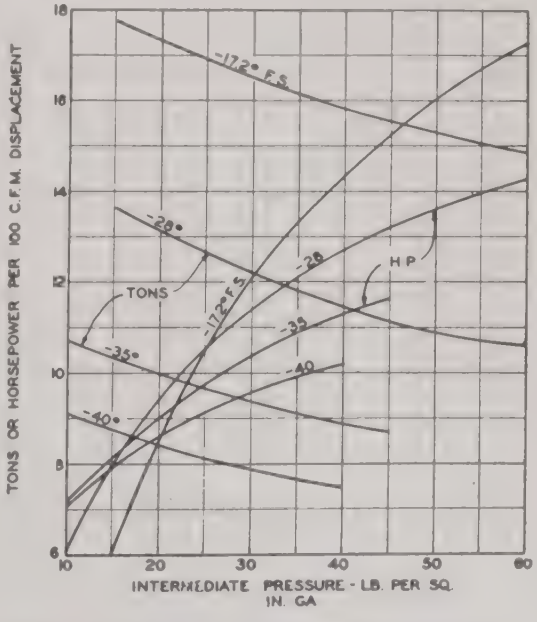


Fig. 31. Horsepower and Capacity Curves for V.S.A. Low-Stage Booster Compressors

[1] Saturated suction hp per 100 cu ft.
[2] Saturated suction tons per 100 cu ft.

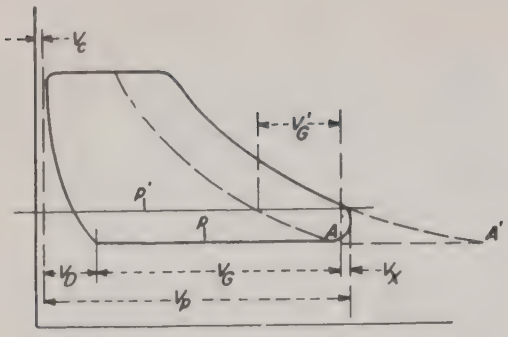


Fig. 32. Theoretical Indicator Diagram

Carbon dioxide compressors are no longer commonly in use for refrigeration but they are used in other departments of refrigerating engineering, especially the manufacture of dry ice. High pressures and low critical conditions are encountered.

Capacity Control

9. Table 10 shows test results on a 6-cylinder, 25-hp Freon-12 compressor, having suction valves on one or more cylinders held open for capacity control. A comparison of percentage capacity with percentage motor input shows a slight reduction in horsepower per ton, with capacity reduction due to the evaporator temperature increasing and the discharge pressure decreasing, with added steps of capacity reduction.

There may be a considerable difference in the losses in various designs of capacity controls, depending upon gas velocities, viscosity and density of the gas, but a capacity control does always reduce the adiabatic efficiency. There usually is, however, an actual saving in power as compared to operating at full capacity on a greatly reduced load. Furthermore, for proper regulation, some adjustment of machine capacity is essential in certain applications. A change of compressor speed, or providing a number of compressors and operating as many as are needed for the load, would be a more efficient means, but far more expensive to install.

Some of the various capacity control means listed in Chapter 23, while feasible, are not current practice. Adjustable clearance, one of the more popular devices, can be finely adjusted for any capacity within its range, whereas other methods are operative in rather wide steps. The actual reduction with any particular clearance setting varies with the compression ratio.

Performance Data

10. Tables 11 and 12 (see pages 36 and 37) give performance figures for ammonia compressors and **Table 13** (see page 37) shows the corresponding volumetric and adiabatic efficiencies. The above tables represent a consensus of compressor manufacturers' values rather than actual tests.

Exhaustive data on an ammonia com-

Table 9. Test Results of CO_2 System with Dual Effect

Temp of evaporation, F	Temp of liquid to exp. valve, F	Pressure of evaporation, psig	Pressure of condensation, psig	Ratio of compression	Temp of inter. receiver, F	Approx press of intermediate receiver, psig	Ratio of compression, hp to int.	Single compr'n theo. displ't saturated, cu in./min./ton	Actual single compr. displ't, cu in./ton, working cond'n	Dual compr'n displacement, cu in./min./ton, working cond'n	Theo. adiabatic hp/ton single compr'n saturated	Actual bhp/ton aver. work. condition single compr'n	Actual bhp/ton dual compr'n	Theo. adiabatic disch temp F 20° superheat single compr'n	Actual disch temp F 20° superheat single compr'n	Actual disch temp F 20° superheat dual compr'n	Coefficient of performance single compr'n theo. adiabatic
−20	85	206	1,130	5.5	19	399	2.83	228	2,960	1,940	2.68	3.2	2.94	213	218	190	1.76
−10	85	247	1,125	4.55	26	445	2.53	1,890	2,440	1,620	2.28	2.75	2.53	206	203	176	2.06
± 0	85	294	1,125	3.83	30	475	2.37	1,570	2,010	1,380	1.88	2.35	2.12	192	192	165	2.50
5	85	320	1,125	3.52	33.5	500	2.26	1,440	1,840	1,270	1.71	2.16	1.96	187	186	161	2.78
10	85	347	1,120	3.24	39.0	540	2.07	1,315	1,685	1,170	1.54	1.97	1.79	181	177	158	3.08
15	85	376	1,120	2.98	41.5	564	1.98	1,215	1,555	1,080	1.41	1.79	1.63	175	171	155	3.36
20	85	407	1,120	2.75	43.5	580	1.93	1,120	1,440	1,010	1.28	1.61	1.48	170	165	151	3.70
25	85	440	1,120	2.54	47.0	610	1.84	1,030	1,310	945	1.16	1.46	1.34	165	160	148	4.06
30	85	475	1,120	2.36	49.5	630	1.78	953	1,230	880	1.06	1.30	1.20	158	156	146	4.48

Table 10. Tests on Freon-12 Compressor, 6-cyl Unit under Varying Load

Number of cylinders operating	Discharge pressure, psi	Evap. temp, F	Capacity, 1,000 Btu/hr	Motor input, kw	Capacity, %	Motor, input, %
6	119	38.0	277	22.1	100	100.0
5	117	41.3	248	19.1	89.5	86.5
4	112	44.5	216	16.1	78.2	73.0
3	107	48.7	117	13.3	63.9	60.0
2	102	53.0	119	10.2	43.0	46.0

pressor appear in **Tables 14, 15, 16** (see page 38). The compressor was a 4.5-in. ×4.5-in. single-cylinder machine running at from 444 to 537 rpm, with feather valves, operated by means of a 10 to 25-hp dynamometer-motor, direct-connected through flexible coupling to compressor shaft. The speed was controlled by resistance grids, permitting the brake and indicated horsepower to be measured. Actual capacity was measured by weighing ammonia from a multitube multipass condenser having two shells, each with seven tubes, with a condensing surface of 93.8 sq ft. The evaporator operated on brine, heated in turn by steam. Suction gas temperature at the machine was held at a few degrees of superheat. Liquid at the expansion valve had the same temperature as the liquid leaving the condenser.

Comparison of the actual compression cycle with the theoretical can be made by reference to Table 16, which appears in connection with studies of volumetric efficiency of the same compressor. The author of these tests commented that the power requirements of this compressor seemed high, but made no statement as to whether they were typical of this whole class of machines.

There is only one set of test data on volumetric efficiency in print, this one based on measurement of ammonia flowing from which was determined the total volumetric efficiency under various operating pressures. Knowing the amount of clearance in the existing compressor, the investigators computed volumetric efficiency due to clearance. They then divided the "total volumetric efficiency" by the "clearance volumetric efficiency" and obtained what was termed the real volumetric efficiency, taking account of the loss due to superheating the gas as it entered the cylinder, and also whatever losses may have existed in leakage. It was then shown that real volumetric efficiency, as well as efficiency due to clearance, is a function of the compression ratio alone. **Fig. 33** shows real volumetric efficiency plotted against compression ratio. The lowest curve is an exact reproduction of the curve of real volumetric efficiency in the tests. Confirming test results falling close to this curve were later obtained on a 3-in. × 3-in. compressor and a $3\frac{1}{2}$-in. × $3\frac{1}{2}$-in. compressor.

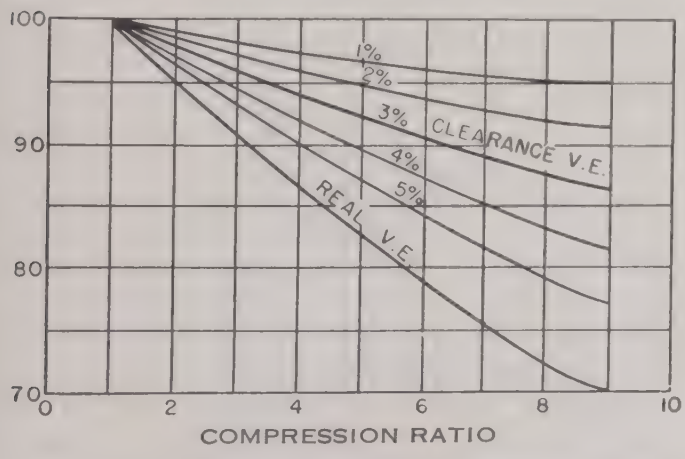


Fig. 33 Real Volumetric Efficiency vs. Compression Ratio

The engineer strives to develop more and more efficient apparatus, but high efficiencies are not an end in themselves. Reliability and low upkeep cost can outweigh a few points of efficiency, and other factors such as quietness, freedom from vibration, reduced size and weight, may be of primary importance.

Table 11. Ammonia Compressor Performance* at Various Operating Conditions for Vertical Single-Acting Ammonia Compressors and Single-Stage Compression (Air Conditioning and Refrigeration Machinery Association)

Pressure and corresponding saturated temperature of suction gas		Discharge pressure, psig															
		135		145		155		165		175		185		195		205	
		Corresponding saturated temperature, F															
		78.7		82.5		86.2		89.7		93.0		96.2		99.3		102.3	
Pressure, psig	Temperature, F	cfm / ton	bhp / ton	cfm / ton	bhp / ton	cfm / ton	bhp / ton	cfm / ton	bhp / ton	cfm / ton	bhp / ton	cfm / ton	bhp / ton	cfm / ton	bhp / ton	cfm / ton	bhp / ton
0	−28.0	12.34	2.51														
5	−17.2	8.42	2.02	8.64	2.12	8.89	2.22	9.16	2.32	9.40	2.41	9.66	2.50				
10	− 8.4	6.36	1.694	6.53	1.782	6.70	1.869	6.84	1.955	7.01	2.04	7.16	2.12	7.35	2.20	7.53	2.28
15	− 1.0	5.09	1.449	5.21	1.531	5.34	1.611	5.45	1.690	5.56	1.765	5.67	1.839	5.79	1.911	5.93	1.980
20	5.5	4.28	1.270	4.39	1.341	4.46	1.412	4.55	1.482	4.64	1.552	4.72	1.621	4.80	1.688	4.90	1.755
25	11.3	3.69	1.127	3.75	1.193	3.84	1.259	3.91	1.324	3.98	1.387	4.06	1.449	4.12	1.509	4.20	1.568
30	16.6	3.24	1.011	3.31	1.073	3.38	1.133	3.44	1.193	3.50	1.251	3.56	1.307	3.63	1.363	3.69	1.419
35	21.4	2.92	0.910	2.96	0.971	3.00	1.029	3.06	1.085	3.11	1.140	3.16	1.193	3.22	1.244	3.27	1.293
40	25.8	2.62	0.825	2.66	0.883	2.71	0.938	2.75	0.992	2.81	1.044	2.85	1.094	2.90	1.144	2.94	1.193
45	30.0	2.39	0.755	2.43	0.807	2.47	0.859	2.50	0.910	2.54	0.961	2.58	1.009	2.62	1.057	2.65	1.103
50	33.8	2.18	0.689	2.21	0.739	2.25	0.789	2.28	0.838	2.33	0.886	2.36	0.934	2.39	0.980	2.24	1.026

* Based on two-cylinder compressors with 6-in. bore and 6-in. stroke. For other compressor sizes, multiply by machine factors of Table 12.
The values above the heavy line are for ratios of compression above 8.0, and for these conditions it is recommended that multi-stage compression be considered.

$\frac{cfm}{ton}$ =average values of swept volume, in cu ft per min, required per ton refrigeration based on a suction superheat of 10 F and on liquid ammonia entering the refrigerant control valve at the saturated temperature corresponding to the discharge pressure.

$\frac{bhp}{ton}$ =average values of actual compressor brake horsepower per ton refrigeration.

Table 12. Machine Factors for Various Compressor Sizes (Air Conditioning and Refrigeration Machinery Association)

	Compressor bore and stroke, in.															
	2½	3	3½	4	4½	5	5½	6	6½	7	7½	8	8½	9	9½	10	10½
cfm ton	1.034	1.030	1.026	1.022	1.017	1.012	1.006	1.000	0.993	0.986	0.979	0.972	0.966	0.960	0.954	0.948	0.943
bhp ton	1.132	1.106	1.083	1.063	1.044	1.027	1.013	1.000	0.988	0.977	0.968	0.960	0.952	0.944	0.937	0.930	0.924

Table 13. Volumetric Efficiency and Adiabatic Efficiency of Vertical Single-Acting Single-Stage Ammonia Compressors from Table 11

Pressure and corresponding saturated temperatures of suction gas		Discharge pressure, psig															
		135		145		155		165		175		185		195		205	
		Corresponding saturated temperature, F															
		78.7		82.5		86.2		89.7		93.0		96.2		99.3		102.3	
Pressure, psig	Temp, F	Vol. eff	Adiab. eff	Vol. eff	Adiab. eff	Vol. eff	Adiab. eff	Vol. eff	Adiab. eff	Vol. eff	Adiab. eff	Vol. eff	Adiab. eff	Vol. eff	Adiab. eff	Vol. eff	Adiab. eff
0	−28.0	62.5	61.0														
5	−17.2	69.1	63.0	68.1	64.2	66.8	64.0	65.4	64.2	64.3	64.0	63.0	64.0				
10	− 8.4	73.6	67.0	72.5	67.0	71.1	67.2	70.4	66.9	69.2	67.0	68.2	67.0	67.1	67.0	65.9	67.0
15	− 1.0	75.8	68.8	75.2	68.8	74.4	69.5	73.5	69.0	72.7	69.0	72.0	69.0	71.2	69.0	69.8	69.0
20	5.5	78.6	70.0	77.2	70.5	76.6	70.5	75.7	70.5	74.9	70.7	74.3	70.7	73.5	71.0	72.6	71.0
25	11.3	80.1	70.7	79.7	71.5	78.6	71.6	77.7	71.7	76.8	72.1	75.7	72.5	75.2	72.3	74.5	72.5
30	16.6	81.2	70.8	80.0	71.4	79.3	72.0	78.5	72.6	77.8	72.9	77.2	73.3	76.3	73.3	75.6	73.8
35	21.4	81.6	72.1	81.3	72.7	80.7	73.0	79.7	73.7	79.1	74.3	78.4	74.5	77.6	74.8	77.3	75.3
40	25.8	82.3	71.7	82.0	72.5	81.3	73.2	80.6	73.8	79.7	74.3	79.3	75.2	78.7	75.2	78.2	75.6
45	30.0	83.2	70.4	82.6	73.0	81.8	72.2	81.5	73.9	81.1	74.5	80.3	75.3	79.4	75.2	79.2	75.5
50	33.8	84.6	70.8	84.2	72.2	83.3	73.2	82.9	74.1	81.8	74.6	81.3	75.3	81.0	75.8	80.6	76.5

Table 14. Capacity of Commercial Ammonia Compressor as a Function of Pressure

(Tabular values are tons refrigerating capacity of machine)

Dis. press, psia	Suction pressure, psia			
	50	40	30	20
110	7.6	5.9	4.3	2.6
130	6.9	5.6	4.1	2.4
150	6.2	5.4	3.9	2.3
170	5.4	4.9	3.7	2.1
190		4.8	3.4	1.9
210			3.2	1.8

Table 15. Horsepower per Ton—for $4\frac{1}{2}\times4\frac{1}{2}$ s.a., single-cyl Ammonia Compressor at 500 rpm

Dis. press, psia	Cond. temp, F	Suction pressure, psia											
		50 (21.67 F)			40 (11.66 F)			30 (−0.57 F)			20 (−16.64 F)		
		1	2	3	1	2	3	1	2	3	1	2	3
100	56.05	0.74	0.38	0.54	0.96	0.50	0.72	1.23	0.66	0.95	1.77	0.95	1.45
120	66.02	0.86	0.49	0.66	1.11	0.63	0.85	1.41	0.81	1.12	2.03	1.11	1.62
140	74.29	0.98	0.60	0.78	1.26	0.74	0.97	1.60	0.94	1.27	2.29	1.25	1.80
160	82.64	1.11	0.70	0.90	1.42	0.85	1.09	1.78	1.05	1.41	2.56	1.37	1.99
180	89.78	1.23	0.81	1.02	1.56	0.94	1.20	1.96	1.15	1.54	2.82	1.50	2.14
200	96.34	1.36	0.91	1.13	1.72	1.01	1.30	2.14	1.25	1.65	3.08	1.60	2.31
220	104.42	1.48	1.02	1.25	1.89	1.08	1.39	2.32	1.34	1.75	3.35	1.69	2.48

1. Brake hp. 2. Indicated hp. 3. Indicated+friction hp.

Table 16. Volumetric Efficiency of Commercial Ammonia Compressor

(Log of tests, varying pressure of suction gas; same machine as in Tables 14, 15.)

Rpm	540	540	532	530	519
Suc. gas, temp, F	51.0	47.0	55.0	56.5	75.2
Dis. gas, temp, F	277	244	269	190	263
Dis. gas, temp F theor.	282	223	246	170	240
Suc. press, psia	24.34	34.71	44.07	54.71	63.28
Dis. press, psia	133.1	134.6	186.0	137.4	215.2
Weight NH_3 circ., lb/min	1.233	1.977	2.386	3.445	3.395
Total real vol. eff	67.6	76.8	74.7	86.7	78.3
Real vol. eff without clearance	80.0	85.8	84.6	92.2	86.4

3. ABSORPTION REFRIGERATING SYSTEMS

ABSORPTION refrigerating machines are **heat-operated,** and make use of at least two working substances, a **refrigerant** and an **absorbent** for the refrigerant. **Liquid-absorbent** refrigerating systems operate in accordance with the laws and properties of solutions, particularly with respect to exchange of materials between the liquid and vapor phases. A simple continuous cycle for a liquid absorbent is shown in **Fig. 1.**

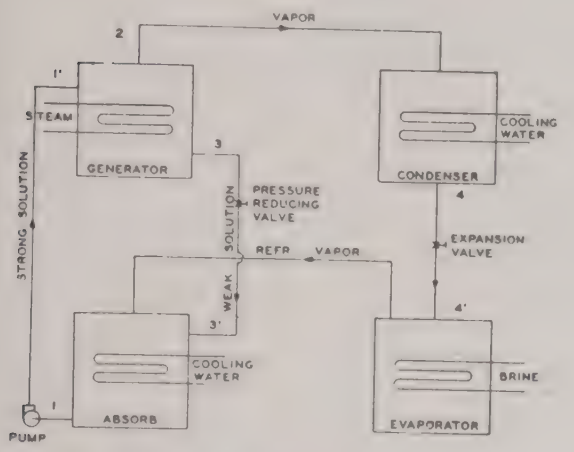


Fig. 1. Simple Absorption Cycle

I. Laws and Properties of Solutions

1. If two or more substances mix uniformly over a range of compositions, to produce only one physically distinct **phase,** such a mixture is called a **solution,** as distinguished from a mechanical mixture of two or more phases. Here, the term "solution" will usually refer to a liquid phase. In describing solutions and gaseous mixtures, the **compositions** may be specified, for instance by giving either the **weight fractions** or the **mol fractions,** or the equivalent percentages, of the pure constituents or components. Or the **concentrations** of the components per unit volume may be specified in any of numerous ways.

For a two-component, refrigerant-absorbent system:

Weight fraction:

$$x_1 = \frac{G_1}{G_1+G_2};\quad x_2 = \frac{G_2}{G_1+G_2};\quad x_1+x_2 = 1$$

Mol fraction:

$$N_1 = \frac{G_1/m_1}{G_1/m_1+G_2/m_2};\quad N_2 = \frac{G_2/m_2}{G_2/m_1+G_2/m_2};$$
$$N_1+N_2 = 1$$

where G_1 = weight of refrigerant in mixture, lb
m_1 = molecular weight of refrigerant
G_2 = weight of absorbent in mixture, lb
m_2 = molecular weight of absorbent

Where no confusion will result, x and N may be used without subscript to refer to the refrigerant. **A weak** solution is one which contains relatively little refrigerant, as compared with a **rich** or **strong** solution.

2. The gas phase. The words "gas" and "vapor" will be used more or less interchangeably, "vapor" being chosen when it is desired to emphasize a relation to a liquid phase. For a pure gas, the P-V-T (pressure-volume-temperature) relations can be obtained from the perfect gas law, or more accurately from an **equation of state,** or a table such as those in Chapter 7, representing actual measurements. If the solution contains more than one volatile component (absorbent as well as refrigerant volatile), or if evaporation is into a space which contains a foreign gas, **Dalton's Law** applies more or less accurately to the resulting gaseous mixture. Each component of a gaseous mixture exerts a partial pressure equal to the total pressure which it would exert if it alone were present, and the total pressure P is the sum of the partial pressures, P_1, P_2, etc., of the individual components.

Dalton's law is a corollary of the perfect gas law. For any perfect gas:

$$PV = nRT \quad \text{or} \quad P = \frac{nRT}{V}$$

Then for any component of a mixture of perfect gases:

$$P_1 = \frac{n_1 RT}{V};\quad P_2 = \frac{n_2 RT}{V};\ \text{etc.}$$

and for the mixture:

$$P = (n_1 + n_2 + \cdots)\frac{RT}{V} = P_1 + P_2 + \cdots$$

Each n gives the number of mols of the gas indicated by the subscript, and R, the "gas constant," is the same for all perfect gases. $R = 10.71$ if P, the absolute pressure, is in psi, V is in cu ft, and T is in absolute F. Dalton's law may also be expressed in the form:

$$P_1 = N_1 P;\quad P_2 = N_2 P;\ \text{etc.}$$

These relations follow from the above if one mol of mixture is taken, for which $P = RT/V$ and $n_1 = N_1$, $n_2 = N_2$, etc. This is the form in which Dalton's law is most commonly applied, but it is less general than the original statement, which permits the calculation of the partial pressures from actual P-V-T data on the individual components. If these are available and the concentrations are known, this form gives better accuracy than does the perfect gas law.

Dalton's law is the law of perfect gaseous mixtures. Mixtures of actual gases follow it fairly well, though not so well as the individual gases follow the perfect gas law, at least if the $P_1 = N_1 P$ form is used. Accordingly, Dalton's law is useful, although not quantitatively accurate, up to fairly high pressures; how high depends on the particular mixture. It always breaks down as the partial pressure of any component and the temperature simultaneously approach the critical values for that component.

3. The **vapor pressure** of a liquid is the pressure at which vapor from the liquid is in equilibrium with the liquid. Applying Dalton's law, each component of a solution is said to have a definite **partial vapor pressure,** which is equal to its partial pressure in the equilibrium vapor. All of the partial vapor pressures, and therefore also the (total) vapor pressure and the equilibrium composition and concentration of the vapor, are physically determined by the temperature and composition of the solution; the quantitative relations depend on the particular solution. For a given composition of solution, the partial and total vapor pressures increase rapidly with temperature, following about the same rules as do the vapor pressures of pure liquids. For a given temperature, the partial vapor pressure of a solution is always less than the vapor pressure of the pure component, increasing with its concentration in solution. The equilibrium composi-

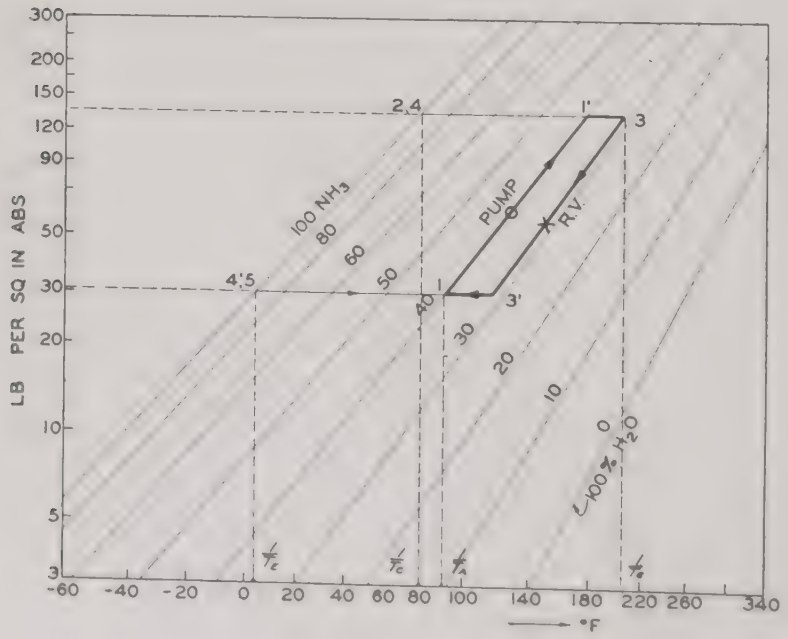


Fig. 2. Log p vs $1/T$ Chart for Ammonia Water

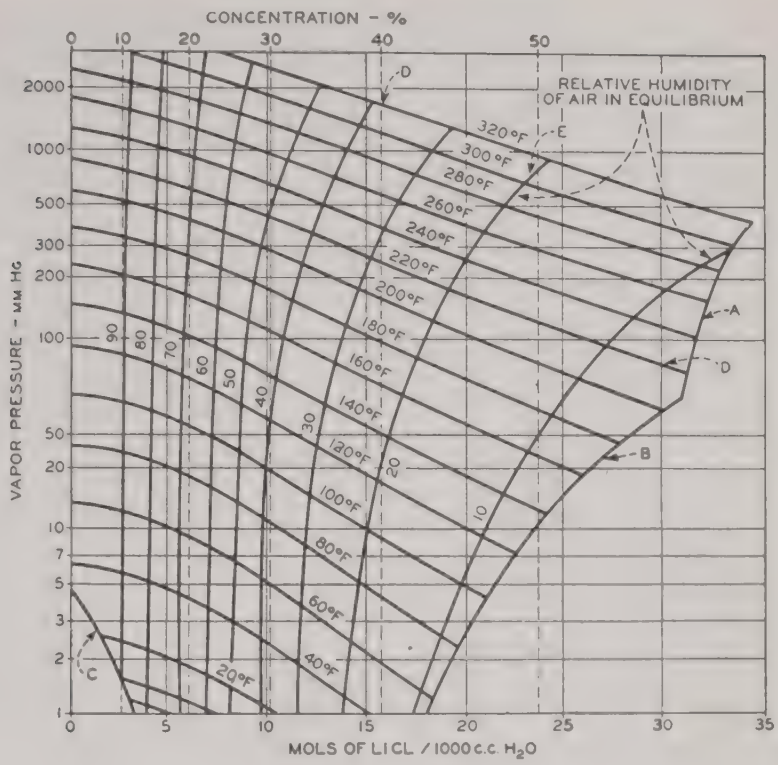


Fig. 3. Water-Lithium Chloride Equilibria

tion of the vapor is in general different from that of the liquid, and usually is richer in the more volatile component.

Quantitatively, the relations between a solution and its vapor are complicated and must be determined experimentally in each case (for example, **Figs. 2, 3** and **4**), but there are two simple limiting laws which apply at the ends of the concentration range. **Henry's Law** applies to the minor component, or solute: The partial vapor pressure of a component whose mol fraction in the solution is sufficiently low, is proportional to the mol fraction:

$$P_1 = k_1 N_1$$

The value of the Henry's law constant, k_1, and the range of mol fractions over which it applies, depend on the particular solution and the temperature, and must be found by experiment. **Raoult's Law** applies to the major component, or solvent: The partial vapor pressure of a component whose mol fraction in solution is sufficiently near unity equals the product of the mol fraction and the vapor pressure of the pure solvent:

$$P_2 = N_2 P_2^0$$

Raoult's law may be considered to be a special case of Henry's law, in which the Henry's law constant is the vapor pressure of the pure component, P_2^0. For a given component, Henry's law applies at one end of the composition range, and Raoult's law at the other. There are no general rules for predicting the range of compositions over which either applies, nor what happens in between.

Perfect solutions, those for which Raoult's law applies to all components over the entire range of compositions, result when the molecular forces around the solvent and solute molecules resemble each other so closely that there are no special attractions or repulsions between solvent and solute molecules. For example, any pair of fairly closely related hydrocarbons may be depended on to follow Raoult's law quite closely, over the whole range of compositions and temperatures. If the components are much different chemically, their solutions are likely to deviate more or less from Raoult's law over most of the composition range. If the partial vapor pressure of a component exceeds the Raoult value at a given mol fraction, the deviation

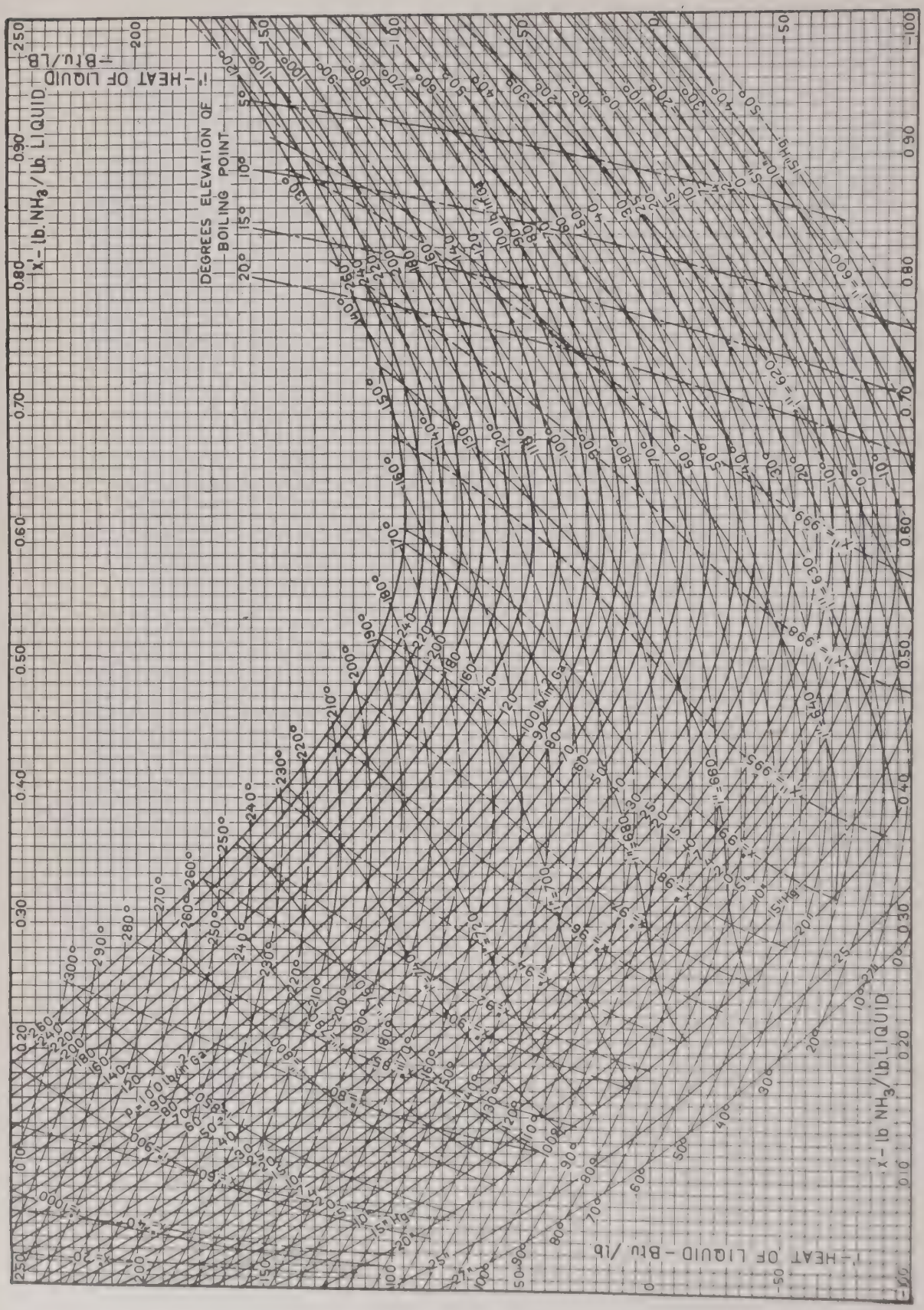


Fig. 4. Enthalpy-Concentration Chart of Properties of Aqua Ammonia

is said to be positive, and the solubility is less than it would be if the solution were perfect. This is always the case when two liquids do not mix completely, and often when they do; it shows that the solvent and solute molecules repel each other.

If the partial vapor pressure of a component is less than the Raoult value, the deviation is negative, and the component is more soluble than it would be if the solution were perfect. This situation arises only in cases where there is special attraction between solute and solvent molecules. Usually, both components of a two-component system deviate from Raoult's law in the same direction, the deviations are in the same direction for all concentrations and temperatures, and the deviations become smaller percentagewise at higher temperatures; but there are occasional exceptions.

4. It is essential for the refrigerant to be highly **soluble** in the **absorbent,** at the temperature and partial pressure at which the absorbent is required to take up the refrigerant, in order to keep down the volume of absorbent required to be circulated, the size of the apparatus, and the heat losses. Large negative deviations from Raoult's law are therefore required; and the larger and more complicated the absorbent molecule, the larger the deviations which are required. On the other hand, the negative deviations must not be so great as to prevent recovery of a large proportion of the dissolved refrigerant by distillation, and must not result in the formation of an insoluble compound between the absorbent and refrigerant. It is not practical to set up definite rules, but an idea of the requirements may be obtained by comparison with the aqua-ammonia system, which is the one most commonly used in absorption refrigeration. In the relatively small-molecule H_2O-NH_3 system, the solubility of ammonia under absorber conditions is typically around four times the Raoult value, and is none too large at that. Very few other combinations of solvent and solute give as large negative deviations without forming insoluble compounds.

The requirements for a successful absorbent-refrigerant combination are extremely severe in numerous respects. The solubility requirements are hardest to meet. Another important consideration is that the absorbent must have a much higher boiling point than the refrigerant, so that practically pure liquid refrigerant can readily be obtained from the solution by fractional distillation. In this respect, the water-ammonia system is far from ideal, and few others are better. Also, irreversible chemical reactions of all kinds, such as decomposition, polymerization, and corrosion, are to be avoided. It is not surprising that extremely few acceptable refrigerant-absorbent combinations have been found.

5. The pure absorbent may be a solid; it is only the absorbent solutions which are required to be liquid. For example, a number of salts such as LiCl and LiBr may be used as absorbents with water as refrigerant, because they are extremely soluble in water and form **hygroscopic brines.** Such salts have the great advantages of being small-molecule, low-specific-heat, completely non-volatile absorbents, with very large negative deviations from Raoult's law within their operating ranges; but the operating ranges are limited by the appearance of solids. This limits the reduction in temperature attainable in the evaporator; and, in any case, a water evaporator must be operated above the freezing point of water. Hygroscopic brines are used to dehumidify air or other gases by direct contact, and also in absorption refrigeration. Properties of LiCl brines are shown in **Tables 1** and **2** and Fig. 3.

Table 1. Specific Heat of LiCl-Water Solution

Concentration, % of LiCl by weight	Specific heat, Btu/lb F
.5	.993
5.0	.960
10.0	.901
20.0	.808
30.0	.730
40.0	.660

6. Preparation of a usable **table** or **graph** from the usually quite limited amount of original data available on the liquid-vapor equilibrium of a given absorbent-refrigerant combination requires special procedures for correlation and interpolation. Several such procedures are familiar to

chemical engineers. For fixed liquid composition and varying temperatures, the logarithms of the partial and total vapor pressures may be plotted against the reciprocal of the absolute temperature; or "Dühring's rule"[1] or Othmer[2] plots may be made. All of these give straight or nearly straight lines which relate the total or partial vapor pressure to the temperature. A skeleton log P-$1/T$ chart is shown for ammonia-water in Fig. 2. For fixed temperature and varying concentration, "Raoult's law" plots, for example, of P, P_1, P_2, $P_1/P_1^0\ N_1$ and $P_2/P_2{}^0\ N_2$ against N_1, can be recommended. The form of the final table or graph is largely a matter of taste; it can well be of some other type.

Thermal data, i.e., total and latent heats, are needed in addition to equilibrium pressure-temperature-composition data, and should be correlated thermodynamically with the latter. Entropies are desirable also. Procedures will not be discussed here, but certain qualitative relations may be pointed out. As a general rule, the latent heats of vaporization of both refrigerant and absorbent are somewhat higher from solution than they are from the pure liquids, because of the large negative deviations from Raoult's law. The increase in latent heat is part of the price of better absorption. It then follows from thermodynamics that heat is evolved on mixing pure liquid refrigerant with pure liquid absorbent or with weak solutions of refrigerant in absorbent; and also that the partial and total vapor pressures of solutions increase more rapidly with temperature (i.e., the slope of log P-$1/T$ plots is greater) than do the vapor pressures of the pure components. A large heat of mixing of liquids (i.e., a large temperature rise) is an excellent indication of large negative deviations from Raoult's law and high solubility.

Charts giving the equilibrium pressure-temperature-concentration relations are shown in Fig. 2 for the ammonia-water system, and in Fig. 3 for the water-lithium chloride system. Fig. 4 shows the equilibrium pressure-temperature-concentration relations and the corresponding total heats of liquid and vapor, for the ammonia-water system. Fig. 4, prepared by Stickney,[3] is of a type suggested by Merkel and Bosnjakovic,[4] in which the coordinates are weight-concentration and enthalpy. A similar chart, for the two phases separately, was prepared by Berestneff.[5] Stickney's Fig. 4 superimposes the data for the liquid phase (index ′) and for the vapor phase (index ″). The use of Fig. 4 is illustrated by **Fig. 5,** which shows a complete set of readings for one point on Fig. 4. Fig. 4 gives no information concerning superheated vapor. The enthalpies (total heats) for liquid in equilibrium with vapor, also apply to subcooled liquid: that is, to liquid of the same composition and temperature, but under pressure in excess of the vapor pressure. The most recent aqua-ammonia tables are those of Scatchard et al in *Refrigerating Engineering* of May 1947.

Table 2. Specific Gravity of LiCl-Water Solution
(Concentration is given in percentage of LiCl in mixture by weight)

% LiCl	Temperature, C								
	0	10	20	30	40	50	60	80	100
1	1.00604	1.00574	1.00411	1.00150	.99806	.99391	.9891	.9779	.9646
10	1.05966	1.05833	1.05594	1.05296	1.04954	1.04571	1.0414	1.0315	1.0199
20	1.11979	1.11789	1.11504	1.11181	1.08550	1.10500	1.1013	1.0927	1.0829
30	1.18464	1.18219	1.17914	1.17602	1.17269	1.16940	1.1659	1.1576	1.1507
40				1.24806	1.24427	1.24081			
45				1.28768	1.28361	1.28003			

II. Principles of Absorption Refrigeration

7. The **basic cycle** of liquid-absorbent refrigerating systems is indicated in Fig. 1. The four major steps are latent-heat processes involving the transfer of refrigerant between the liquid and vapor states, in the

evaporator, absorber, generator and condenser, respectively. Nearly pure liquid refrigerant boils in the evaporator, absorbing heat and producing refrigeration at a low temperature. The vapor passes to the absorber, where it is absorbed by weak solution, producing rich solution and evolving

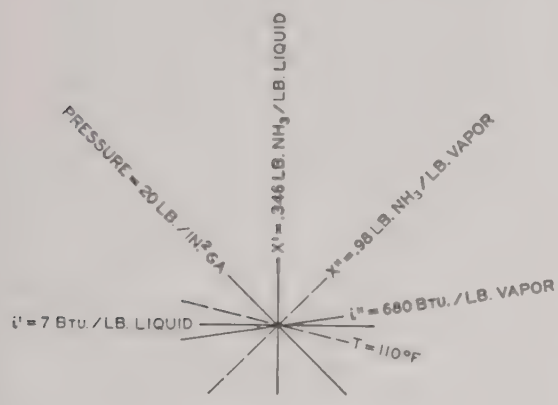


Fig. 5. Illustrating Use of Fig. 4

waste heat which is rejected to the cooling medium at ordinary temperature. The evaporator and absorber operate at the low-side pressure, which is determined by the vapor pressure of the refrigerant at evaporator temperature. Rich solution is pumped to the generator, where high-temperature heat is supplied, to boil out refrigerant vapor. The vapor then passes to the condenser, where it condenses and evolves waste heat which is rejected to the cooling medium at ordinary temperature. The generator and condenser operate at the high-side pressure, which is determined by the vapor pressure of the refrigerant at the condensing temperature. The weak solution produced in the generator is fed to the absorber through a valve, and the liquid refrigerant produced in the condenser is fed to the evaporator through another valve.

8. Consider the unrealizable ideal case for which the absorbent is non-volatile, the specific heats of both absorbent and refrigerant are zero, the four processes listed above are the only ones of thermodynamic significance, and these are carried out under conditions of perfect reversibility, with rich and weak solutions of the same concentration, namely, that for which the partial vapor pressure of refrigerant at absorber conditions equals the vapor pressure of pure refrigerant at evaporator temperature. The refrigerant vapor would obey the perfect gas law, and the latent heats would be independent of temperature at constant concentration, and for the ideal combination, as for any other, would presumably be slightly higher from the solution than from the pure refrigerant. Defining the cycle efficiency or performance factor PF as the ratio of the heat absorbed in the evaporator to the heat absorbed in the generator, the PF of the ideal cycle would be slightly less than one under all circumstances, decreasing slightly as the specified difference between evaporator and absorber increased. A simple thermodynamic expression for the ideal PF can readily be obtained. Identify the following quantities:

	Absolute temperature	*Heat absorbed or evolved*
Evaporator	T_1	Q_1
Absorber and condenser	T_2	Q_2
Generator	T_3	Q_3

Let w equal the theoretical work, for the ideal reversible cycle, which is derived from Q_3 supplied at T_3, and is used to remove Q_1 at T_1, the heat-sink being at T_2 for both. Then by the first law of thermodynamics:

$$Q_2 = Q_1 + Q_3$$

and by the second law:

$$w = Q_1 \frac{T_2 - T_1}{T_1} \text{ and also } w = Q_3 \frac{T_3 - T_2}{T_3}$$

Then

$$\frac{Q_1}{Q_3} = PF = \frac{T_3 - T_2}{T_2 - T_1} \cdot \frac{T_1}{T_3}$$

The latter expression is not to be regarded as giving theoretical values of PF which vary with the arbitrary choice of T_1, T_2, and T_3, because, as mentioned above, Q_1/Q_3 is fixed by other considerations and has a value of about 1; say 0.9. What it really means is that

$$(T_3 - T_2) = 0.9 \frac{T_3}{T_1} (T_2 - T_1)$$

is approximately correct for the ideal relation between the three temperatures. The ideal $T_3 - T_2$ is thus of the same order as $T_2 - T_1$, usually somewhat smaller. Increasing $T_3 - T_2$ above the ideal value for a given $T_2 - T_1$ would not improve the theoretical *PF*, but it would provide an excess of available work over that ideally required, for use in overcoming irreversibility in the cycle. For example, it could be used to produce weaker solution and thus to provide driving head for the absorber process.

9. Actual cycles are never so good as the ideal cycle described above; whatever differences there are, are always of a sort which produce thermodynamic loss. Some of the processes waste high temperature heat, some waste refrigeration, and others waste available work in some other form. Both T_3 and Q_3 must be increased above the ideal values to compensate for these losses. There is considerable room for choice between increases in T_3 and Q_3; usually the optimum value of T_3 may be considered to be the one for which *PF* is a maximum.

The *PF* of actual absorption cycles has an upper limit of one, no matter how efficiently they are carried out nor how small the temperature differences between the evaporator and the condenser and absorber may be. On the other hand, the *PF* falls off only slowly as these temperature differences are increased, most of the additional available work required being obtainable through increases in T_3 instead of Q_3. These characteristics are very different from those of compression systems, for which the energy input per unit of refrigeration goes up in somewhat the same proportion as the temperature differences between the evaporator and the condenser, and the thermodynamic efficiency, instead of the performance factor, remains relatively constant.

10. Analysis of simple cycle. Design, operating characteristics and performance can be analyzed by setting up appropriate material and heat balances. For any chosen section of an absorption refrigerating system in a steady state, intake must equal outgo, for refrigerant, for absorbent and for the sum of the enthalpy of absorbent plus refrigerant and heat from external sources. Giving intake and outgo opposite signs, these conditions can be represented by the equations:

$$\Sigma W = 0; \quad \Sigma Wx = 0; \quad \Sigma(Wi + Q) = 0$$

where $W = G_1 + G_2$ = weight of absorbent plus refrigerant
$Wx = G_1$ = weight of refrigerant
i = enthalpy, Btu per lb
Q = heat added or rejected

Consider an aqua-ammonia system operating on the simple cycle shown in Fig. 1. The volatility of water under generator conditions is a major consideration in any aqua-ammonia system. The proportion of water in the vapor leaving the generator depends on the composition of the liquid and the temperature. In the simple cycle, no provision is made for removing water from the vapor before it enters the condenser, hence the water passes through the condenser to the evaporator; and being almost non-volatile under evaporator conditions, it concentrates as evaporation proceeds, and raises the boiling point of the remaining refrigerant. To keep the evaporator operating, the accumulated water must be purged to the absorber, either continuously or from time to time, as a liquid of high ammonia content.

Assume that vapor and weak solution leave the generator in equilibrium at chosen values of generator temperature and condenser pressure; that condensation is at equilibrium for vapor of the composition so determined; that rich solution leaves the absorber at a chosen temperature and at a composition such that its vapor pressure equals a chosen suction pressure; and that vapor and purge liquid leave the evaporator together, in equilibrium at the chosen suction pressure and a chosen outlet temperature. Assume the following conditions:

Generator temperature, F	220
Condenser pressure, psig	155
Suction pressure, psig	20
Temperature of suction gas and purge liquid leaving evaporator, F	20
Temperature of rich (or strong) liquid leaving absorber, F	90

To find the material and heat balances, prepare **Table 3.** From Fig. 4, find x and i for points 1, 2, and 3 in Fig. 1, and enter

them in the table. For point 1, strong liquid leaving the absorber, the intersection of the 20 psi line with the 90 F line, gives for the liquid, $x = .405$ and $i = -16$. Points 2 and 3, gas and weak liquid leaving the generator, are found at the intersection of the 155 psi line and the 220 F line, which gives, for the gas, $x_2 = .920$ and $i_2 = 755$, and for the liquid, $x_3 = .316$ and $i_3 = 133$.

values of W_2, x_1, x_2 and x_3, combining the two equations and solving:

$$W_1 = \frac{.920 - .316}{.405 - .316} = 6.79; \quad W_3 = 5.79$$

The Wi values are next filled in by multiplying together the W and i for each point, completing Table 3.

The following quantities may now be

Table 3. Conditions in the Cycle

Point	p	t	x	i	W	Wi
1	20	90	.405	−16	6.79	−109
2	155	220	.920	755	1.00	755
3	155	220	.316	133	5.79	770
4	155	92	.920	113	1.00	113
5	20	20	.920	414	1.00	414

Point 4 is liquid leaving the condenser and point 5 is the mixture of vapor and purge liquid leaving the evaporator. The material passing through the condenser and evaporator is that leaving the generator as gas, hence, on the customary basis of one pound of liquid leaving the condenser:

$$x_2 = x_4 = x_5 = .920$$

and

$$W_2 = W_4 = W_5 = 1$$

For point 4, at $x = .920$ for the liquid and a pressure of 155 psi, the chart gives $i_4 = 113$ and a condenser temperature of 92 F. For point 5, at 20 psi and 20 F, the chart gives, for the purge liquid, $x_5' = .753$ and $i_5' = -27$, and for the vapor, $x_5'' = .999$ (nearly) and $i_5'' = 622$. Neglecting the trace of water in the vapor, the weight of purge liquid is

$$W_2 = \frac{x_5'' - .92}{x_5'' - x_5'} = \frac{.079}{.246} = .321$$

Then i_5 for the mixture is:

$$i_5 = .321\ (-27) + (1 - .321)\ 622 = 414$$

For the generator, the material balances give:

$$W_1 - W_2 - W_3 = 0$$

and

$$W_1 x_1 - W_2 x_2 - W_3 x_3 = 0$$

Substituting the previously determined

calculated. Per pound of liquid leaving the condenser:

Refrigerating effect $= i_5 - i_4 = W_5 i_5 - W_4 i_4$ = 301 Btu

Heat input to generator by steam $= W_2 i_2 + W_3 i_3 - W_1 i_1$ = 1,634 Btu

Heat removed from condenser $= W_2 i_2 - W_4 i_4$ = 642 Btu

Heat removed from absorber $= W_3 i_3 + W_5 i_5 - W_1 i_1$ = 1,293 Btu

To get the refrigerating effect in tons, per pound of liquid leaving the condenser **per minute,** divide 301 by 200 which equals 1.5.

The ratio of refrigerating effect to heat input is the performance factor, PF, of the cycle:

$$PF = \frac{301}{1634} = .184$$

This is decidedly low in comparison with the ideal PF of around .9 obtained in paragraphs 8 and 9. The largest losses are those due to heat carried away from the generator by the weak liquid, and to the water volatilized in the generator and carried into the evaporator. Additional equipment such as described later reduces the losses.

III. Industrial Absorption System

1. Conventional machine. The simple ammonia-water cycle as described previously is very uneconomical and can be

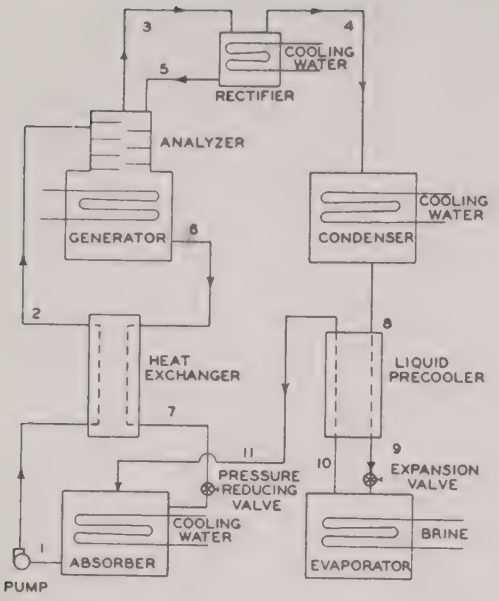


Fig. 6. Simple System with Auxiliary Apparatus

greatly improved by auxiliary apparatus. Some of the auxiliary equipment described in the following paragraphs may be omitted in actual application, depending on the combination that will give the best economic balance between first cost and operating cost.

Two parts for reducing the amount of water in the condenser and evaporator are in general use: the analyzer and the rectifier. The **analyzer** consists of a chamber through which the vapors leaving the generator pass in counterflow contact with the strong liquid coming to the generator. Either bubble cups or packing may be used in order to bring the vapor into better contact with the liquid. As the vapor passes upward through the analyzer it is cooled and enriched by ammonia, and the liquid is heated. Thus the vapor going to the condenser is lower in temperature and richer in ammonia, and the heat input to the generator is decreased. The ammonia concentration of the vapors can be increased still further by refluxing above the point where the strong liquid is introduced, but this will increase the heat input to the generator.[1]

The **rectifier** is simply the inlet part of the condenser arranged in such a manner that its condensate drains back to the generator or analyzer. The rectifier may be cooled by water or by strong liquid.

It is universal practice to use a **counterflow heat exchanger** for the strong liquid flowing from the absorber to the generator and for the weak liquid flowing from the generator to the absorber. The strong liquid leaves the absorber far below its boiling point and the weak liquid must be cooled before it enters the absorber; therefore the liquid heat exchanger permits a substantial reduction in input and saves appreciably on cooling water.

Liquid ammonia passing from the condenser to the evaporator is usually cooled by bringing it into **heat exchange** with the **suction gas** leaving the evaporator. Since a cooler liquid allows greater refrigerating effect per unit of weight, this heat exchanger gives additional refrigeration at no cost except a slightly higher temperature cooling water from the absorber.

The **weak liquid precooler** is a water-cooled section for cooling the weak liquid

Table 4. Conditions in Cycle

Point	p	t	x	i	W	Wi
1	20	90	.405	−16	7.64	−122
2	155	187	.405	94	7.64	720
3	155	187	.963	716	1.08	773
4	155	130	.993	667	1.00	667
5	155	130	.600	53	.08	4
6	155	220	.316	133	6.64	882
7	155	108	.316	6	6.64	40
8	155	86	.993	136	1.00	136
9	155	20	.993	61	1.00	61
10	20	10	.993	580	1.00	580
11	20	76	.993	665	1.00	655

before it enters the pressure-reducing valve. It has nearly the same effect as the above-mentioned liquid gas precooler and may be installed in addition to the latter.

Stickney[6] and others[7] have worked out the effect on the heat balance of the addition of each of the above auxiliary devices to the simple cycle, and also their effects in combination. **Fig. 6** gives the hook-up of a machine employing some of them and **Table 4** gives the heat balance figures for this cycle, based on the following conditions and arrived at by the means explained previously.

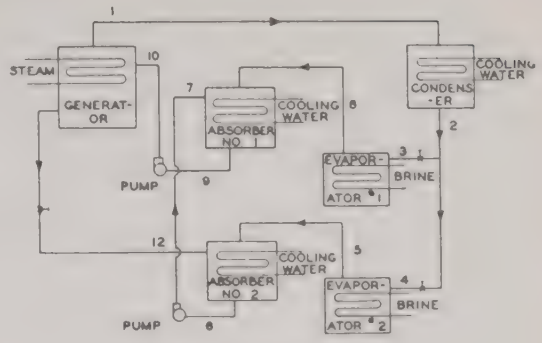


Fig. 7a. Two-Stage System

Suction pressure, psig	20
Suction temperature, F	10
Temperature of strong liquid from absorber, F	90
Generator temperature, F	220
Condenser pressure, psig	155
Temperature of vapors leaving rectifier, F	130

The liquid heat exchanger is of such a size that the strong liquid leaving it is saturated. The rectifier and liquid gas precooler are of such a size that the liquid ammonia is cooled to 20 F and the suction gas is heated to 10 F below liquid ammonia temperature leaving the condenser.

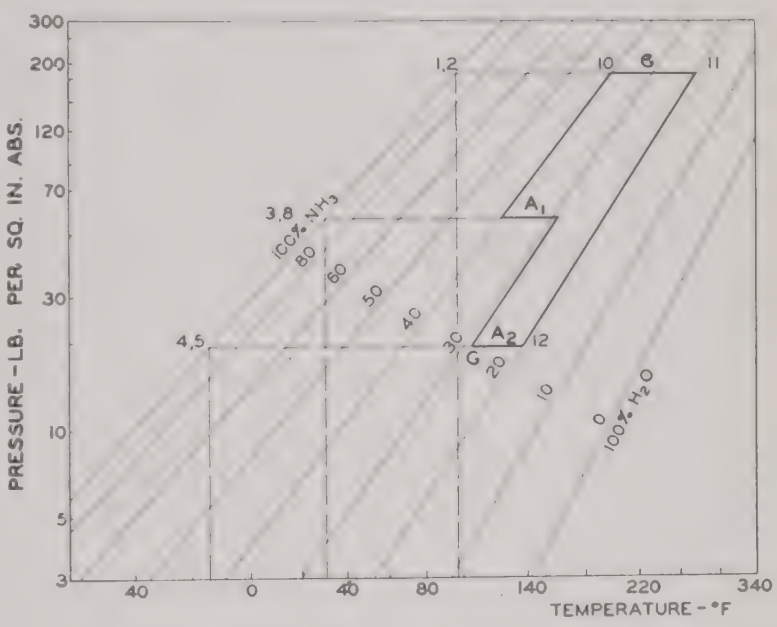


Fig. 7b. Cycle for Two-Stage System

From the table the following quantities can be obtained:

Refrigerating effect, $(W_9i_9 - W_{10}i_{10})$	= 519 Btu
Heat input to the generator $(W_2i_2 - W_3i_3 + W_5i_5 - W_6i_6)$	= 931
Total heat to the system	=1,450

Heat rejected in the absorber $(W_7i_7 + W_{11}i_{11} - W_1i_1)$	= 817
Heat rejected in the condenser $(W_4i_4 - W_8i_8)$	= 531
Heat rejected in rectifier $(W_3i_3 - W_4i_4 - W_5i_5)$	= 102
Total heat rejected	=1,450

All the above figures are per pound of the liquid from the condenser. The cycle efficiency of this process is then 519/931 =55.6%. Thus it is seen that the auxiliary apparatus used in this machine has increased the efficiency almost three times as compared with the simple ammonia-water cycle described in Part II. At the same time the temperature in the evaporator is reduced by 10 F for the same suction pressure, which is also a desirable feature.

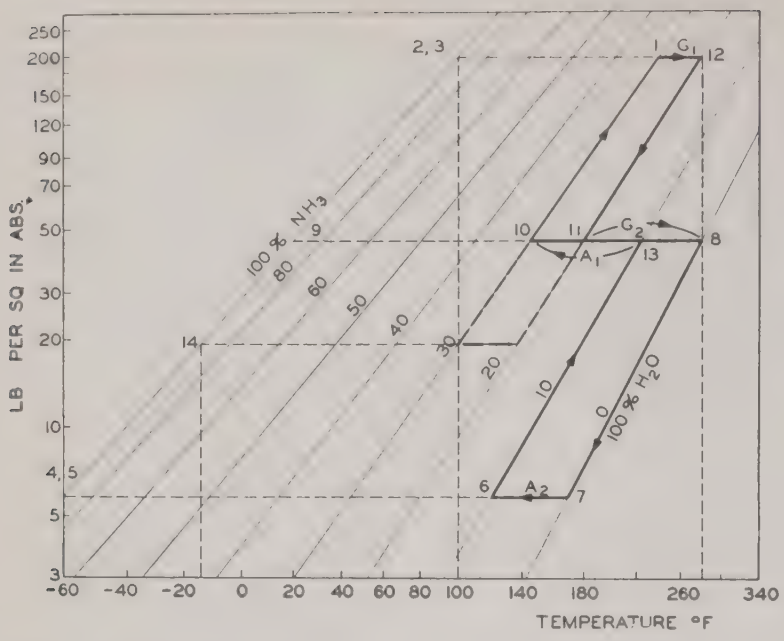


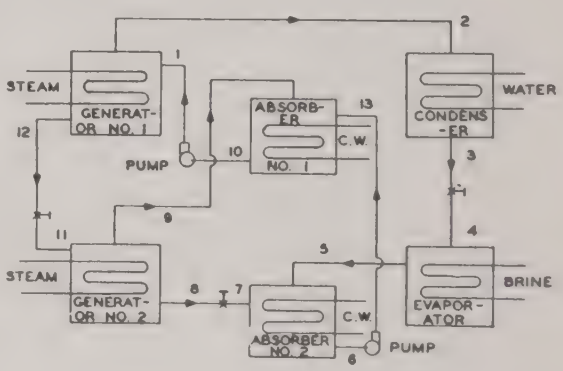
Fig. 8a (below). Another Type of Two-Stage Ammonia-Water Absorption System

Fig. 8b (left). Cycle for System of 8a

(More detailed description of the ammonia-water absorption machine can be found in reference 8.)

2. Multi-stage absorption machines. An absorption system may consist of two or more stages at different pressure levels. One arrangement, omitting for simplicity the auxiliary apparatus, is seen in **Fig. 7a.** There is one common generator and one common condenser for both stages, but special evaporators and absorbers for each stage. The flow of vapor and liquid are clear from the chart, whereby the cycle of Fig. 7b can be easily followed by using the reference numbers in Fig. 7a identifying the characteristic points. Cooling water is used in series through the condenser, low-stage absorber 2, and high-stage absorber 1. Accordingly, all points on Fig. 7b are located from left to right. The conditions at different points of the cycle can be found and the balance equations can be set up in the same manner as before.

Another example of a two-stage machine is shown in **Fig. 8a.** Here are two generators and two absorbers, one for each stage, but only one condenser and one evaporator. The advantage of this arrangement is that it permits a lower evaporator temperature at the same temperatures of the heating medium and cooling water, but at the same time it almost doubles the heat requirement. (Some other arrangements of the multi-stage machine can be found in reference 9.)



3. Resorption system. In the processes considered above, the evaporator and condenser are at constant temperature, which corresponds to the Carnot cycle. Lorenz[10] recognized and appreciated the fact that the cycle can be improved thermodynamically by using changing temperatures in the evaporator and condenser of a system similar to the changing temperature conditions taking place in the generator and absorber of the system. By arranging a counterflow between the solution leaving and entering the apparatus, the heat may be regained and the processes made more efficient.[11]

This cycle is represented in **Fig. 9a** and **b.** It differs from the absorption cycle, in

that after being boiled off in the generator, the refrigerant vapor is not condensed to a pure liquid as at point 1, but is absorbed by a special weak solution passing through the apparatus called a resorber, points 5, 2.

As the concentration changes, the temperature changes as well. Cooling water going through the resorber in a closed circuit removes the heat of condensation and absorption. From the resorber the solution goes to the evaporator through the pressure-reducing valve. There it flows in counterflow with the brine, picks up heat which boils out the refrigerant. Again, as the concentration changes the temperature changes.

The resorber and evaporator are thus fed by the special solution circuit, requiring an extra pump. Against this there are advantages of the cycle in that it can produce more cooling effect at a lower level than the usual absorption system (see the cycle shown on the dotted line), with the same temperature of condensing water without a higher pressure. It is necessary thereby to go to a lower pressure on the low side. By changing the concentration of the resorber circuit, it is even possible to reduce the high-side pressure and still reach the desired temperature in the evaporator. From the above, it follows that the absorption system may be considered a particular case of the resorption system with 100% concentration of the resorber circuit.

The resorption and absorption systems can be combined in different ways. **Fig. 10** shows one combination, which results in almost doubling the refrigerating effect of the system, as the same heat is required to run the single effect (shown by the dotted line on Fig. 10a) and the double effect of the combined system. The pressure conditions must be selected properly. Note that more equipment is required, raising the first cost of the system.

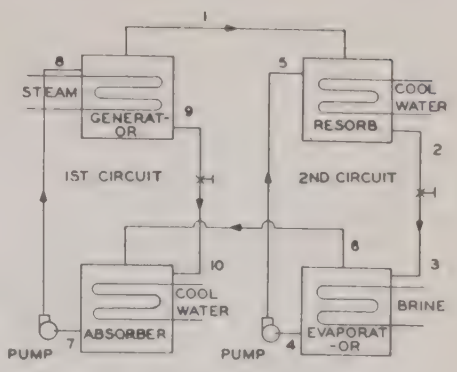


Fig. 9a. Simple Resorption System

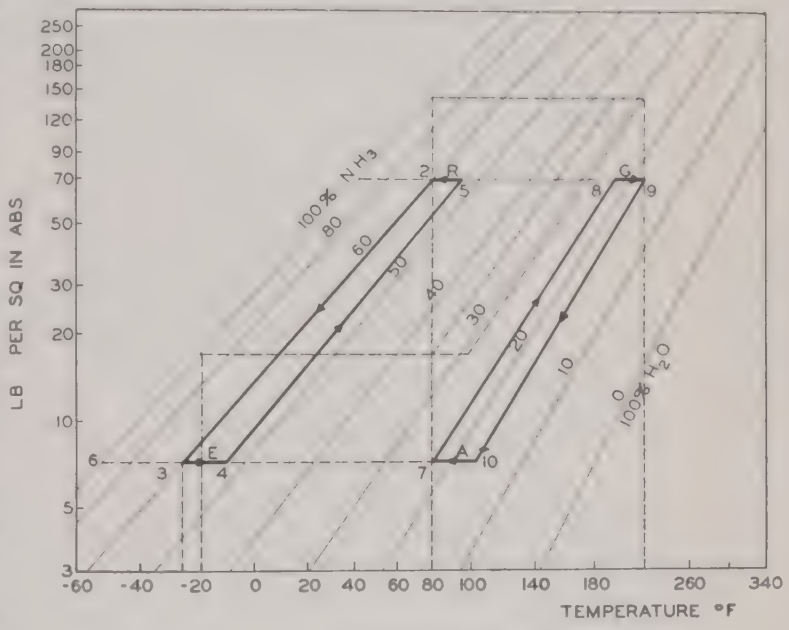


Fig. 9b. Cycle for System of Fig. 9a

Another combination for getting a low refrigerating temperature with a high condensing temperature, and a comparatively low temperature of the heating medium, was suggested by Maiuri.[12]

IV. Absorption for Air Conditioning

The absorption system may be applied to air conditioning by using it as a closed or an open system. The closed system cools and dehumidifies air by means of a cold contact surface in the same way as a compression system. The open absorption system and the open adsorption systems, which are closely related, are both known as dehydration systems. They operate by bringing the air into direct contact with

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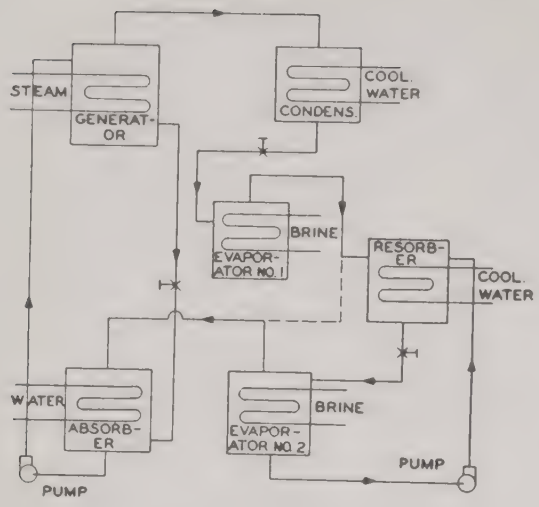

Fig. 10a. Combination Absorption-Resorption System

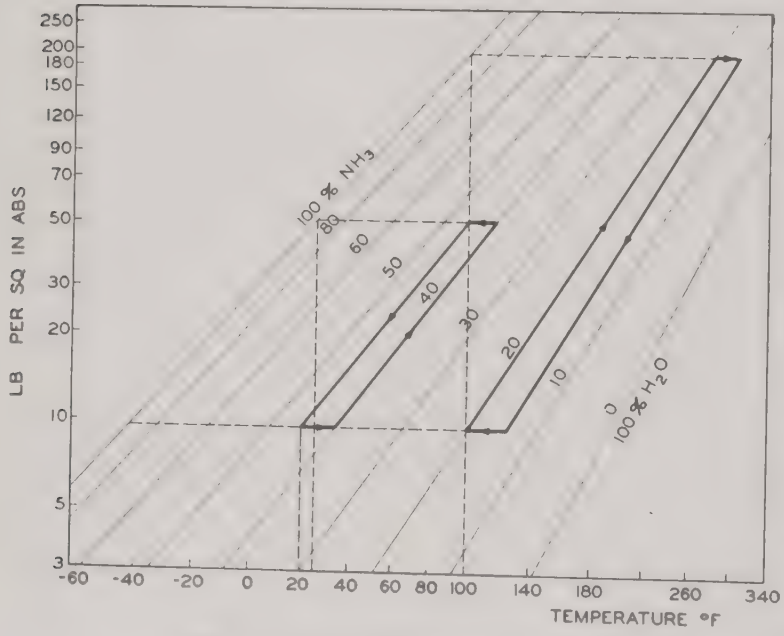

Fig. 10b. Cycle for System of Fig. 10a

the absorbent or adsorbent which has a high affinity for moisture. This dehydrates the air but also raises its temperature, therefore this sensible heat must be removed by means of a dry-air surface cooler. The air may be further cooled by evaporating water into it, provided the dehydration in the first step is extensive enough. According to May,[13] the following units were either available in 1944 or were to be some time thereafter.

1. Servel unit. This unit is a closed absorption system using water as the refrigerant and lithium bromide solution as the absorbent. It is an all-year air conditioner having cooling units of 3- and 5-ton capacity with gas inputs of 69,000 and 112,000 Btu per hr, and having heating sections of 120,000 and 180,000 Btu per hr gas input ratings. The operation of the unit is best explained by referring to **Fig. 11.**

No solution pump is used for circulating the solution. Instead, refrigerant vapor is produced in the generator in several vertical tubes, surrounded by a steam jacket, so that vapor lifting of the weak (in refrigerant) solution is obtained. From the separator at the top of the generator the refrigerant vapor passes to a water-cooled condenser from which it flows through an orifice to the evaporator. This orifice serves to maintain the pressure difference between condenserand absorber, pressure in the former being approximately 1 psia and in the latter approximately 0.15 psia. The evaporator is of the direct-expansion type in which the refrigerant is evaporated inside extended surface coils, with air passing over the outside surface. From the separator weak solution flows by gravity (and by pressure difference) through the liquid heat exchanger to the absorber. Here refrigerant vapor is absorbed and the strong solution returned by gravity through the liquid heat exchanger to the generator leveling chamber.

The water-cooled purge pump serves to collect non-condensible gases from the absorber and deliver them to a gas storage chamber.

A special boiler supplies steam under atmospheric pressure to the cooling unit generator when refrigeration is desired and to a heating coil when heating is desired. More detailed descriptions of this system may be found in references 13 and 14.

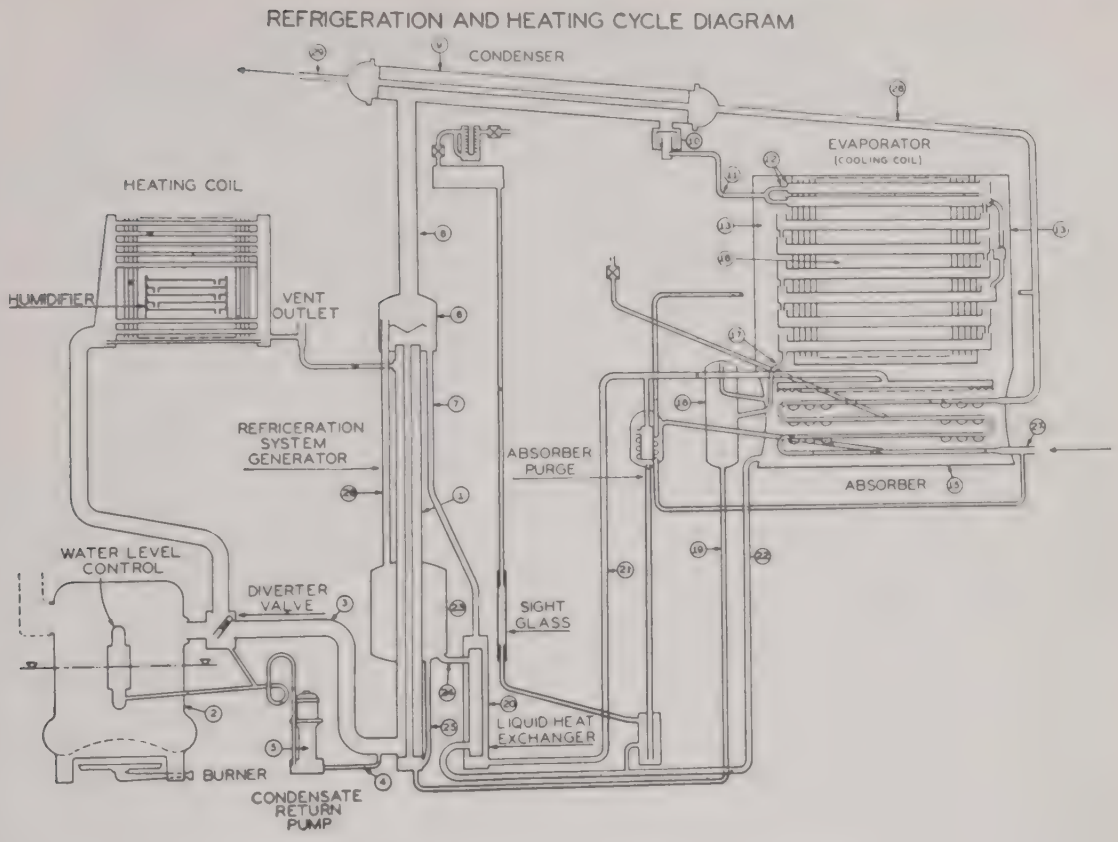

Fig. 11. Diagram of Servel All-Year Air Conditioning Unit

The cycle efficiency of the absorption cooling unit is approximately 60%.

2. The Williams Air-O-Matic unit is a closed absorption system using methylene chloride as refrigerant and dimethyl ether of tetraethylene glycol as absorbent.[16,17] With ratings of 7, 15 and (perhaps) 32 tons it is suitable for large residences and small commercial applications. It is a chilled water[16] type unit and requires a supply of low pressure steam. The boiler is not integral with the refrigeration unit.

The operation of this unit may be seen by referring to **Fig. 12.** Low-pressure steam enters the steam space of the heater where it comes in thermal contact with coils containing solution strong in refrigerant. Heated strong solution is discharged from the coils into an inner vapor release chamber where refrigerant vapor separates from the solution at a pressure of 1 to 3 psig. The vapor passes from the top of the release chamber down to the condenser, from which refrigerant liquid is discharged through a drain pipe and passed through the absorbent liquid-return heat exchanger, then through the refrigerant float valve into the water chiller. Here it is vaporized at low pressure (21 to 24 in. mercury vacuum) by absorbing heat from water circulating to the air conditioner.

Weak solution remaining after refrigerant vapor is expelled in the inner vapor release chamber flows from the bottom of the chamber through the liquid heat exchanger, and thence to the top of the absorber where it is sprayed over cooling coils. Refrigerant vapor from the water chiller passes up through a baffle plate to the top of the absorber and is absorbed by the solution passing down over the cooling coils. Strong solution from the bottom of the absorber is pumped by the solution pump through the liquid heat exchanger and on to the heater coils in the steam space.

Liquid refrigerant entering the water chiller contains a small amount of absorb-

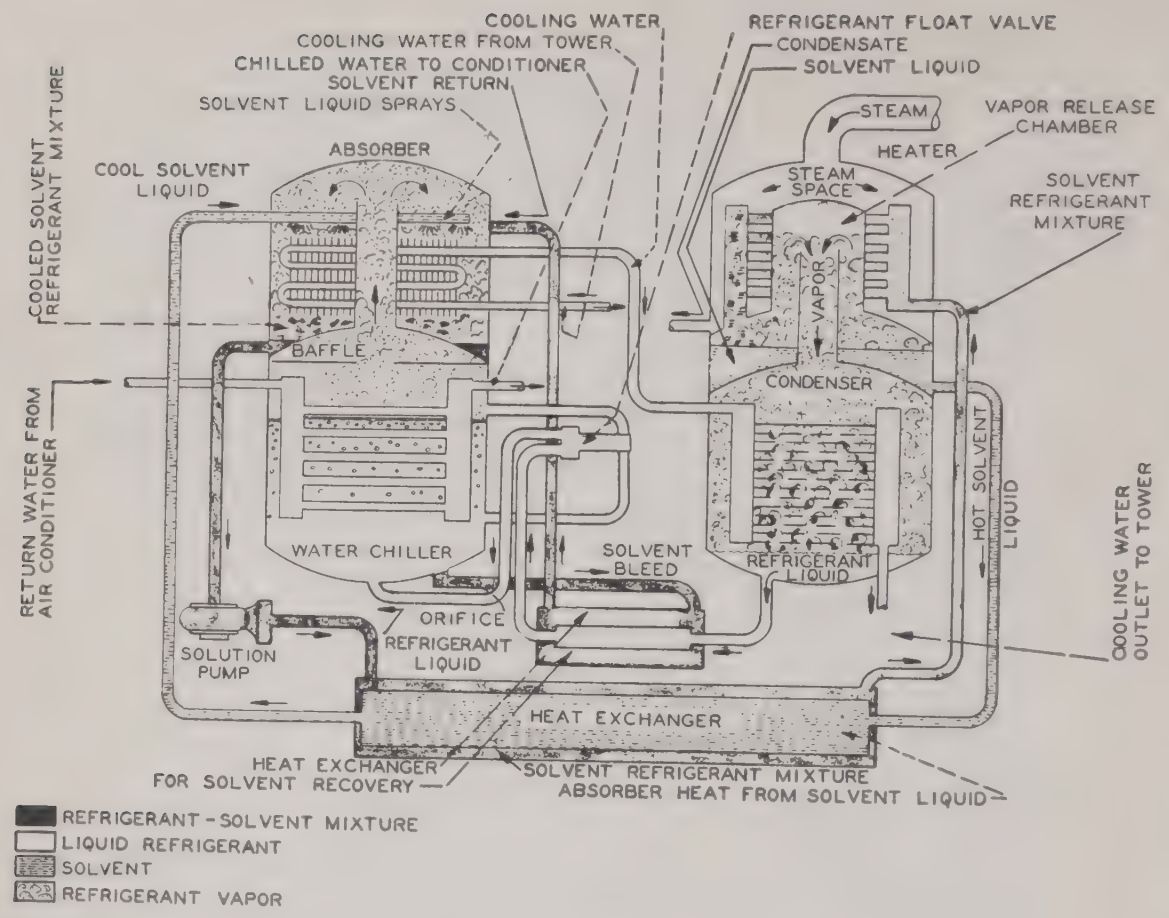

Fig. 12. Williams Air-O-Matic Absorption Unit

ent that must be removed from the chiller. This is done by continuously bleeding off a little liquid through an orifice at the bottom of the chiller and passing it in heat exchange with liquid refrigerant entering the chiller. Part of the refrigerant in the bleed-off mixture is evaporated in the heat exchanger and raises the remaining solution to the top of the absorber.

The cycle efficiency of the latest make, according to the manufacturer's catalog, is approximately 50 %.

3. The **Kathabar unit** for dehumidifying air, is an open absorption system employing Kathene (lithium chloride brine) as the absorbent. According to the flow chart of **Fig. 13,** circulated air is passed through a spray of Kathene. Moisture is absorbed from the air by the brine which then flows to a regenerator where the solution is regenerated by heating it and passing outside air over it to carry off the excess water vapor.

If insufficient moisture is present in the circulating air the system may be reversed and moisture given up to the air by the brine. Whether moisture is added to or removed from the circulated air depends on the setting of an automatic humidistat control.

A series of four standard self-contained units, including contactor and regenerator section, and with capacities of 750, 1,500, 3,500, and 5,000 cfm, is available. For build-up systems contactor cells are made in two sizes having air capacities of 3,000 and 4,000 cfm. For large installations these cells are combined in batteries and regenerating systems are sized as required to handle these batteries.

4. The **Bryant unit** is an open adsorption unit using silica gel as the adsorbent. This

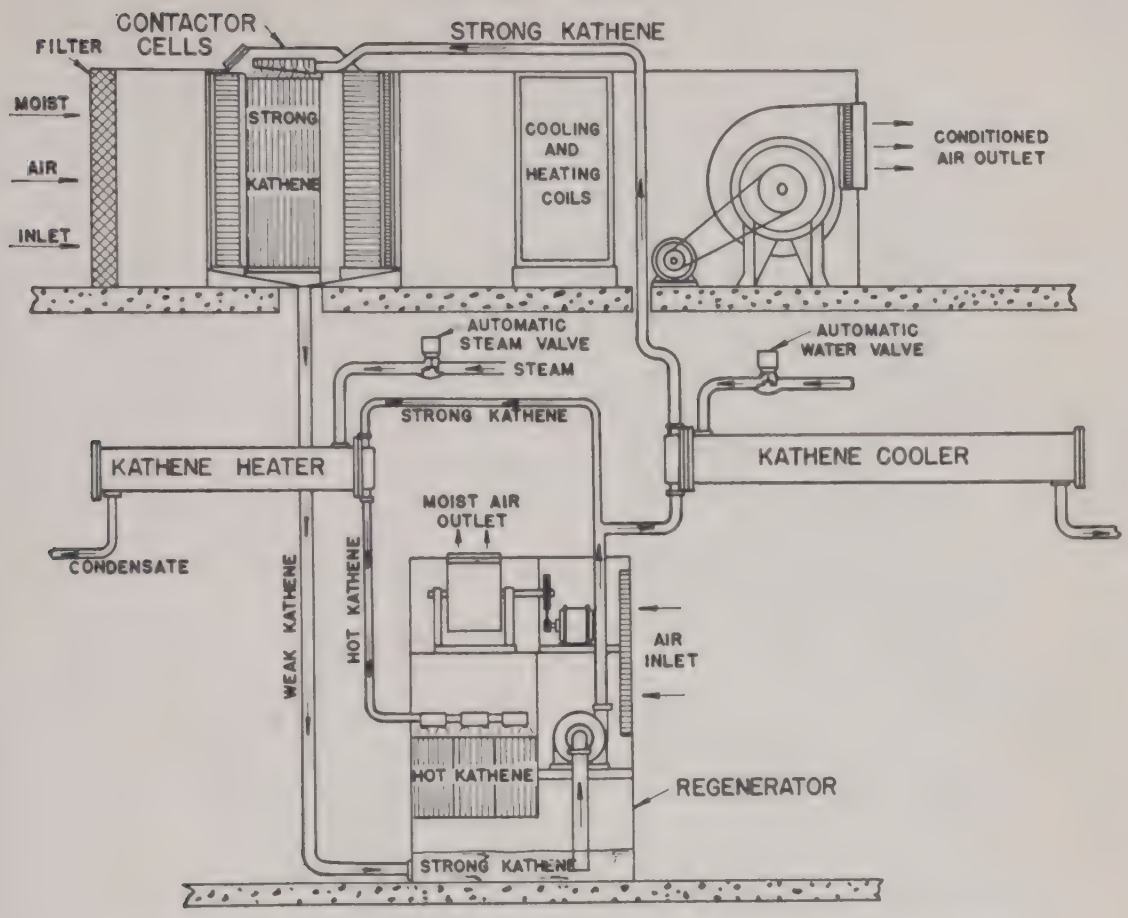

Fig. 13. Air Dehumidification under Summer Conditions with Kathene Solutions

unit, available in four sizes with nominal air capacity ratings of 500, 800, 1,300 and 2,900 cfm, may be used for commercial and residential air conditioning as well as for industrial processes. For comfort cooling purposes, companion dry air coolers and resaturators are obtainable from the manufacturer.

The cycle is illustrated in **Fig. 14.** A hollow revolving drum constructed of two concentric cylinders of fine mesh screen contains the silica gel in granular form between its walls. Interior of the drum, as well as surrounding space, is divided by means of wiper blades into two comparatively airtight sections. The rest of the cycle is apparent from the illustration. The drum is rotated at a rate of approximately one revolution in seven minutes. In the resaturation cooler air is forced through a wetted rotating drum made of large mesh screen.

In addition to the above unit, Bryant makes two sizes of stationary bed-type dehumidifiers which have air capacities of 2,700 and 5,000 cfm.

5. The **Lectrodryer unit** is an open adsorption unit using activated alumina as adsorbent. The manufacturer specializes primarily in air and gas drying equipment for all sorts of industrial processes and produces standard units in sizes ranging from 350 to 15,000 cfm air capacity. **Fig. 15** shows the arrangement of one type of Lectrodryer unit. Air to be dehumidified and heated reactivation air pass alternately through beds of activated alumina. If cooling is desired in addition to dehumidification, water coils are supplied, either imbedded in the alumina beds, or installed in outlet air ducts.

V. Costs of Systems

From past experience it has been known

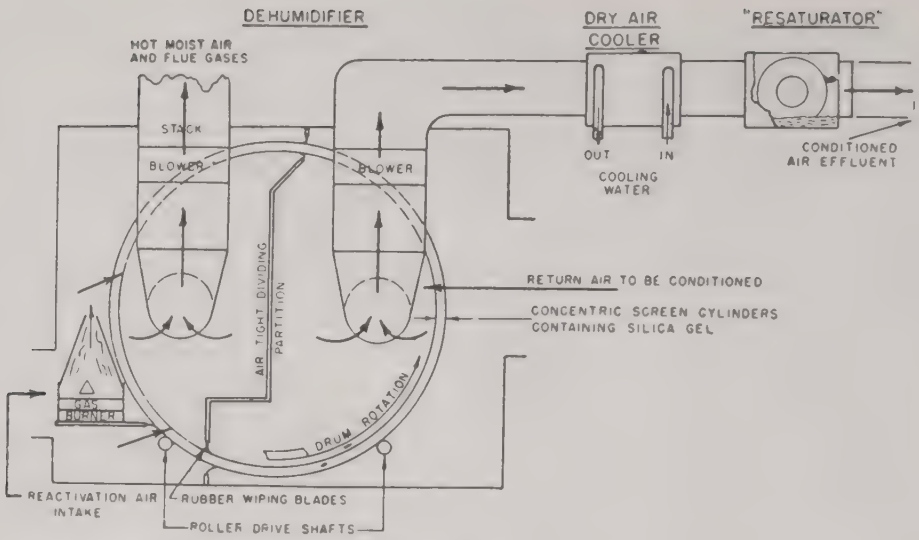

Fig. 14. Flow Diagram of Bryant Dehumidifier with Dry Air Cooler and Resaturator

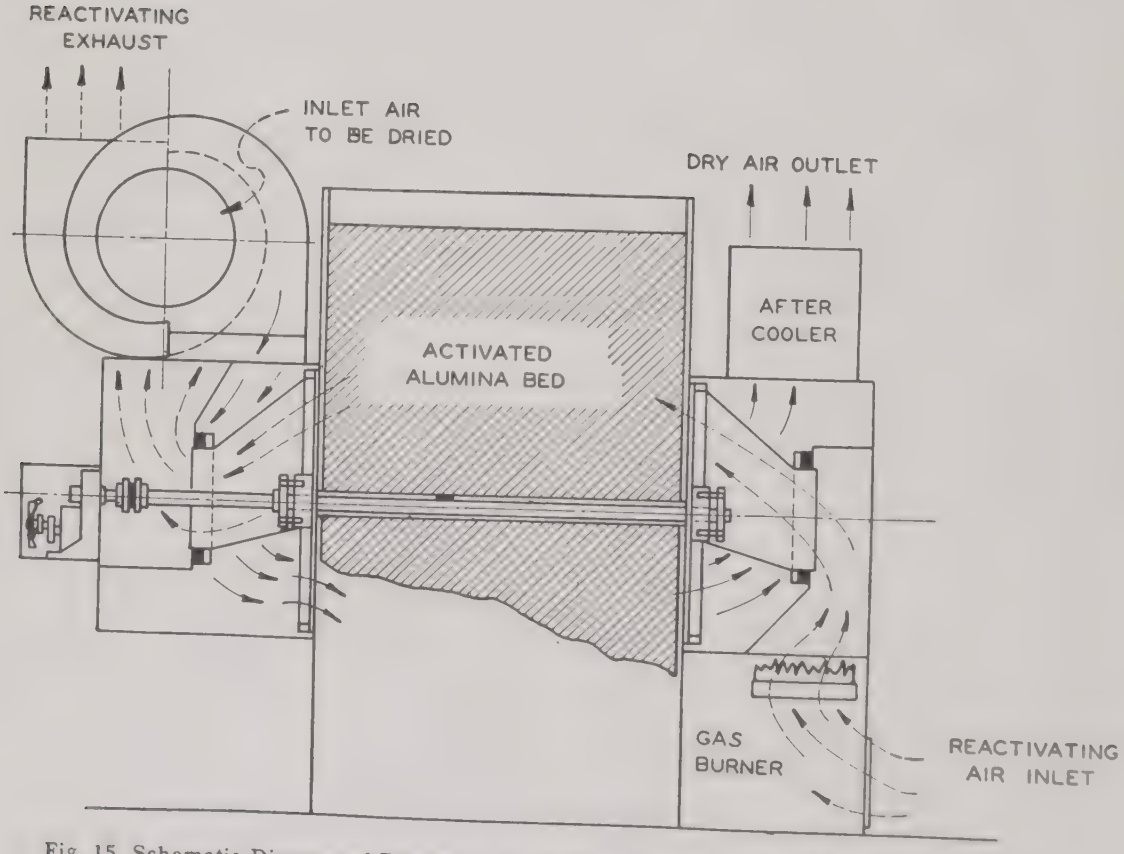

Fig. 15. Schematic Diagram of Pittsburgh Lectrodryer Using Activated Alumina as Drying Agent

that at high evaporating temperatures, possibly 0 F and up, the first cost of an absorption system is higher than that of a compression system. At lower temperatures the absorption system is comparable or even cheaper, especially when compared to reciprocating machines. Yet results obtained in recent development indicate that a well-designed absorption machine can be built inexpensively for any temperature.

At a high temperature, say 45 F, the overall efficiency of the absorption machine, including boiler losses, can be made as high as 60% for one-stage evaporation. This value is to be compared with the efficiency of transforming heat into electrical energy, a maximum at 30% with a practical value of between 20 and 25% multiplied by the coefficient of performance of the compression cycle, of about 4.5. The product is $.25 \times 4.5 = 1.125$.

Thermodynamically the compression system seems to be much better, but this is only a part of the story of operating cost, because to the cost of fuel some fixed charges must be added, which are considerably greater for the power plant than for a steam boiler or a direct-fired generator of the absorption system. If this is taken into consideration, operating cost of the absorption system will compare favorably with the compression system even at high temperatures. The cost of the absorption machine, both first and operating, is less influenced by the evaporating temperature than that of the compression machine, so that at low temperatures the absorption machine will compare still more favorably.

Other important points are: (1) The absorption machine has no prime moving parts; (2) no oil is needed at any point in this machine; (3) the machine is inherently easy to control at partial load, maintaining its full load efficiency; and (4) the machine is very adaptable to any set of working conditions, and can be fed with a low-pressure steam often available as waste.

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4. PSYCHROMETRY

I. Properties of Air

THE properties of air discussed here are presented in tabular form in Chapter 10, which should be referred to when tabular values are needed.

1.1—Dry Air Properties

Volume. Except for small, local variations in the fraction of carbon dioxide, dry atmospheric air has practically a constant composition from sea level to about 50,000-ft altitude as follows:

	Percent by volume	*Composition by weight*
Nitrogen	78.03	75.47
Oxygen	20.99	23.19
Monatomic gases, principally argon	0.94	1.30
Carbon dioxide	0.03	0.04
Hydrogen	0.01	0.00

Its equivalent molecular weight, derived by the laws of gaseous mixtures, from the above composition, is 28.966. Dividing this figure into the universal gas constant, 1545.3 ft-lb per mol deg F, results in a corresponding gas constant, R_a, of 53.349 ft-lb per lb deg F. The ideal gas equation of state resulting is:

$$v_a = \frac{R_a T}{p} = \frac{53.349 T}{p} \qquad (1)$$

$$v_a = \frac{0.754307 T}{p} \qquad (1.1)$$

Where v_a = specific volume of dry air (cu ft per lb)
T = absolute temperature, t + 459.7 (deg F abs)
p = absolute pressure, psfa in Equation (1) and in. Hg in Equation (1.1)

The volumes of dry air at standard atmospheric pressure given in the psychrometric tables of Chapter 10 are derived from a modified form of the Beattie-Bridgeman equation and are more exact than those which result from Equations (1) and (1.1). The deviations are small, being of the order of 0.25% at −100 F and less than 0.1% from 32 F to 200 F.

Enthalpy. Previously, the specific heat of dry air was taken as a constant, 0.24, so that enthalpy above 0 F could be expressed as $h_a = 0.24\ t$. Extension of the data to higher temperatures has made it imperative to recognize the **variation** of the **specific heat** of **air** with **temperature.** It seems to be generally accepted that values of the zero pressure specific heat of dry air derived from the zero pressure specific heats of its constituent gases, as calculated from spectroscopic data, are more accurate than values derived from direct thermal measurements.

The constant pressure specific heat at other pressures may be derived from the zero pressure data and an equation of state like the Beattie-Bridgeman equation using exact thermodynamic relations. This work of others was used in constructing **Fig. 1,** as follows: The zero pressure curve is a mean plot of the values given in Cornell University Experiment Station Bulletin No. 30,[1] and the Sage and Lacey tables.[2] A cross plot of the graphical data in Bulletin No. 30 indicated that c_p was essentially a linear function of pressure in the range of temperatures from 0 F to 200 F. This information, together with the correlated zero pressure data, served to locate accurately the curves of c_p at standard atmospheric pressure and the other higher pressures. Values of c_p at 29.921 in. Hg abs, calculated from the Goff and Gratch tables,[3] are indicated, providing a good check of the standard atmosphere line.

It has been noted above that the variation of specific heat with pressure is related to the fact that the volume of air does not exactly follow the simple equation of state (Equations (1) and (1.1)). In spite of this apparent inconsistency, the use of the specific heat data in Fig. 1 will improve the accuracy of computations at low temperatures and high pressures, even though

the volumes used in the reduction of air flow data are derived from the simple equation. For exceedingly precise work, the Beattie-Bridgeman equation[1] may be used

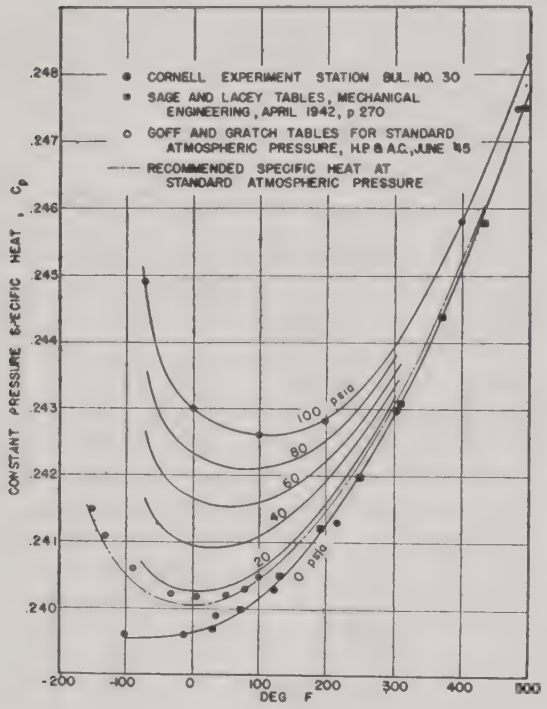

Fig. 1. Constant Pressure Specific Heat of Dry Air

in place of Equation (1) for computing the volume of dry air.

1.2—Water Properties

Vapor pressure. The most accurate and consistent vapor pressure data available are apparently those developed by Goff and Gratch.[4] Below the triple point temperature and pressure (32.0176 F and 0.18167 in. Hg, respectively), a metastable liquid state exists. In other words, pure liquid water resists freezing if no ice crystals are present to initiate the process. The vapor pressure of liquid water below the triple point is higher than that of ice at the same temperature. This, it will be seen, results in two possible values for certain of the psychrometric properties of moist air. In the tables, emphasis is given the properties corresponding to the vapor pressure of the stable ice phase.

Enthalpy, h_g. The enthalpy of saturated water vapor, h_g, is used in preparing moist air properties. An accurate value becomes of increasing importance at higher temperatures where more water vapor may be contained by the air. In preparing the psychrometric tables, values for h_g, in Btu per lb of steam, as given in the tables of water vapor properties,[4] were used, but the following empirical equations in terms of Fahrenheit temperature, t, will generally provide satisfactory accuracy:

For the range t $= -100$ F to 70 F
$$h_g = 1061.7 + 0.439\ t$$
For the range t $= 70$ F to 200 F
$$h_g = 1063.9 + 0.410\ t$$

Similar equations for the enthalpy of liquid water and ice are:

$$h_f \text{ (for water)} = (t - 32)$$

In the range 32 F to 212 F

$$h_f \text{ (for ice)} = -159 + 0.48\ t$$

Below 32 F

Specific gravity referred to air, G. By definition, the specific gravity of water vapor referred to air is the ratio of the densities of water vapor and dry air at the same temperature and pressure. Alternatively, it may be expressed as the ratio of specific volumes which, of course, are simply the reciprocals of the densities. Thus

$$G = \left(\frac{v_a}{v_w}\right)_{p_w, T} = \frac{53.349}{R_w} \qquad (2)$$

Where G = specific gravity of water vapor referred to air
v_a = specific volume of dry air from Equation (1) at pressure p_w and temperature T
v_w = specific volume of water vapor at its own partial pressure, p_w, and the absolute dry-bulb temperature T

$$R_w = \frac{p_w v_w}{T}$$

Since water vapor does not follow the ideal gas laws exactly, R_w is a variable depending on pressure and temperature, resulting in a corresponding variation in G as shown in **Fig. 2.** If water vapor behaved as a perfect gas, its specific gravity

referred to air as defined above would be constant, and equal to the ratio of molecular weights of water vapor and air.

$$G = \frac{M_w}{M_a} = \frac{18.016}{28.966} = 0.6220$$

It will be noted that the value of G approaches the value of this ratio at the lower temperatures.

1.3—Moist Air Properties

To get a clear picture of the properties of moist air, we must first understand exactly what moist air is. Suppose one pound of air absolutely devoid of any water content is placed in a container in which the pressure may be maintained constant at atmospheric pressure. If a quantity of liquid water is now added to the container, a portion of it will evaporate and become mixed with the air in the form of water vapor. The remainder will continue to exist as a liquid in the bottom of the container. The process of evaporation requires the addition of a certain quantity of heat which, if the container is uninsulated, will be supplied by the surroundings. In this case, the temperature of the surroundings will be assumed to be 50 F.

If the container remains long enough at 50 F, an equilibrium condition will be established in which the air, water vapor, and liquid water will all be at exactly the same temperature. When this condition is established, the air is said to be saturated with water vapor and the material in the container, exclusive of the liquid water, is called **saturated air.**

The theory of psychrometrics based on Dalton's Principle assumes that, under saturation conditions such as those described above,

(1) The total pressure, p, is equal to the sum of the water vapor pressure, p_g'', at dew-point temperature and the partial pressure of the dry air, p_a.

$$p_a + p_g'' = p$$
$$p_a = p - p_g'' \qquad (3)$$

(2) The partial pressure of the water vapor is equal to the saturation pressure of water as found in the steam tables corresponding to 50 F.

(3) Both the air and water vapor fractions in the saturated air behave as though the other were not present. Consequently, the volume of the mixture per lb of dry air may be calculated from Equation (1.1), us-

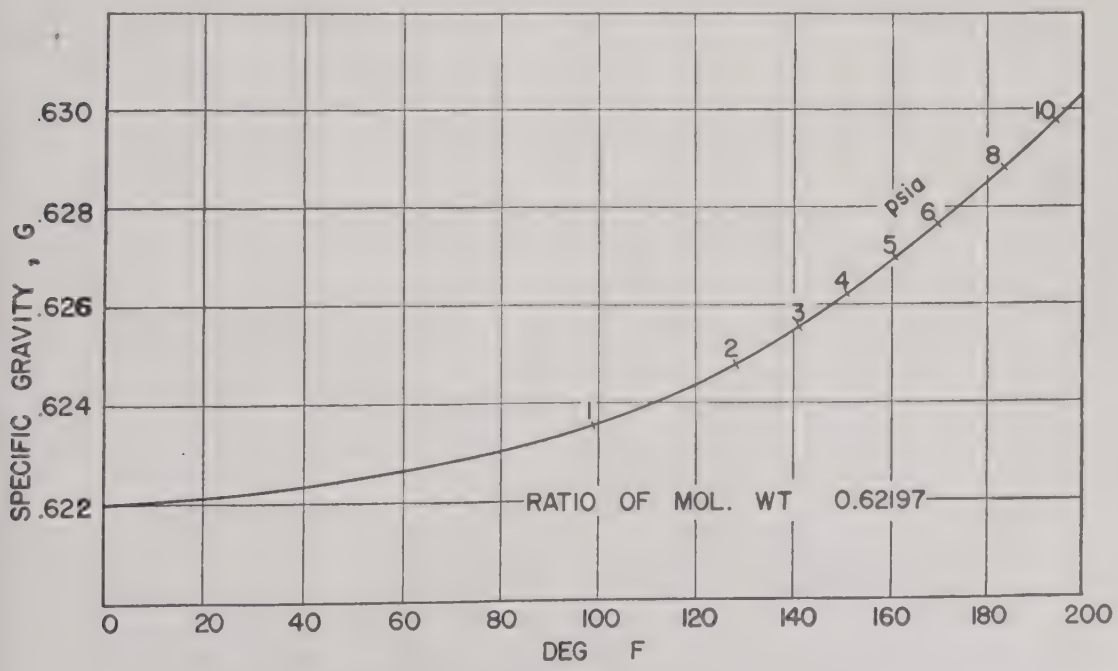

Fig. 2. Specific Gravity of Water Vapor Referred to Dry Air

ing the air partial pressure deduced from Equation (3).

$$v = \frac{R_a T}{p_a} = \frac{0.75430T}{p - p_g''} \qquad (4)$$

Where v = volume of moist air, lb per lb dry air
R_a = gas constant for air, 0.75430
T = absolute temperature, $t + 459.7$ (deg F abs)

(4) The enthalpy of the air-water vapor mixture is equal to the sum of the specific enthalpy of each constituent in Btu per lb, multiplied by the pounds of the constituents present in the mixture.

$$h = h_a + Wh_g$$
$$h - h_a = Wh_g \qquad (5)$$

Where h = enthalpy of moist air (Btu per lb dry air)
h_a = enthalpy of dry air (Btu per lb)
h_g = enthalpy of saturated water vapor at dry bulb temp, t (Btu per lb dry air)
W = specific humidity of air (lb per lb dry air).

Returning to the container with its saturated air and liquid water at 50 F, suppose the water is removed and the remaining moist air is heated at atmospheric pressure to 80 F. Obviously, the specific humidity, W, remains constant. It will be seen later that the water vapor pressure also remains substantially constant, although it does increase slightly. The condition of the water vapor is therefore changed and, corresponding to a pressure slightly above the vapor pressure at 50 F, is at a temperature of 80 F. It has become superheated steam. The air in this condition is said to be unsaturated because more water vapor could be added until the vapor pressure increased to that of water at 80 F.

If, instead of heating the 50 F saturated air, an attempt is made to cool it below 50 F, the actual vapor pressure will tend to exceed that corresponding to the lower temperature and condensation will occur. It is clear, then, that the temperature at which air of a certain moisture content becomes saturated upon cooling at constant pressure is the temperature at which condensation will begin. This temperature is called the **dew point.**

To avoid confusion, it must be noted that the moisture content, enthalpy and volume of moist air are always given **per lb of dry air in the mixture.** This practice, which deviates from that used in most other fields (wherein such properties are usually given **per lb of mixture**), is used because in psychrometric problems the quantity of water vapor continually varies while that of the dry air remains unchanged. The moisture content associated with 1 lb of dry air is called **specific humidity.**

Specific humidity, W. If the water vapor and dry air in a moist air sample are considered as occupying the same volume, the **moisture content**, usually called specific humidity, in lb of water vapor per lb of dry air, is defined by the ratio of the water vapor (ρ_w), and **air densities** (ρ_a) computed at their respective partial pressures. Since **specific volume** is the reciprocal of density, we may write

$$W = \frac{\rho_w}{\rho_a} = \frac{v_a}{v_w} \qquad (6)$$

From Equation (1) the specific volume of air at a given temperature is inversely proportional to pressure. Using Equation (2), the above equation may therefore be written for the dew-point condition

$$W = \left(\frac{v_a}{v_g}\right)_{p_g'', t''} \times \frac{p_g''}{p - p_g''} = G\,\frac{p_g''}{p - p_g''} \qquad (7)$$

Where W = specific humidity, lb of water vapor per lb of dry air
p_g'' = vapor pressure of water at dew-point temperature, t''
p = total or barometric pressure
v_a = specific volume of dry air
v_g = specific volume of water vapor at partial pressure p_g'' and absolute dew-point temperature t''.

Volume. By the assumption that in the mixture of water vapor and air, each constituent behaves as though the other were not present, it is clear that the net effect of adding water vapor to one pound of dry

air is simply the reduction of the air pressure below the total pressure, p, by the amount of the water vapor pressure,

$$v = v_a \times \frac{p}{p - p_g''} \qquad 8)$$

$$v - v_a = v_a \left(\frac{p}{p - p_g''} - 1 \right)$$

$$= v_a \frac{p_g''}{p - p_g''} = v_a \frac{W}{G} \qquad (9)$$

Where v = volume of moist air, cu ft per lb dry air
v_a = specific volume of dry air at the same temperature.

The last expression in Equation (9) is obtained by substituting from Equation (7) the ratio of specific humidity, W, to specific gravity of water vapor, G (at the dew-point temperature).

Enthalpy, h. It is obvious that Equation (5) represents the enthalpy of saturated air in Btu per lb of dry air. It is not so obvious, however, that this same equation also satisfactorily defines the enthalpy of unsaturated air, provided h_g is taken as the enthalpy of saturated water vapor at the dry-bulb temperature. To be exact, it would be necessary to use a value of enthalpy for the water vapor corresponding to its superheated condition, but at the relatively low vapor pressures encountered in moist air the enthalpy of superheated steam is almost exactly equal to that of saturated steam at the same temperature. Even at the upper limit of the psychrometric tables (200 F), a maximum error of less than 0.2% would occur from this approximation in the derived value of enthalpy for unsaturated air. The enthalpy of dry air and of saturated air would, of course, be more exact.

Relative humidity and saturation ratio. These two properties of air are defined by the following equations:

$$\text{Relative humidity,} \quad \phi = p_g''/p_g \qquad (10)$$

$$\text{Saturation ratio,} \quad \mu = W/W_s \qquad (11)$$

Where p_g'' = vapor pressure of water at dew-point temperature (in. Hg or psia)
p_g = vapor pressure of water at dry-bulb temperature (in. Hg or psia)
W = specific humidity, of air (lb per lb dry air)
W_s = specific humidity of saturated air at dry-bulb temperature (lb per lb dry air).

It is easy to show that, although these two properties are nearly equal at the lower end of the temperature range (say below 100 F), they are not actually identical. Writing Equation (7) for saturated and unsaturated air at the same temperature and total pressure, p, it is found that

$$\mu = \frac{W}{W_s} = \frac{p_g''}{p - p_g''} \Big/ \frac{p_g}{p - p_g}$$

$$= \frac{p_g''}{p_g} \times \frac{p - p_g}{p - p_g''} = \phi \times \frac{p - p_g}{p - p_g''} \qquad (12)$$

Where p = total or atmospheric pressure.

Other symbols have the same significance as in Equations (10) and (11) above.

Moist air properties tables. The psychrometric tables in Chapter 10 are for atmospheric pressure but they include all the properties of water necessary for the computation of moist air properties at pressures higher or lower than atmospheric pressure. In the tables W_s = specific humidity of saturated air in grains per pound of dry air (1 lb = 7000 grains)

v_a = specific volume of dry air (cu ft per lb)
v_s = volume of saturated air (cu ft per lb of dry air)
h_a = enthalpy of dry air (Btu per lb)
h_s = enthalpy of saturated air (Btu per lb dry air).

It is obvious from inspection of Equations (5) and (9) that the increase in enthalpy due to the addition of water vapor is directly proportional to the specific humidity, W, and the increase in volume is practically so. This enables the determination of the volume or enthalpy of unsaturated air by a process of interpolation, using values of volume and enthalpy given in the psychrometric tables.

Thus

$$h = h_a + \mu(h_s - h_a) = h_s - (1-\mu)(h_s - h_a) \quad (13)$$

$$v = v_a + \mu(v_s - v_a) = v_s - (1-\mu)(v_s - v_a) \quad (14)$$

Interaction effects. It has already been noted that the individual properties of dry air and water vapor deviate slightly from the simple gas laws. Similarly, there is some evidence that the combination represented by moist air does not strictly follow the dictates of Dalton's Principle. Interactive molecular forces, and the slight solubility of air in any liquid water in contact with saturated air, result in a very small increase in moisture content of saturated air and other related properties.

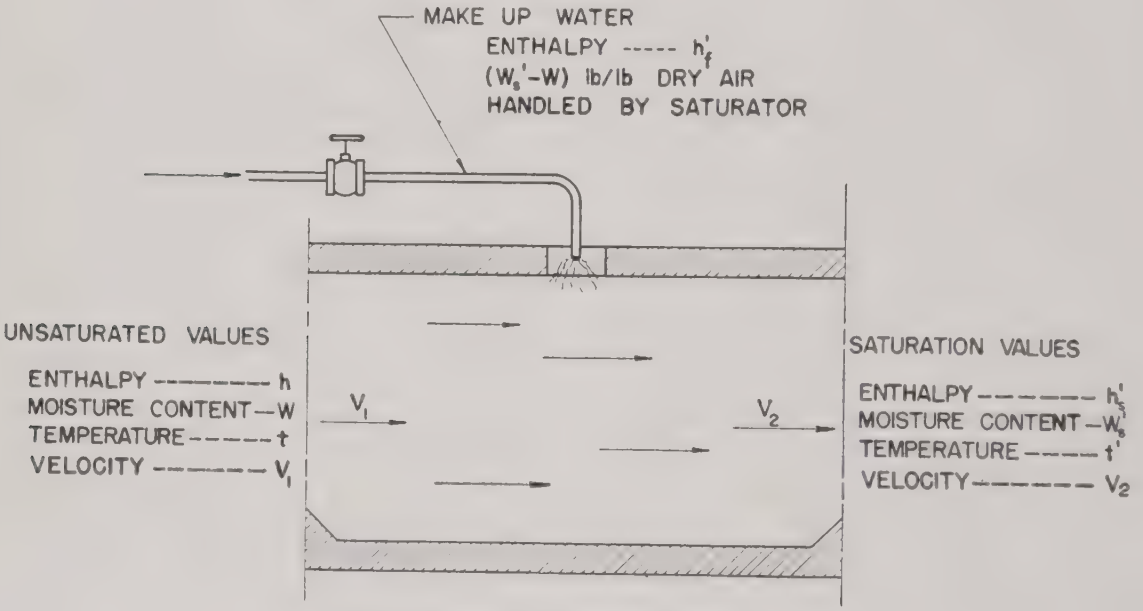

Fig. 3. Diagram of Adiabatic Saturator

Goff[3] has considered these factors in the preparation of his psychrometric data for standard atmospheric pressure. **Table 1** shows a comparison of his values with values from the tables in Chapter 10, which are based on the simpler psychrometric theory outlined above and ignore the interaction and other secondary effects. Little or no information exists on the effect of pressure on these phenomena.

Evaporative cooling and thermodynamic wet bulb. Referring to **Fig. 3,** if a stream of unsaturated air having temperature t, enthalpy h, and specific humidity W, is passed through a perfectly insulated chamber where it comes in contact with large wetted surfaces, it will leave in a substantially saturated condition with temperature t', enthalpy h_s' and moisture content W_s'. If the process is continous, a quantity of liquid water $(W_s' - W)$ must be supplied for each pound of air passing through the saturator. Now, let it be assumed that:

1. There is no pressure drop in the process.
2. The temperature of the water added equals t' and its kinetic energy is zero.
3. There is no mechanical work or heat added during the saturation process.

Under such conditions, the "steady flow" equation of thermodynamics requires that

$$h + (1+W)\frac{V_1^2}{2gJ} + (W_s' - W)h_f' = h_s' + (1+W_s')\frac{V_2^2}{2gJ} \quad (15)$$

The superscripts indicate values at the thermodynamic wet-bulb temperature, t', defined below.

Where, in addition to the symbols described above,

V = velocity, fps

g = acceleration of gravity, 32.2 fps^2

Table 1. Saturation Properties

	Neglecting interaction effects (Section 10)			Including interaction effects[3]		
Temp, F	W_s	h_s	v_s	W_s	h_s	v_s
−50	.2893	−11.97	10.31	.2914	−11.969	10.314
−60	.1471	−14.40	10.06	.1483	−14.394	10.059
Diff*	.1422	2.43		.1431	2.425	
50	53.40	20.28	13.00	53.61	20.301	13.001
40	36.34	15.21	12.70	36.49	15.230	12.695
Diff*	17.06	5.07		17.12	5.071	
200	16052	2675.6	77.14	16065	2677	77.142
190	7682	1298.9	44.94	7693	1301	44.959
Diff*	8370	1376.7		8372	1376	

* Differences in specific humidity (W) and enthalpy (h) for a 10 F temperature range, since comparison of absolute values has little practical significance.

J = mechanical equivalent of heat (778 ft-lb per Btu)

If the velocity terms are neglected, this is reduced to

$$h+(W_s'-W)h_f'=h_s' \quad \text{or}$$
$$h-Wh_f'=h_s'-W_s'h_f' \tag{16}$$

It will be noted that the enthalpy of the saturated air, h_s', its specific humidity, W_s', and the enthalpy of liquid water, h_f', are all defined by the leaving air temperature, t'. Therefore, for a given t'

$$h_s'-W_s'h_f'=\Sigma$$

where this equation defines the term Σ known as the Sigma function.

Obviously, there are any number of unique moist air samples which will satisfy Equation (16), ranging from dry air, whose enthalpy $=h_s'$, to saturated air at t'. All such samples are said to have the same thermodynamic wet-bulb temperature.

Equation (16), which defines the **thermodynamic wet bulb** (twb) and the process of evaporative cooling which it describes, are important because both are very closely representative of the conditions in the film of air surrounding a wet-bulb thermometer (thermometer with bulb covered by a wetted wick). In fact, it is possible to calibrate, either theoretically or experimentally, the readings of a wet-bulb thermometer so that the twb may be obtained. Given the twb and the dry-bulb temperature, Equation (16) may be solved for W and all the related psychrometric properties may be determined.

Hence it may be said that the **process** of **evaporative cooling** to the twb is the fundamental principle on which the wet-dry bulb psychrometer is based.

The velocity terms of Equation (15) were neglected in the development of the twb concept. There is no objection to this procedure under most circumstances. At higher velocities, of the order of 70 fps and above and for precise laboratory work, the magnitude of the velocity terms in Equation (15) should be investigated. If the kinetic energy entering the saturator is dissipated in friction, the pressure remaining constant, the resulting rise in twb is less than the corresponding rise of db. The wet-bulb depression is consequently greater. On the other hand, if the kinetic energy is reduced to zero by compression, a somewhat higher twb rise results and the wb depression deviates less from that determined by neglecting the velocity terms.

II. The Psychrometric Chart

2.1—Description of the Charts

The psychrometric chart is the most convenient means for determining the **properties** of **moist air.** It is particularly useful for representing processes in which air is cooled, heated, humidified, dehumidified, or mixed with air from other sources.

The charts used here contain an outstanding improvement over charts formerly available. Enthalpy of unsaturated air has previously been difficult to show on the psychrometric chart, because the constant enthalpy lines are so nearly parallel to the wet-bulb lines that both cannot be shown without causing confusion. Thus some previous charts have shown enthalpy scales that were only approximately correct, tending to large errors at high temperatures. Other charts used the "Sigma heat function" which, by definition, required a correction for the enthalpy of the moisture contained in the air. In the present charts the enthalpy is given on a scale of enthalpy at saturation. The readings from this scale are used in combination with a set of deviation contours showing the **deviation of enthalpy** for conditions other than saturation. This combination gives exact values, or when desired the deviation lines may be ignored with results that will be sufficiently correct for most problems in the comfort air conditioning range. The new type of chart is particularly valuable at temperatures below 32 F and at high temperatures above comfort conditions. In both of these ranges, the deviation of enthalpy for conditions other than saturation cannot be ignored without causing large errors.

The following properties can be determined from the charts, as illustrated in **Fig. 4:**

Dry-bulb temperature, represented by vertical lines, is read along the lower edge of the chart.

Wet-bulb temperature, represented by diagonal lines, is read at the intersection of these lines with the saturation curve. Below 32 F the charts show properties of moist air based upon the vapor pressure over ice.

Specific humidity in grains or pounds per pound of dry air, represented by horizontal lines, is read on the scales at the right-hand side of the chart.

Dew-point (saturation) temperatures, constant along horizontal lines parallel to the specific humidity lines, are read along the saturation curve. Note that dry-bulb, wet-bulb and dew-point temperatures coincide at the saturation curve.

Percent relative humidity is represented by a family of curved lines with values indicated thereon. Interpolation between lines of relative humidity is done along vertical or dry-bulb lines, since spacing is approximately uniform in that direction.

Volume per pound of dry air is indicated by diagonal lines with values marked thereon.

Enthalpy at saturation is read by extending the wet-bulb lines to intersection with the diagonal scale.

Enthalpy deviation for conditions other than saturation is indicated by curved, dashed lines with values marked thereon. Note that below 32 F wet bulb these lines are nearly parallel to the saturation line and are much closer together, indicating a greater deviation. Interpolation between enthalpy deviation lines should be done along wet-bulb lines, since spacing is uniform in that direction.

Enthalpy of water or ice added to or rejected from the system is read on small diagrams included with the main chart. The enthalpy of the rejected or added water constitutes a small but definite part of the heat balance. For the approximate solution of comfort air conditioning problems, it is often neglected, and the error from this source seldom exceeds 1%. The error is much greater, however, at high temperatures and at temperatures below freezing. The enthalpy of the rejected or added water can be conveniently read from the small diagrams. In order to use these diagrams, simply determine the difference in moisture between the entering and leaving state, determine the temperature at which the water leaves or enters the system, and read its enthalpy directly from the small chart. Note that where the moisture freezes on the coil, a much greater enthalpy is involved and it is negative, so that it becomes an addition to the cooling load.

Sensible heat factor. In the application of air conditioning equipment, certain methods of calculation involve the ratio of the sensible heat to the total heat load. This ratio is approximately defined as the change in enthalpy which takes place between the two dry-bulb temperatures at the mean moisture content to the total enthalpy difference. A scale along the right-hand margin of the normal tempera-

ture chart, together with an origin at 80 db and 50% relative humidity, affords a convenient means of reading the sensible heat factor.

Specific properties. It has been found convenient in practice to represent the enthalpy, volume and moisture content of air on the basis of the weight of dry air or **per pound of dry air.** The weight of mixture is obtained by simply adding the weight of the moisture or by multiplying the weight of dry air by $(1+W)$, where W equals the pounds of moisture (i.e., specific humidity) per pound of dry air. Simularly, the volume occupied by one pound of mixture is obtained by dividing the volume per pound of dry air by $(1+W)$.

Air is generally measured and referred to on a cubic foot basis and, therefore, it is often necessary to determine moisture content and heat content per cubic foot. This is conveniently accomplished by dividing the moisture content or enthalpy per pound of dry air by the volume per pound of dry air. The result will be moisture content or enthalpy per cubic foot of mixture.

Barometric pressure. The relative proportion of moisture in the air for any particular set of wet-bulb and dry-bulb temperatures depends upon the total or barometric pressure. The psychrometric chart represents the properties of air for standard barometric pressure of 29.92 in. of mercury. A convenient table is provided for extending these properties to other barometric pressures. For psychrometric problems in the comfort air conditioning range, where the highest accuracy is not required and the correction for barometric pressure does not exceed 1 in. of mercury, it is sometimes ignored.

Symbols Used in Text and Examples

t = dry-bulb temperature, F
t' = wet-bulb temperature, F
t'' = dew-point temperature, F
W = specific humidity, lb of water vapor per lb of dry air or grains of water vapor per lb of dry air (7,000 grains = 1 lb)
W_s = Specific humidity of air saturated with water vapor at the dry-bulb temperature, lb per lb of dry air or grains per lb of dry air
T = absolute temperature, equals (F temperature + 459.7)
p = total or barometric pressure, inches of mercury (in. Hg)
p_a = partial pressure of air, in. Hg
p_g'' = pressure of saturated water vapor at the dew-point temperature, in. Hg
p_g = pressure of saturated water vapor at the dry-bulb temperature, in. Hg
v = volume, cu ft per 1 lb of dry air with its associated water vapor
v_a = sp volume of air; volume in cu ft occupied by 1 lb of dry air at pressure p_a and temperature t
v_w = volume in cu ft occupied by 1 lb of water vapor at its partial pressure p_w and temperature t
ρ_a = density $= 1/v_a$
ρ_w = density $= 1/v_w$
ϕ = relative humidity, p_g''/p_g
μ = saturation ratio (also known as percentage humidity) $= W/W_s$
h = enthalpy of moist air, Btu per lb of dry air (and associated water vapor)
h_s' = enthalpy of air saturated at the wet-bulb temperature, Btu per lb dry air
h_a = enthalpy of dry air, Btu per lb
h_g = enthalpy of saturated water vapor at dry-bulb temperature, Btu per lb
h_m = enthalpy change from water added or rejected from system, in Btu per lb of dry air
d = enthalpy deviation, Btu per lb of dry air
$h = h_s' + d$ = enthalpy (true) of moist air as found from charts, Btu per lb of dry air
m = Weight of moisture added to or rejected from air stream per lb of dry air, in grains or lb
q = Heat added to or removed from system, Btu per lb of dry air

Subscripts 1, 2, 3 etc. indicate entering and progressive state points in the system (t_1, t_2, t_3, etc.)

2.2—Examples

Example 1. Properties of Moist Air (see Fig. 4). Find the properties of moist air when the dry bulb is 80 F and the wet bulb is 67 F.

Solution (read directly from psychrometric chart for normal temperatures)

Specific humidity (moisture content) = (W) = 78.2 gr per lb of dry air
Enthalpy at saturation (h_s') = 31.62 Btu per lb of dry air
Enthalpy deviation (d) = −0.1 Btu per lb of dry air
True enthalpy (h) = 31.52 Btu per lb of dry air
Specific volume (v) = 13.8 cu ft per lb of dry air
Relative humidity (ϕ) = 51%
Dew point (t'') = 60.3 F

Example 2. Heating Process (see **Fig. 5**). Air is heated by a steam coil from 30 F dry bulb and 80% relative humidity to 75 F dry bulb. Find the relative humidity, wet-bulb temperature, and dew point of the heated air.

Solution (from normal-temperatures psychrometric chart)

(t'') Dew point = 25.2 F
(t_2') Wet-bulb temperature of heated air = 51.5 F
(ϕ_2) Relative humidity = 15%

Example 2a. Find the heat added per lb of dry air in Example (2).

Approximate Solution
$h_{s1}' = 10.1$
$h_{s2}' = 21.1$
$q = h_{s2}' - h_{s1}' = 11$ Btu per lb of dry air

(Note that this result is 1.5% higher than the following exact solution.)

Exact Solution
$h_1 = h_{s1}' + d_1 = 10.1 + .06 = 10.16$ Btu
$h_2 = h_{s2}' + d_2 = 21.1 - 0.1 = 21.0$ Btu
$q = h_2 - h_1 = 10.84$ Btu

Example 3. Spray or Evaporative Cooler (see **Fig. 6**). Air at 95 db and 70 wb passes through a water spray where its relative humidity is increased to 90%. The spray water is recirculated and the make-up water enters at 70 F. Determine the leaving dry-bulb temperature, wet-bulb temperature, change in enthalpy of the air, and the amount of moisture added per lb of dry air.

Solution. Since the make-up water enters at the wet-bulb temperature of 70 F and no heat is added to or removed from the system, this is by definition an adiabatic process. There will be no change in the wet-bulb temperature and the leaving dry-bulb temperature can, therefore, be determined from the chart, by following the 70 F wet-bulb line to the 90% relative humidity line, to be 72.2 F.

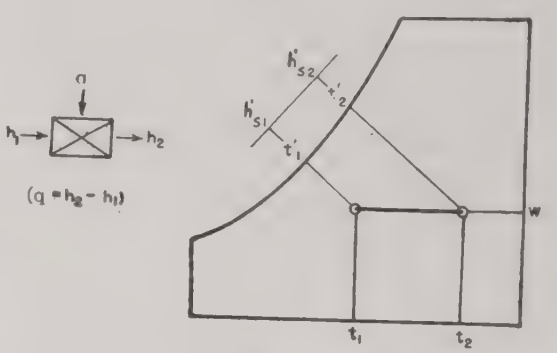

Fig. 5. Heating Process

The only change of enthalpy of the air is that due to the enthalpy of the make-up water. This can be demonstrated as follows:

$W_1 = 70$ gr per lb of dry air
$W_2 = 107$ gr per lb of dry air
$m = W_2 - W_1 = 37$ gr per lb of dry air
$h_1 = h_{s1}' + d_1 = 34.1 - 0.22 = 33.88$ Btu
$h_2 = h_{s2}' + d_2 = 34.1 - 0.02 = 34.08$ Btu
h_m = (from small diagram for 37 gr at 70 F) = 0.2 Btu
$q = h_1 - h_2 + h_m = 33.88 - 34.08 + 0.2 = 0$

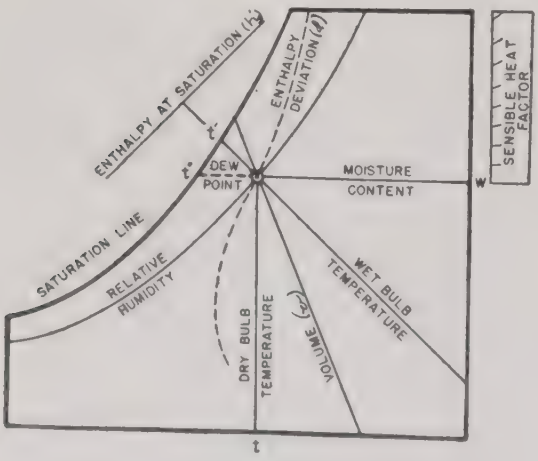

Fig. 4. Diagram of Psychrometric Chart Showing Properties of Moist Air

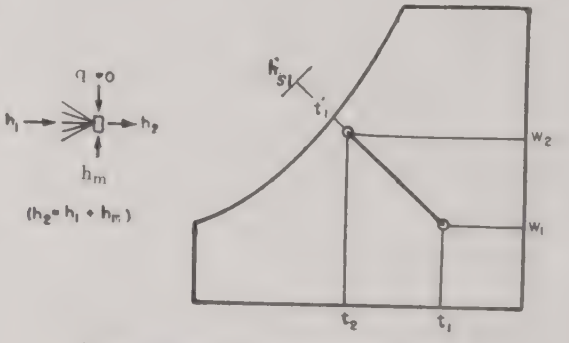

Fig. 6. Spray or Evaporative Cooler

Example 4. Cooling and Dehumidifying Process. (see **Fig. 7**). Find the cooling effect per pound of dry air where the air enters a cooling coil at 83 db, 69 wb and leaves at 56 db, 55 wb and the condensate is rejected at 55 F.

Approximate Solution
$h_{s1}' = 33.24$ Btu
$h_{s2}' = 23.21$ Btu
$q = h_{s1}' - h_{s2}' = 10.03$ Btu

(Note that this value is 1.9% greater than the following exact solution.)

Exact Solution

$h_1 = h_{s1}' + d_1 = 33.24 - 0.12 = 33.12$ Btu
$h_2 = h_{s2}' + d_2 = 23.21 - 0.01 = 23.20$ Btu
$m = W_1 - W = 84 - 63 = 21$ gr
h_m = (from small diagram, 21 gr at 55 F) = 0.08
$q = h_1 - h_2 - h_m = 33.12 - 23.20 - 0.08 = 9.84$

Example 4a. For the conditions of Example (4) determine the sensible heat factor.

Solution. As shown in Fig. 7, draw a line between the entering and leaving conditions. Then extend a parallel line from the origin located at 80 db and 50% relative humidity to the sensible heat factor scale and read the sensible heat factor of 0.68.

Example 4b (see Fig. 7). Find the cooling load per pound of dry air due to infiltration of room air at 80 db, 67 wb into a cooler maintained at 30 db, 28 wb, where moisture freezes on the coil which is maintained at 20 F.

Approximate Solution

$h_{s1}' = 31.62$ Btu
$h_{s2}' = 10.10$ Btu
$q = h_{s1}' - h_{s2}' = 21.52$ Btu

(Note that this result is 5% lower than the following exact solution.)

Exact Solution

$h_1 = h_{s1}' + d_1 = 31.62 - .1 = 31.52$ Btu
$h_2 = h_{s2}' + d_2 = 10.10 + .06 = 10.16$ Btu
$m = W_1 - W_2 = 78 - 19 = 59$ gr
h_i = (from diagram, 59 gr, 20 F) = -1.26 Btu
$q = h_1 - h_2 - h_i = 31.52 - 10.16 + 1.26 = 22.62$

Example 5. Cooling Tower (see **Fig. 8**). Determine the water consumption and the amount of heat dissipated per 1,000 cu ft of entering air at 90 db, 70 wb, when the air leaves at 110 F saturated and make-up water is at 75 F.

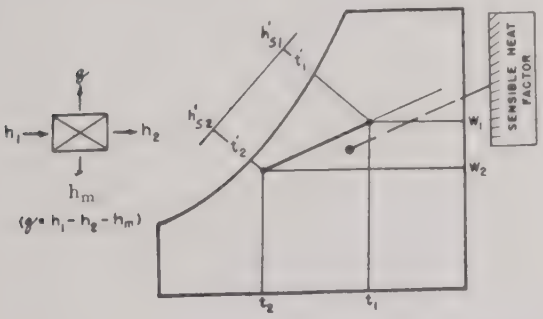

Fig. 7. Cooling and Dehumidifying Process

Approximation Solution (high temperature chart)

$m = W_2 - W_1 = 416 - 78 = 338$ gr added
$q = h_{s2}' - h_{s1}' = 92.34 - 34.1 = 58.24$ Btu added
Specific volume of entering air = 14.1 cu ft per lb of dry air

$$\text{Water consumption per 1,000 cfm} = \frac{338 \times 1000}{7{,}000 \times 14.1} = 3.43 \text{ lb}$$

$$\text{Heat dissipated per 1,000 cfm Btu} = \frac{58.24 \times 1{,}000}{14.1} = 4{,}130$$

(Note that this result is 3.4% greater than the following exact solution.)

Exact Solution

$m = W_2 - W_1 = 416 - 78 = 338$ gr
$h_m = 2.08$ Btu (from diagram on high temperature chart, 338 gr, 75 F)
$h_1 = h_{s1}' + d_1 = 34.1 - 0.18 = 33.92$
$h_2 = h_{s2}' + d_2 = 92.34 + 0 = 92.34$
$q = h_2 - h_1 - h_m = 92.34 - 33.92 - 2.08 = 56.34$
Specific volume of entering air = 14.1 cu ft per lb of dry air

$$\text{Heat dissipated per 1,000 cfm} = \frac{56.34 \times 1{,}000}{14.1} = 3{,}990$$

Water consumption will be the same as shown above.

Example 6. Mixture of Air (see **Fig. 9**). Inside air at 75 db and 62 wb is mixed with outside air at 95 db and 75 wb in the proportion of one part outside air to three parts inside air by weight. Find the properties of the resulting mixture.

Solution. Draw a straight line between the inside and outside state points. Measure off one-fourth of the length of this line, starting from the inside state. This will establish a point representing the mixture of one part outside air with three parts inside air. Properties of the mixed air are then determined as explained in Example (1).

It is often convenient to use the dry-dulb temperature scale or the specific humidity scale as the means of dividing the mixture line into the desired proportions. In this example, for instance, one-fourth of the difference between the inside and outside dry-bulb temperatures is 5 F, and thus the mixture of one part outside air with three parts inside air will be at a temperature of 75+5=80 F. At the intersection of 80 F dry-bulb temperature and the line representing the mixture, read the specific humidity of 71.3 gr, wet bulb of 65.6 F and enthalpy of 30.6 Btu.

When the two air quantities being mixed are at widely different temperatures, there will be a very

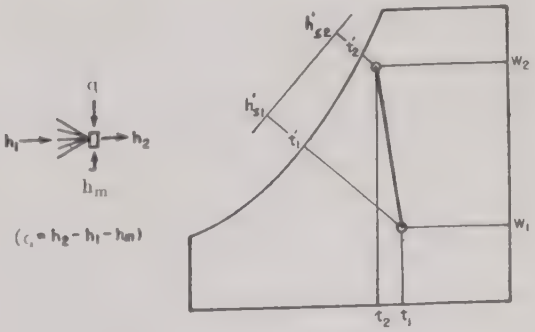

Fig. 8. Cooling Tower

small error in the above method due to the slight variation of the specific heat of moist air. When an exact solution is required under such conditions as laboratory work, it is recommended that the mixture be calculated on the basis of specific humidity and enthalpy.

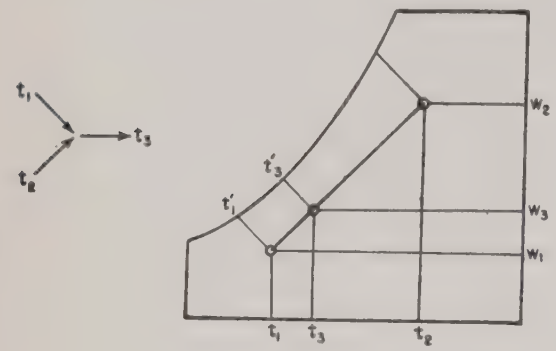

Fig. 9. Mixture of Air

Use of the Chart at Barometric Pressures other than 29.92 in. Mercury

Each chart is provided with a table showing the change of specific humidity at saturation, $\Delta W_s'$, and change of enthalpy, Δh, for various barometric pressures. True values of **specific humidity** (W) and **enthalpy** (h) are obtained by adding the corrections corresponding to the actual barometer reading to the values taken from the chart.

The specific **volume** of moist air in cu ft per lb of dry air can be determined for any barometric pressures by use of the formula:

$$v = \frac{0.754(t+460)}{p}\left(1+\frac{W}{4{,}360}\right)$$

Where: v = volume of moist air per lb of dry air, cu ft
t = dry-bulb temperature
p = barometric pressure, in. Hg.
W = specific humidity in gr per lb of dry air corrected for the actual barometric pressure.

Relative humidity and **dew point** can be determined for other than standard barometric pressures by the use of a table of vapor pressures and a simple calculation. To determine either relative humidity or dew point, it is first necessary by use of the following formula to calculate the vapor pressure of the moisture that is present in the air.

$$p_v'' = \frac{W \times p}{4{,}360+W}$$

Where p = barometric pressure
p_v'' = vapor pressure, in same units used for barometric pressure
W = specific humidity, in gr per lb of dry air

Dew point is then read directly from a table of vapor pressures as the temperature corresponding to the calculated vapor pressure.

Relative humidity is determined by dividing the vapor pressure as obtained above by the vapor pressure corresponding to saturation at the dry-bulb temperature.

$$\text{Relative humidity} = \frac{p_v''}{p_v}$$

Where: p_v'' is the vapor pressure of the moisture in the air calculated as shown above.
p_v is the vapor pressure corresponding to saturation at the dry-bulb temperature.

Note that the factor 4,360 is obtained by multiplying the specific weight of water vapor referred to air by 7,000, the number of grains per pound. The specific weight of water vapor, as shown in Fig. 2, varies slightly with temperature and for accurate results the above factors should be modified as follows for different temperature ranges:

Up to 90 F, use 4,360
90 F to 150 F, use 4,380
150 F to 200 F, use 4,400

Example 7. For a barometric pressure of 25.92 in. Hg ($\Delta p = -4$), and a reading of 90 F dry-bulb and 70 F wet-bulb temperature, determine the following:

Specific humidity (W)
Enthalpy (h)
Dew point (t'')
Relative humidity (ϕ)
Volume per lb of dry air (v_a)

Solution. From the chart read specific humidity of 78 gr, corresponding to 90 db and 70 wb. From the table under $\Delta p = -4$ and $t' = 70$, read $\Delta W_s' = 17.6$ gr. This is the barometric correction for the saturated conditions, and as noted on the chart this correction should be reduced for conditions other than saturated in the ratio of 1% for each 24 F of wet-bulb depression. Thus

$$\Delta W = 17.6\left(1-\frac{20}{24}\times.01\right) = 17.5$$

and $W = 78+17.5 = 95.5$.

Enthalpy at saturation of 34.10 Btu is read on the diagonal scale of the chart for 70 wb and corrected for the deviation of 0.18 at 90 db, giving the value of $34.10-0.28 = 33.92$. To this must be added the correction of 2.75 read from the table for $\Delta p = -4$ and $t' = 70$, giving the true enthalpy of $33.92+2.75 = 36.67$.

To determine dew point first calculate the vapor pressure as follows:

$$p_v'' = \frac{W \times p}{4360+W} = \frac{95.5 \times 25.92}{4360+95.5} = .555 \text{ in. Hg}$$

Then from the table of vapor pressures determine the dew-point temperature of 61.4 F.

Relative humidity is determined by dividing the

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Barometric Pres

ADDITIVE CORRECTIONS FOR
PRESSURE DIFFERS FRO

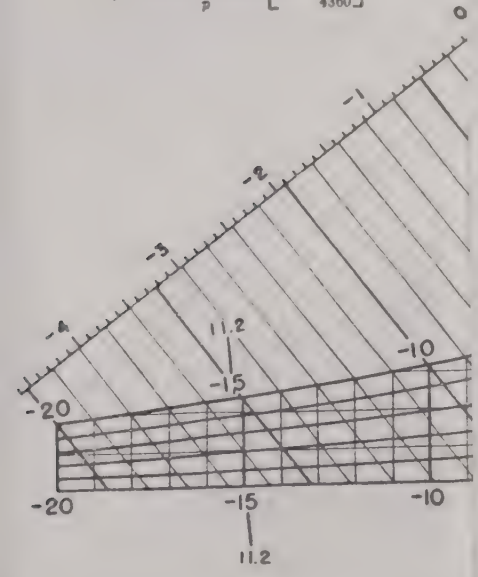
Wet Bulb Temp. t'	Sat. Vapor Press. In. Hg	APPROXI					
		−900		900		1800	
		$\Delta p = +1$		$\Delta p = -1$		$\Delta p = -2$	
		$\Delta W'_s$	Δh	$\Delta W'_s$	Δh	$\Delta W'_s$	Δh
−20	0.013	−0.06	−0.01	0.06	0 01	0.13	0.0
−18	0.014	−0.07	−0.01	0.07	0 01	0.14	0 0
−16	0.016	−0.07	−0 01	0.08	0 01	0.16	0 0
−14	0.018	−0.08	−0 01	0.09	0.01	0.18	0 0
−12	0.020	−0.09	−0.01	0.10	0 01	0.21	0.0
−10	0.022	−0.10	−0 02	0.11	0 02	0 23	0 0
− 8	0.025	−0.12	−0 02	0 12	0 02	0.26	0.0
− 6	0.027	−0.13	−0 02	0.14	0 02	0 29	0.0
− 4	0.030	−0.14	−0 02	0.15	0 02	0 32	0.0
− 2	0 034	−0.16	−0 02	0.17	0 02	0 35	0 0
0	0 038	−0.18	−0 03	0.19	0 03	0 39	0 0
2	0.042	−0 20	−0 03	0 21	0 03	0 44	0 0
4	0 046	−0 22	−0 03	0 23	0 03	0.48	0 0
6	0.051	−0 24	−0 04	0 26	0 04	0.54	0.0
8	0.057	−0.27	−0 04	0.29	0 04	0 59	0 0
10	0.063	−0.30	−0 04	0 32	0 05	0 66	0 1
12	0 069	−0 33	−0 05	0 35	0 05	0.73	0.1
14	0.077	−0 36	−0 05	0.39	0 06	0.81	0.1
16	0.085	−0.40	−0 06	0.43	0 06	0.89	0.1
18	0.093	−0.44	−0 07	0 47	0 07	0.98	0.1
20	0.103	−0.49	−0 08	0 52	0.08	1 08	0.1
22	0.113	−0 5	−0.08	0.6	0 09	1.2	0.1
24	0.124	−0 6	−0 09	0.6	0 10	1.3	0 2
26	0.137	−0 7	−0 10	0.7	0 11	1.4	0 2
28	0.150	−0.7	−0.11	0 8	0.12	1.6	0.2
30	0.165	−0.8	−0.12	0.8	0 13	1.7	0 2
32	0 180	−0.9	−0 13	0.9	0 14	1.9	0.2
34	0.197	−0 9	−0 14	1 0	0 15	2.1	0.3
36	0 212	−1.0	−0.15	1.1	0 17	2 2	0.3
38	0 229	−1.1	−0.17	1 2	0.18	2 4	0 3
40	0 248	−1.2	−0.18	1.3	0.20	2.6	0.4
42	0 268	−1.3	−0.20	1.4	0.21	2 8	0.4
44	0 289	−1.4	−0 22	1.5	0 23	3.1	0.4
46	0 312	−1 5	−0.23	1 6	0 25	3.3	0 5
48	0 336	−1.6	−0.25	1.8	0 27	3.6	0.5

t = Dry bulb temperature (F).
t' = Wet bulb temperature (F).
p = Barometric pressure (in. of Hg).
Δp = Pressure difference from standard barometer (in. of
W = Specific humidity of air (gr per lb of dry air).
W'_s = Specific humidity of air saturated at wet bulb tem
(gr per lb of dry air).
ΔW = Specific humidity correction of air when baromet
differs from standard barometer (gr per lb of dry ai
$\Delta W'_s$ = Specific humidity correction of air saturated at wet b
ature when barometric pressure differs from standar
(gr per lb of dry air).

NOTE: To obtain ΔW reduce value of $\Delta W'_s$ by
$t - t' = 24$ F and correct proportionally when $t - t'$ i

h = Enthalpy of moist air (Btu per lb of dry air).
Δh = Enthalpy correction when barometer pressure d
from standard barometer, for saturated or unsatu
air (Btu per lb of dry air).
v = Volume of moist air (cu ft per lb of dry air).

$$v = \frac{.754\,(t + 459.7)}{p}\left[1 + \frac{W}{4360}\right]$$



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NORMAL TEMPE
Barometric Pressure 29.

ADDITIVE CORRECTIONS FOR W A
PRESSURE DIFFERS FROM ST

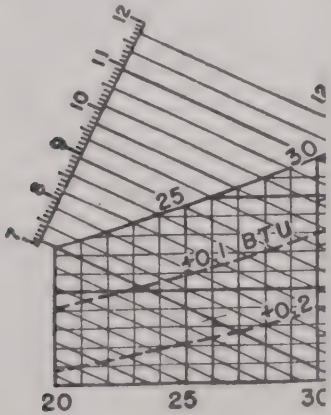
Wet Bulb Temp. t'	Sat. Vapor Press. In. Hg	APPROXIMATE ALTI							
		−900		900		1800		2700	
		$\Delta p = +1$		$\Delta p = -1$		$\Delta p = -2$		$\Delta p = -3$	
		$\Delta W'_s$	Δh	$\Delta W'_s$	Δh	$\Delta W'_s$	Δh	$\Delta W'_s$	Δh
20	.1027	−0.5	−0.08	0.5	0.08	1.1	0.17	1.7	0.26
21	.1078	−0.5	−0.08	0.5	0.08	1.1	0.17	1.8	0.27
22	.1130	−0.5	−0.08	0.6	0.09	1.2	0.18	1.9	0.28
23	.1186	−0.6	−0.09	0.6	0.09	1.3	0.19	2.0	0.30
24	.1243	−0.6	−0.09	0.6	0.10	1.3	0.20	2.1	0.31
25	.1303	−0.6	−0.10	0.7	0.10	1.4	0.21	2.2	0.32
26	.1366	−0.7	−0.10	0.7	0.11	1.4	0.22	2.3	0.33
27	.1431	−0.7	−0.11	0.7	0.11	1.5	0.23	2.4	0.35
28	.1500	−0.7	−0.11	0.8	0.12	1.6	0.24	2.5	0.36
29	.1571	−0.8	−0.12	0.8	0.12	1.7	0.26	2.6	0.38
30	.1645	−0.8	−0.12	0.8	0.13	1.7	0.27	2.7	0.40
31	.1723	−0.8	−0.13	0.9	0.13	1.8	0.28	2.8	0.41
32	.1803	−0.9	−0.13	0.9	0.14	1.9	0.29	3.0	0.43
33	.1878	−0.9	−0.14	1.0	0.15	2.0	0.30	3.1	0.45
34	.1955	−0.9	−0.14	1.0	0.15	2.1	0.32	3.2	0.46
35	.2034	−1.0	−0.15	1.0	0.16	2.1	0.33	3.4	0.5
36	.2117	−1.0	−0.15	1.1	0.17	2.2	0.35	3.5	0.5
37	.2202	−1.0	−0.16	1.1	0.17	2.3	0.36	3.6	0.54
38	.2290	−1.1	−0.17	1.2	0.18	2.4	0.37	3.8	0.5
39	.2382	−1.1	−0.18	1.2	0.19	2.5	0.39	3.9	0.6
40	.2477	−1.2	−0.18	1.3	0.20	2.6	0.41	4.1	0.6
41	.2575	−1.2	−0.19	1.3	0.20	2.7	0.42	4.3	0.6
42	.2676	−1.3	−0.20	1.4	0.21	2.8	0.44	4.4	0.6
43	.2781	−1.3	−0.21	1.4	0.22	3.0	0.45	4.6	0.7
44	.2890	−1.4	−0.22	1.5	0.23	3.1	0.47	4.8	0.7
45	.3002	−1.4	−0.22	1.6	0.24	3.2	0.49	5.0	0.7
46	.3119	−1.5	−0.23	1.6	0.25	3.3	0.51	5.2	0.8
47	.3239	−1.6	−0.24	1.7	0.26	3.4	0.53	5.4	0.8
48	.3363	−1.6	−0.25	1.8	0.27	3.6	0.56	5.6	0.8
49	.3491	−1.7	−0.26	1.8	0.28	3.7	0.58	5.8	0.9
50	.3624	−1.7	−0.27	1.9	0.29	3.9	0.60	6.1	0.9
51	.3761	−1.8	−0.28	2.0	0.30	4.0	0.63	6.3	0.9
52	.3903	−1.9	−0.29	2.0	0.32	4.2	0.65	6.5	1.0
53	.4049	−1.9	−0.30	2.1	0.33	4.4	0.68	6.7	1.0
54	.4200	−2.0	−0.31	2.2	0.34	4.5	0.70	7.0	1.0
55	.4356	−2.1	−0.32	2.3	0.35	4.7	0.73	7.3	1.1
56	.4518	−2.2	−0.34	2.4	0.37	4.9	0.76	7.6	1.1
57	.4684	−2.3	−0.35	2.4	0.37	5.1	0.79	7.9	1.2
58	.4856	−2.3	−0.37	2.5	0.39	5.3	0.82	8.2	1.2
59	.5033	−2.4	−0.38	2.6	0.41	5.4	0.85	8.5	1.3
60	.5216	−2.5	−0.40	2.7	0.42	5.7	0.88	8.8	1.3
61	.5405	−2.6	−0.41	2.8	0.44	5.9	0.91	9.2	1.4
62	.5599	−2.7	−0.43	2.9	0.46	6.1	0.95	9.5	1.4
63	.5800	−2.8	−0.44	3.0	0.48	6.3	0.98	9.9	1.5
64	.6007	−2.9	−0.46	3.2	0.49	6.5	1.02	10.2	1.5
65	.6221	−3.1	−0.48	3.3	0.51	6.8	1.06	10.6	1.6
66	.6441	−3.2	−0.50	3.4	0.53	7.1	1.10	11.0	1.7
67	.6668	−3.3	−0.51	3.5	0.55	7.3	1.14	11.4	1.7
68	.6902	−3.4	−0.53	3.7	0.57	7.6	1.18	11.8	1.8
69	.7143	−3.5	−0.55	3.8	0.59	7.9	1.23	12.2	1.9
70	.7392	−3.7	−0.57	3.9	0.61	8.1	1.27	12.7	1.9
71	.7648	−3.8	−0.59	4.1	0.64	8.4	1.32	13.1	2.0
72	.7911	−3.9	−0.61	4.2	0.66	8.7	1.36	13.6	2.1
73	.8183	−4.1	−0.63	4.4	0.69	9.0	1.41	14.1	2.2
74	.8463	−4.2	−0.66	4.6	0.71	9.4	1.46	14.6	2.2
75	.8750	−4.4	−0.68	4.7	0.74	9.7	1.52	15.1	2.3
76	.9047	−4.5	−0.71	4.9	0.77	10.0	1.57	15.7	2.4
77	.9352	−4.7	−0.73	5.1	0.79	10.4	1.63	16.3	2.5
78	.9667	−4.9	−0.76	5.2	0.82	10.8	1.69	16.9	2.6
79	.9990	−5.0	−0.79	5.4	0.85	11.2	1.75	17.5	2.7
80	1.032	−5.2	−0.82	5.6	0.88	11.6	1.82	18.1	2.8
81	1.067	−5.4	−0.85	5.8	0.91	12.0	1.88	18.8	2.9
82	1.102	−5.6	−0.88	6.0	0.94	12.5	1.96	19.5	3.0
83	1.138	−5.8	−0.91	6.2	0.97	12.9	2.02	20.2	3.1
84	1.175	−6.0	−0.94	6.4	1.00	13.3	2.10	20.9	3.2

= Dry bulb temperature (F).
= Wet bulb temperature (F).
= Barometric pressure (in. of Hg).
= Pressure difference from standard barometer (in. of Hg).
= Specific humidity of air (gr per lb of dry air).
= Specific humidity of air saturated at wet bulb temperatur
(gr per lb of dry air).
ΔW = Specific humidity correction of air when barometric press
differs from standard barometer (gr per lb of dry air).
$\Delta W'_s$ = Specific humidity correction of air saturated at wet bulb tem
ature when barometric pressure differs from standard barom
(gr per lb of dry air).

NOTE: To obtain ΔW reduce value of $\Delta W'_s$ by 1% wl
$t - t' = 24$ F and correct proportionally when $t - t'$ is not 24

= Enthalpy of moist air (Btu per lb of dry air).
= Enthalpy correction when barometer pressure differs
from standard barometer, for saturated or unsaturated
air (Btu per lb of dry air).
= Volume of moist air (cu ft per lb of dry air).

$$v = \frac{.754\,(t + 459.7)}{p}\left[1 + \frac{W}{4360}\right]$$



60
65
14 CU FT
13 CU FT
60
70
80
90
100
110
120
130
140
150
160
170
180
190
200
210
220
230
240
250
0
50
100
DRY BULB TEMPERATURE F

actual vapor pressure (.555 in. Hg) by the vapor pressure corresponding to saturation at 90 F (1.422 in. Hg) to obtain relative humidity of 39.0%.

Volume of one pound of dry air with contained moisture is determined from:

$$v = \frac{.754(t+460)}{p}\left[1+\frac{W}{4360}\right]$$

$$= \frac{.754(90+460)}{25.92}\left[1+\frac{95.5}{4360}\right]$$

$$= 16.4 \text{ cu ft per lb of dry air}$$

III. The Measurement of Moist Air Properties[5]

Humidity measurements in comfort air conditioning and industrial processing are made almost exclusively by the **wet-bulb psychrometer,** consisting of a combination of wet-bulb and dry-bulb thermometer placed in relative motion to the air. In meteorology, the psychrometer finds universal use even at the extremely low temperature of Arctic regions. The psychrometer may also be used in engineering practice at temperatures below freezing if suitable precautions are taken in its use.

Of the several methods available for measuring humidity at low temperatures, the psychrometer combines the advantages of simplicity, low cost, ease of manipulation and commercial availability. When properly used, it is entirely reliable.

The purpose here is to (1) recommend the practices and precautions necessary to obtain reliable readings with the wet-bulb psychrometer, and (2) establish the magnitude of errors that may exist under various operating conditions.

3.1—Selection of Thermometers

As in any other process, the precision required for thermometers used in psychrometry will depend upon the property being determined and the degree of accuracy required in the results. It so happens that the permissible error in thermometer readings can be easily estimated.

Regardless of temperature range, the wet-bulb reading is generally more important than the dry-bulb reading. In the comfort air conditioning range, for example, an error in the wet-bulb reading will indicate an error in specific humidity approximately three times as great as a similar error in the dry-bulb reading. This ratio decreases with temperature, until at very low temperatures the dry-bulb and wet-bulb readings approach equal importance. The determination of enthalpy depends almost entirely upon the wet-bulb reading.

For any particular process, the required accuracy of wet-bulb reading will depend upon the overall change in wet-bulb temperature. In air conditioning practice, it is not uncommon to work with changes in the wet-bulb temperature in the order of 5 to 10 F between air entering and leaving the equipment. Thus for a process involving a change of 5 F in the wet-bulb temperature, it can be assumed that an error in the wet-bulb reading of 0.1 F will result in an error of approximately 2% in either specific humidity difference or enthalpy difference. Air coolers operating in the low temperature range generally have a change in **dry-bulb temperature** between entering and leaving conditions that is considerably less than would be used in the comfort air conditioning range. However, it will be found that the change in **wet-bulb temperature** again runs in the range of 5 F. Thus for the testing of air coolers, a thermometer accuracy of 0.1 F is generally required in order to insure that the error in enthalpy difference or specific humidity difference does not exceed 2%.

For processes covering a wide range between entering the leaving wet-bulb temperatures, the higher wet-bulb temperature becomes more important than the lower one, as will be noted by referring to **Table 2.** This table, which shows the deviation of enthalpy and specific humidity corresponding to various errors in the wet-bulb readings at various temperature ranges, is useful for determining the accuracy of wet-bulb reading necessary to achieve a specified accuracy of enthalpy or specific humidity and in turn for achieving a specified percent of accuracy for the process.

Notes on the wet-bulb wicks. A suitable material for the wick is **cotton tubing** of a fairly soft, fine mesh weave. Before use, the wick should be thoroughly cleaned by washing or boiling. A snug fit on the thermometer bulb is necessary and in order to prevent excessive conduction of heat from the stem, the wick should cover about one

inch of the stem as well as the bulb. With continued use, wicks become encrusted with impurities that interfere with the proper action. It is, therefore, highly important that wicks be frequently cleaned or replaced.

Table 2

Wet bulb temp, F	Deviation of enthalpy and specific humidity, corresponding to various errors in the wet-bulb reading			
	1.0 F Error	0.5 F Error	0.2 F Error	0.1 F Error
80	1.08 Btu 5.3 gr	.54 Btu 2.6 gr	.22 Btu 1.1 gr	.11 Btu .5 gr
60	.7 Btu 2.8 gr	.35 Btu 1.4 gr	.14 Btu .56 gr	.07 Btu .03 gr
40	.46 Btu 1.4 gr	.23 Btu .70 gr	.09 Btu .28 gr	.05 Btu .14 gr
20	.35 Btu .7 gr	.18 Btu .35 gr	.07 Btu .14 gr	.04 Btu .07 gr
0	.28 Btu .3 gr	.14 Btu .15 gr	.05 Btu .06 gr	.03 Btu .03 gr
−20	.26 Btu .1 gr	.13 Btu .05 gr	.05 Btu .02 gr	.03 Btu .01 gr

The ice-coated bulb. The wick covering for the wet-bulb thermometer at temperatures below freezing no longer serves its usual purpose, for ice, unlike water, does not respond to capillary forces. True wet-bulb readings are obtained only when the surface of the wick is completely covered with a layer of ice. The ice held within the wick is useless.

As a result of the reduced vapor pressure at low temperatures, a longer time is necessary to reach equilibrium than at higher temperatures. This condition is offset, however, by the ice remaining on the bulb for a much longer period of time. Readings of the wet-bulb thermometer must be continued over a sufficiently long period to insure that equilibrium has been reached.

Careful tests indicate that reliable results are obtained by discarding the wick and freezing a layer of ice directly on the thermometer bulb. An ice film 0.02 in. thick has been found to have a life of roughly one hour when exposed to an air velocity of 900 fpm with a saturation deficiency of about 1.4 gr of moisture. The ice film is best formed by dipping the chilled thermometer into distilled water at approximately 32 F. The thermometer is then removed from the water and the film allowed to freeze. The process may be repeated several times if necessary to build up a suitable film thickness.

There is evidence[6] that the mechanical expansion of the ice in freezing might set up stresses in certain types of thermometers to cause an error. One method of overcoming this condition is to build a tube the same size as the thermometer bulb, freeze a coating of ice over it, thaw it with warm water inside the tube, and place the cup of ice over the thermometer.

In order to prevent excessive conduction of heat along the glass stem of the thermometer to the bulb, it is important that the ice film cover about one inch of the stem as well as the bulb.

3.2 —Super-cooled Water

The tendency of water to resist freezing at temperatures below freezing is well known. This condition of super-cooled water can and does exist on the wick of the wet-bulb thermometer and may give rise to considerable error. While it is possible by the aid of a special psychrometric chart to obtain correct readings of psychrometric properties with the wick wetted with super-cooled water, it is generally more desirable to coat the bulb with ice.

3.3—Effect of Air Stream Velocity

In practice the wet-bulb thermometer does not always read the true thermodynamic wet-bulb temperature and under certain conditions may deviate considerably from it. The relative rates of diffusion of moisture from the wetted surface as compared to the diffusion of the surrounding air to the wetted surface tends to make the actual wet-bulb temperature lower than the thermodynamic wet-bulb temperature. Heat received by radiation from surrounding objects at dry-bulb temperature tends to make the wet-bulb thermometer read high. Fortunately, these two effects tend to balance each other, with the result that the wet-bulb thermometer may correspond to the thermodynamic wet-bulb

temperature at certain air velocities, and will generally deviate from the theoretical values less than if either one of these effects were present without the other.

Fig. 10 shows the error in the wet-bulb depression for various air velocities. This curve may be used to correct the reading of wet-bulb thermometers where it is impractical to maintain a suitably high velocity. It should be noted that for temperatures in the comfort air conditioning range, a velocity of approximately 1,000 fpm is desirable for minimum error. At lower temperatures, a considerably lower velocity is indicated.

Note that the curves shown in Fig. 10 apply to a thermometer bulb of conventional size (slightly under $\frac{1}{4}$ in. diameter). For other diameters the velocity scale will be directly proportional to the diameter.

Radiation from coils and other objects. If the wet-bulb thermometer is located close to cooling or heating surfaces, there may result an error due to radiation from such surfaces. In order to avoid this error, a shield constructed from polished metal should be placed between the thermometer and the radiating surface.

Wet-bulb thermometers located in air ducts should not be shielded from the duct wall if the wall is at or near dry-bulb temperature. If necessary, the outside of the duct should be insulated to avoid a large difference between the duct wall and the inside dry-bulb temperatures.

Radiation from the observer. If the observer enters the refrigerated space, precaution must be taken to avoid radiation from the observer's body to the wet-bulb and dry-bulb thermometers. Furthermore, heat from the observer's body and his exhalation of breath may change the temperature and relative humidity in the vicinity in which the readings are being taken. Reliable results can generally be obtained if the observer keeps in motion while using a sling psychrometer and takes readings quickly while avoiding breathing on the thermometers.

3.4—Use of Thermocouples

Thermocouples are entirely satisfactory for taking psychrometric readings if suitable equipment is available and proper precautions are taken in its use. The wet- and dry-bulb thermocouples may be connected

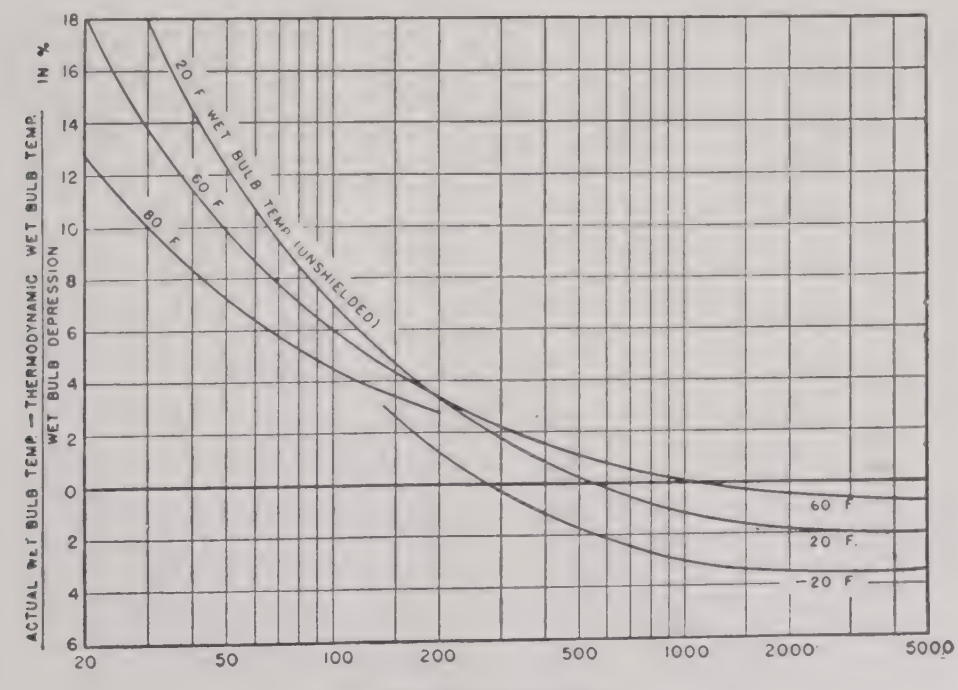


Fig. 10. The Effect of Air Stream Velocity on the Wet-Bulb Psychrometer (For conventional size thermometer bulb)

in series to give a direct reading of the depression and, when measured on a potentiometer of suitable accuracy, such a series couple forms one of the most reliable methods of obtaining psychrometric data.

Thermocouple wires most commonly used in refrigeration practice are copper and constantan or iron and constantan. The author prefers copper-constantan for freedom from rust. Copper has a high thermal conductivity and may introduce errors due to the conduction of heat from the surrounding air, at dry-bulb temperature, to the lower wet-bulb temperature. This condition is best remedied by selecting the copper wire about twelve sizes smaller than the constantan. For example, No. 36 copper wire should be selected for use with No. 24 constantan.

The wick must fit snugly over the thermocouple wire. Small size cotton tubing can be obtained for this purpose. For No. 24 constantan and No. 36 copper wire, the wick should cover about one inch of the lead wires beyond the junction, care being taken to insulate the leads from each other to prevent short circuiting. With larger size wire, a much longer length of lead must be included within the wick. At temperatures below freezing, the wick may be discarded and ice frozen directly on the thermocouples.

The small size of the thermocouple makes it particularly suited for psychrometric determinations at low air stream velocities since as previously stated the velocity scale of Fig. 10 is proportional to diameters. Thus a wet-bulb thermocouple having a diameter of 1/16 in. will have the same correction at 100 fpm as the $\frac{1}{4}$-in. thermometer has at 400 fpm.

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5. HEAT TRANSMISSION

Introduction

1. Heat always tends to flow from a source of high temperature to a substance at a lower temperature. Heat may be transmitted from one temperature level to a lower temperature level by three different methods—conduction, convection and radiation. **Conduction** denotes the flow of heat from one part of a body to another part of the same body or to another body in contact, without displacement of the particles. In conduction, heat is transmitted by molecular motion. The molecules in the material where heat is applied are made to vibrate more rapidly as they are heated, and they transmit their energy as energy of motion to adjoining molecules which in turn are sped up and affect their neighbors, raising their temperatures without any appreciable displacement of material.

Heat transmission through fluids is called **convection.** Actually, there is conduction present when heat is transmitted through fluids, but when the fluid is not prevented from flowing, the transmission is said to be caused by convection. Usually, "free" or "natural" convection is caused by a physical change in the material being heated or cooled. Thus, when water is heated in a pot, the water is first warmed at the bottom and an expansion takes place, making the heated water less dense than the cooler water at the top. The warm water flows upward and the cooler water flows downward, causing "convection currents." This mixing action serves to carry the heat to all portions of the fluid. If the fluid is restricted or the heat is applied in such a manner that free flow does not result, then heat is transmitted only by conduction. If a test tube of water is held over a flame so that only the upper part is heated, there are no forces set up to cause the fluid to flow, so the bottom of the test tube will stay cool until actual conduction has heated it. When the fluid motion is caused by an external force, a fan or pump, for example, the heat interchange is called "forced" convection.

In a sense convection really permits conduction by removing the heated fluid, allowing heat to flow to an unheated portion of the fluid. At the boundary layer at the edge of the fluid, frictional and viscous forces will tend to hold a relatively small layer of the fluid at rest or at a very low velocity. This boundary layer is known as a "fluid film," and heat is transmitted through this film to the main body of the fluid essentially by conduction.

A special case of heat transmission by convection occurs when evaporation or condensation takes place. In a closed container, where vapor and liquid exist in equilibrium, all parts tend to remain at a constant temperature, for if a given part becomes cooler, its local vapor pressure will be reduced. Surrounding vapor pressure will then force more vapor to the cool spot and the vapor will condense, releasing its latent heat and tending to warm the spot. Refrigerant condensers and pressure-cookers provide examples of this form of transmission.

The transmission of heat by **radiation** involves the direct transfer of energy from one body to another through space, at any distance and without the need for intervening matter. It is an electromagnetic wave phenomenon similar to light, but of longer wave length.

All bodies give off heat in the form of radiant energy, and when a system of bodies at different temperatures exists, each body will radiate and absorb heat, in amounts depending on the composition of the bodies and their surface conditions and temperatures, until eventually an equilibrium will exist, when all bodies are at the same temperature. When equilibrium has been attained, the radiation continues, but then the radiation and absorption are equal, so there is no further temperature change.

Nomenclature

2. The following **nomenclature** will be used throughout this chapter:

A = area perpendicular to the direction of heat flow, sq ft
c = specific heat of fluid, Btu/(lb) (deg F); c_p at constant pressure
C = conductance, Btu/(hr) (deg F)
D = diameter, ft; D_e equivalent diameter
d = prefix indicating differential; dimensionless
e = base of natural logarithm = 2.718.
G = mass velocity, lb/(hr) (sq ft of cross section)
g = acceleration due to gravity, 4.17×10^8 ft/(hr) (hr)
h = heat transfer coefficient, Btu/(hr) (sq ft) (deg F)
k = thermal conductivity, Btu/(hr) (sq ft) (deg F per ft)
L = length, ft
$\log_e$ = natural logarithm = $2.303 \log_{10}$
Q = quantity of heat, Btu
q = rate of heat flow, Btu/hr; q_c by convection and conduction, q_r by radiation
R = thermal resistance (deg F) (hr)/Btu
t = temperature, deg F
T = absolute temperature, deg F abs = $460 + t$
U = overall coefficient of heat transfer Btu/(hr) (sq ft) (deg F)
V = velocity, ft per hr = G/ρ
x = length of conduction path, ft

Greek

β (Beta) = coefficient of volumetric expansion, for gases = 1/deg F abs
Δ (Delta) = symbol indicating difference; dimensionless
ϵ (Epsilon) = emissivity; dimensionless
θ (Theta) = time, hr
λ (Lambda) = latent heat of condensation, Btu/lb
μ (Mu) = absolute viscosity, lb/(hr) (ft) = centipoises $\times 2.42$
π (Pi) = 3.1416. . . .
ρ (Rho) = density, lb per cu ft
ϕ (Phi) = prefix, designating function; dimensionless
Σ (Sigma) = prefix indicating a summation

Conduction

3. The basic equation for the conduction of heat states that the instantaneous rate of heat flow equals the product of the area perpendicular to the direction of flow, the temperature gradient, and a constant, k, known as the **thermal conductivity**:

$$\frac{dQ}{d\theta} = -kdA\frac{dt}{dx} \tag{1}$$

where dQ = the amount of heat flowing in time $d\theta$

$\frac{dQ}{d\theta}$ = the instantaneous **rate** of heat flow

dA = area perpendicular to the direction of flow

$\frac{dt}{dx}$ = the temperature gradient or change in temperature, t, per unit length, x

For steady state conditions, where the temperature does not vary with time, and for a body of uniform cross-sectional area, the steady rate of heat flow, q, is expressed

$$q = \frac{kA\Delta t}{x} \tag{2}$$

where x = length of path of heat flow, A is the area through which the heat flows, and k is in consistent units.

When heat is flowing radially through the walls of a cylinder, such as the heat flow through pipe insulation, Equation (2) may still be used, if A_m, the logarithmic mean area, is used for A. A_m may be found from

$$A_m = \frac{A_2 - A_1}{\log_e (A_2/A_1)} = \frac{A_2 - A_1}{2.303 \log_{10} (A_2/A_1)} \tag{3}$$

When the outside area is not more than twice the inner area ($A_2/A_1 = 2$ or less), the arithmetic mean, $(A_2 + A_1)/2$, will be within 4% of the logarithmic mean and may be used without serious error in most engineering work.

The units used for the thermal conductivity, k, frequently cause confusion. Strictly, in the English system k is expressed as Btu per hr per sq ft per unit temperature gradient, deg F per ft of length of path, but in the refrigeration, air conditioning and building professions the value is frequently expressed per **inch** of length of path. However, for consistency, in this chapter we will use

$$k = \text{Btu/(hr) (sq ft) (deg F per ft)} \tag{4}$$

Care must be used when selecting values of

k from a table to note the units used, since some tables are based on a foot of thickness and others are based upon an inch.

The values of k vary greatly between different materials, being greatest in metals and lowest in gases and insulators, with the values in liquids falling between. Thus, k for copper is about 220, for water about 0.35, for air about 0.015, and for cork about 0.025. Further, k will vary in a given material, depending upon the apparent density; thus in asbestos with a density of 3.65 lb per board ft, $k=0.135$, and with a density of 2.44, $k=0.088$. Thermal conductivity also varies with temperature, so mean values should be used in heat transfer problems. For values of k, see Chap. 12, page 229.

Frequently the flow of heat through several bodies in contact with each other is of importance. In this discussion it is convenient to use the term **resistance,** $R = x/kA$. Then Equation (2) may be written

$$q = \frac{\Delta t}{x/kA} = \frac{\Delta t}{R} \tag{5}$$

For steady state flow through a system of bodies in contact, with no losses due to edge effects, it is obvious that the same amount of heat must pass through each body in a given time. Thus

$$q = \frac{\Delta t_1}{R_1} = \frac{\Delta t_2}{R_2} = \frac{\Delta t_3}{R_3} = \cdots \frac{\Delta t_n}{R_n} \tag{6}$$

where R_1, R_2, etc., are the individual resistances of each body, and $\Delta t_1, \Delta t_2$, etc., are their respective temperature differences. From Equation (6), adding $(qR_1 = \Delta t_1)$ $(qR_2 = \Delta t_2)$, etc.

$$q = \frac{\Delta t_1 + \Delta t_2 + \Delta t_3 + \cdots \Delta t_n}{R_1 + R_2 + R_3 + \cdots R_n} = \frac{\Sigma(\Delta t)}{R_T} \tag{7}$$

in which R_T is the total series resistance, the sum of all the individual resistances. Then

$$q = \frac{\Sigma(\Delta t)}{\frac{x_1}{k_1 A_1} + \frac{x_2}{k_2 A_2} + \frac{x_3}{k_3 A_3} + \cdots \frac{x_n}{k_n A_n}} \tag{8}$$

In Equation (8), k_1, k_2, etc., should be taken as the mean thermal conductivities, and A_1, A_2, etc., as mean areas perpendicular to the direction of flow. Thus in cylinders, the logarithmic mean areas (Equation 3) should be used for each individual material.

Conductance is a term used to indicate the overall transfer of heat, when conduction, convection and radiation all may be present. It is defined by the equation

$$C = \Sigma q/\Delta t \tag{9}$$

where C = conductance, Σq is the total heat transmission rate, and Δt the total temperature drop. Conductance of a unit area, or unit conductance, C' equals C/A,

$$C' = \Sigma q/A\Delta t \tag{10}$$

When conduction alone is the method of heat transmission, $q = kA\Delta t/x$ (Equation 2), conductance becomes kA/x, the reciprocal of resistance.

When conditions are not steady, as in intermittent heating and cooling, the conduction problem becomes complex. Since such factors as shape, volume, specific heat, density, and thermal conductivity are involved, generalization is difficult. For a numerical method of solution of these problems see Dusinberre, "Numerical Methods for Transient Heat Flow," Trans. ASME, November, 1945, pp. 703–712.

Convection

4. When heat is applied to a fluid, the heat may be transferred not only by conduction, but also by the mixing of the heated portion with the rest of the fluid, which process is known broadly as **convection.** There are two distinct types of convection—**free** or **natural** convection, which is the transfer of heat from a surface to a fluid causing the fluid to flow only because of its change in density, and **forced** convection, when the circulation of the fluid is caused by mechanical means, such as a fan or a pump.

In either case, whenever there is a fluid in contact with a surface, there is a **fluid film,** which is stationary at the surface and moves only at low velocity near the surface. The study of convection is largely a study of the transfer of heat through this film.

It is assumed that all the resistance to heat transfer in a fluid lies in the film, and that once the heat passes through the film turbulence will distribute it throughout the rest of the fluid. It is conduction through a

film of indeterminate thickness that produces convection. So, in Equation (2), if the term k/x be replaced by h, the **film coefficient,** we have

$$q = hA\Delta t \tag{11}$$

Free convection. A general equation for film coefficients for free convection, first proposed by Lorenz in 1881, is reasonably accurate for vertical planes:

$$h_c = \frac{0.55k}{L}\left[\left(\frac{c_p\mu}{k}\right)\left(\frac{L^3\rho^2\beta g\Delta t}{\mu^2}\right)\right]^{1/4} \tag{12}$$

where symbols are as noted under nomenclature (page 76) and β, the coefficient of expansion, is $1/T$ for ideal gases. For horizontal cylinders, the outside diameter D is used in place of L.

For surfaces where L is greater than about 2 ft, the effect of L is not appreciable, and

$$h_c = 0.13k\left[\left(\frac{c_p\mu}{k}\right)\left(\rho^2\beta g\Delta t\right)\right]^{1/4} \tag{13}$$

For free convection in air, McAdams[5] recommends the following simplified equations:

Horizontal plates, facing upward:

$$h_c = 0.38\ (\Delta t)^{0.25} \tag{14}$$

Horizontal plates, facing downward:

$$h_c = 0.2\ (\Delta t)^{0.25} \tag{15}$$

Vertical plates more than 1 ft high:

$$h_c = 0.27\ (\Delta t)^{0.25} \tag{16}$$

Vertical plates less than 1 ft high:

$$h_c = 0.28\ (\Delta t/L)^{0.25} \tag{17}$$

Vertical pipes more than 1 ft high, and also horizontal pipes:

$$h_c = 0.27\ (\Delta t/D_0)^{0.25} \tag{18}$$

In the above, h_c is the film coefficient for convection. Radiation may account for a substantial amount of the total heat transferred, so the total heat would be

$$q = h_cA(t_s - t_a) + h_rA(t_s - t_w) \tag{19}$$

where h_r is the coefficient of heat transfer by radiation (see Equation (49), page 88), and t_s, t_a, and t_w are the temperatures of the surface in question, the air, and the surrounding walls, respectively.

Forced convection. A general equation for the film coefficient for velocities greater than the critical was developed by Nusselt, by dimensional analysis:

$$\frac{hD}{k} = \phi\left(\frac{DG}{\mu}\right)\psi\left(\frac{c_p\mu}{k}\right) \tag{20}$$

in which the dimensionless quantities DG/μ and $c_p\mu/k$ are known as Reynolds number and Prandtl's number, respectively. The functions ϕ and ψ are determined experimentally, and G is the mass velocity, pounds of fluid per hour per square foot of cross section. Although there is no abrupt transition from free to forced convection, free convection usually occurs with Reynolds numbers below 2,100, and forced convection usually produces Reynolds numbers over 3,000.

For heating fluids, whose viscosities are not more than twice that of water, with Reynolds numbers above 2,100, McAdams[4] gives

$$\frac{hD}{k} = .023\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{0.4} \tag{21}$$

which can also be used for cooling, with 0.3 instead of 0.4 as the exponent for $c_p\mu/k$. For high viscosity liquids and Reynolds numbers over 10,000, Sieder & Tate[10] give

$$\frac{hD}{k} = 0.027\left(\frac{\mu}{\mu_s}\right)^{0.14}\left(\frac{DG}{\mu}\right)^{0.8}\left(\frac{c_p\mu}{k}\right)^{1/3} \tag{22}$$

where μ_s is the viscosity at the surface temperature and μ is the viscosity at the bulk temperature.

For diatomic gases, including air, for which $c_p\mu/k = 0.74$, Equation (21) becomes

$$hD/k = 0.02\ (DG/\mu)^{0.8} \tag{23}$$

For heating or cooling viscous liquids with velocities below the critical (Reynolds number less than 2,100), in horizontal or vertical tubes, the recommended equation is

$$\frac{hD}{k}\left(\frac{\mu_s}{\mu}\right)^{0.14} = 1.86\left[\left(\frac{DG}{\mu}\right)\left(\frac{c_p\mu}{k}\right)\left(\frac{D}{L}\right)\right]^{1/3} \tag{24}$$

where μ is the viscosity at the bulk temperature, and μ_s the viscosity corresponding to the average inside surface temperature.

For oils in turbulent flow an approximate equation for heating is

$$h = 70V/Z^{0.63} \qquad (25)$$

where V is in fps and Z is in centipoises. For cooling oils, reduce h from Equation (25) by 25%.

For gases heated or cooled outside tube bundles, flowing parallel to the tubes, Equation (21) may be used by substituting for D an equivalent diameter equal to four times the hydraulic radius. The hydraulic radius is obtained by dividing the free area between the tubes by the sum of the perimeters of all the tubes.

For annular spaces, use Equation (21), substituting for D an equivalent diameter equal to the difference between the outer and inner diameters of the annular space.

For gases flowing at right angles to single cylinders, McAdams[5] recommends, on the basis of data for both wires and pipes, for any gas, for D_0G/μ_f from 0.1 to 1,000:

$$\frac{hD_0/k_f}{(c_p\mu_f/k_f)^{0.3}} = 0.35 + 0.47\left(\frac{D_0G}{\mu_f}\right)^{0.52} \qquad (26)$$

and when D_0G/μ_f has values from 1,000 to 50,000

$$\frac{hD_0/k_f}{(c_p\mu_f/k_f)^{0.3}} = 0.26\left(\frac{D_0G}{\mu_f}\right)^{0.6} \qquad (27)$$

which, for air and other diatomic gases, with Prandtl number = 0.74, becomes

$$hD_0/k_f = 0.24\ (D_0G/\mu_f)^{0.6} \qquad (28)$$

where k_f and μ_f are at film temperature, which is usually taken as a mean between surface and bulk temperatures, and D_0 is the outside diameter.

The same reference recommends

$$\frac{hD_0}{k_f} = 0.33\left(\frac{c_p\mu_f}{k_f}\right)^{1/3}\left(\frac{D_0G_{max}}{\mu_f}\right)^{0.6} \qquad (29)$$

for both liquids and gases flowing normally to banks of staggered tubes, for D_0G_{max}/μ over 2,000, where G_{max} is the mass velocity through the minimum free area between the tubes. For banks in line (not staggered), Equation (29) should be used, with the constant 0.33 reduced to 0.26.

Figs. 1 through **4** give values for film coefficients for water and Freon-12, both inside and outside, single tubes.

Condensing vapors. When a single vapor condenses, the condensate normally wets the tube, and film type condensation occurs. For condensation outside **horizontal** tubes,[6]

$$h = 0.73\left(\frac{k_f^3\rho_f^2\lambda g}{D\mu_f N\Delta t_m}\right)^{1/4} \qquad (30)$$

where Δt_m is the mean temperature difference between the surface temperature and saturation temperature, λ the latent heat of condensation, and N is the number of rows in a vertical plane.

For **vertical** tubes

$$h = 1.13\left(\frac{k_f^3\rho_f^2\lambda g}{L\mu_f\Delta t}\right)^{1/4} \qquad (31)$$

The film coefficient for Freon-12 is roughly 200 to 400 in horizontal tubes, and that for ammonia about 1,000. Clean steam will give coefficients ranging from 1,000 to 3,000 for film type condensation, and when dropwise condensation of steam occurs on highly polished or treated surfaces, coefficients from 7,000 to 14,000 have been obtained.

Boiling liquids. When a liquid is boiled, as the temperature difference between the boiling liquid and the surface is increased the rate of boiling (and the amount of heat flowing) increases, but reaches a maximum with a **critical temperature difference,** above which the rate of boiling decreases. As the temperature difference becomes greater than this critical difference, the vapor formed by boiling acts as an insulator, impeding the transfer of heat. The critical difference for water is about 45 F.[9]

In general, the rate of boiling is increased as the surface becomes rougher, and as the liquid being boiled is agitated. It is increased with an increase of temperature difference up to the critical point, and is reduced by scale and dirt deposits.

For boiling, it is usually necessary to measure overall transfer coefficients for the particular liquid and physical conditions in question, as there are few published data available.

Overall Transmission

5. In most cases in refrigeration and air conditioning, heat is transmitted from one fluid through a solid wall to another fluid. It is known that in such transmission, the rate of heat transfer, dq, is directly proportional to the area perpendicular to the di-

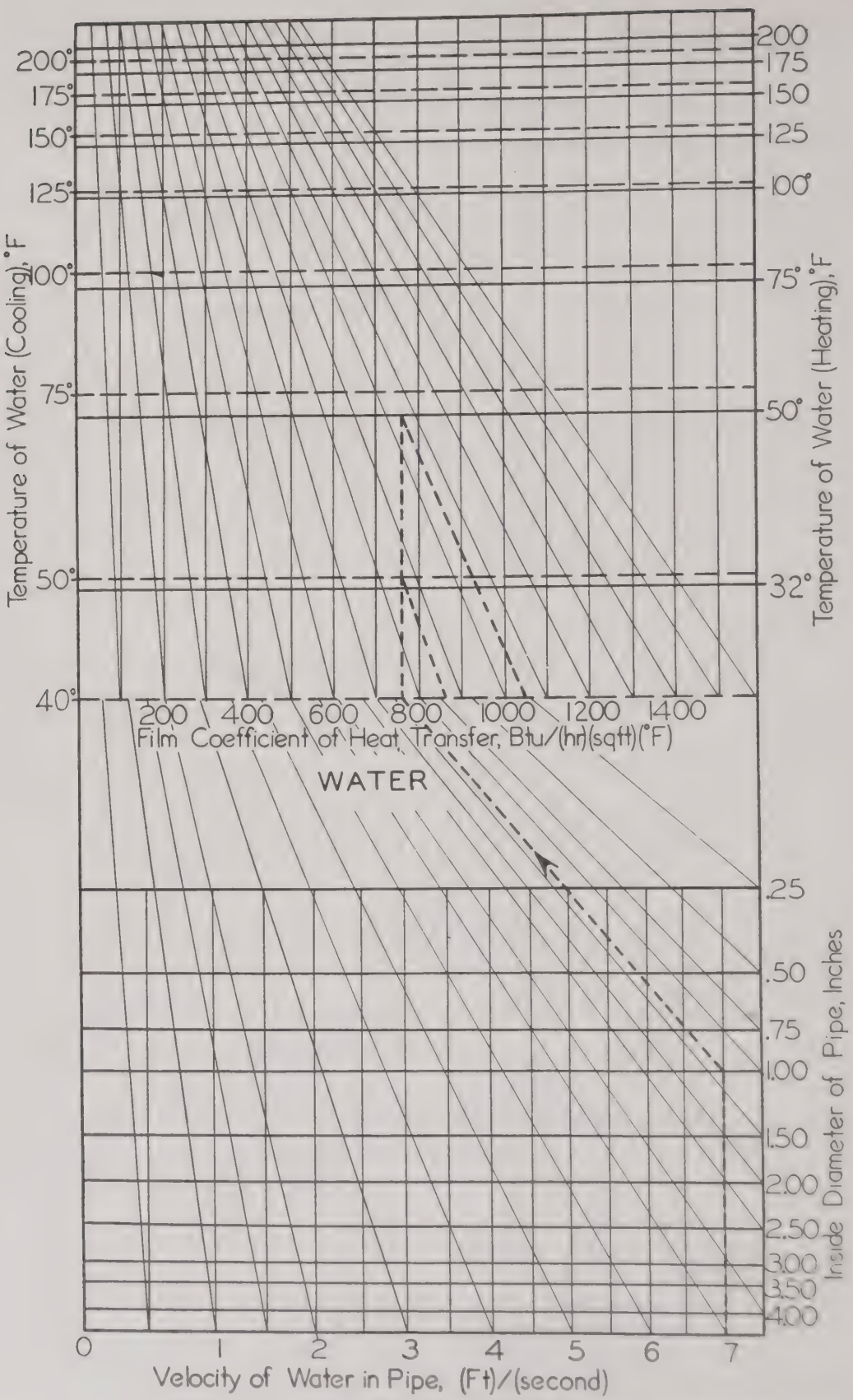

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Fig. 1. Film Coefficient for Water Flowing Inside a Pipe

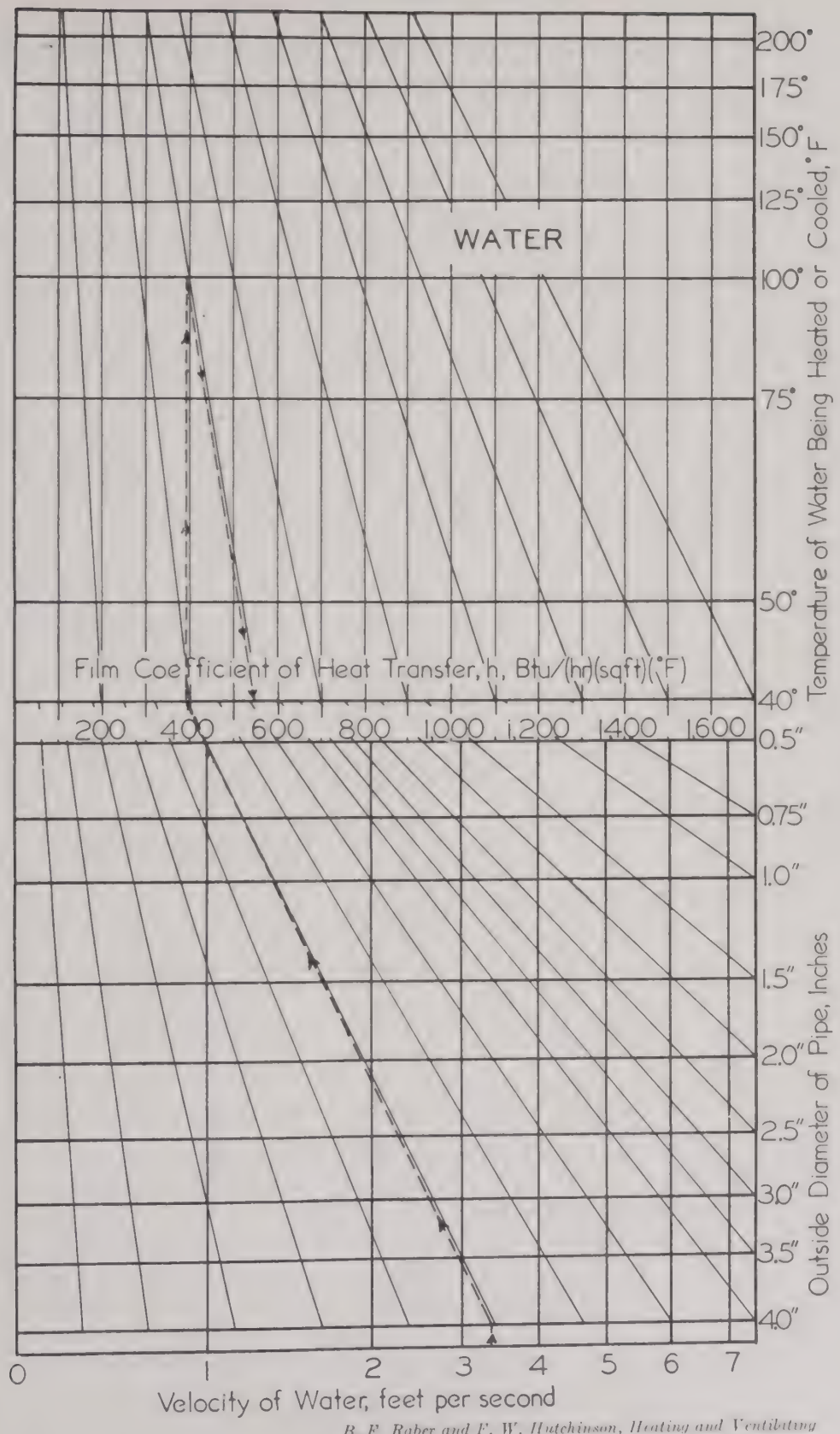

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Fig. 2. Film Coefficient for Water Flowing Outside and Normal to a Single Pipe

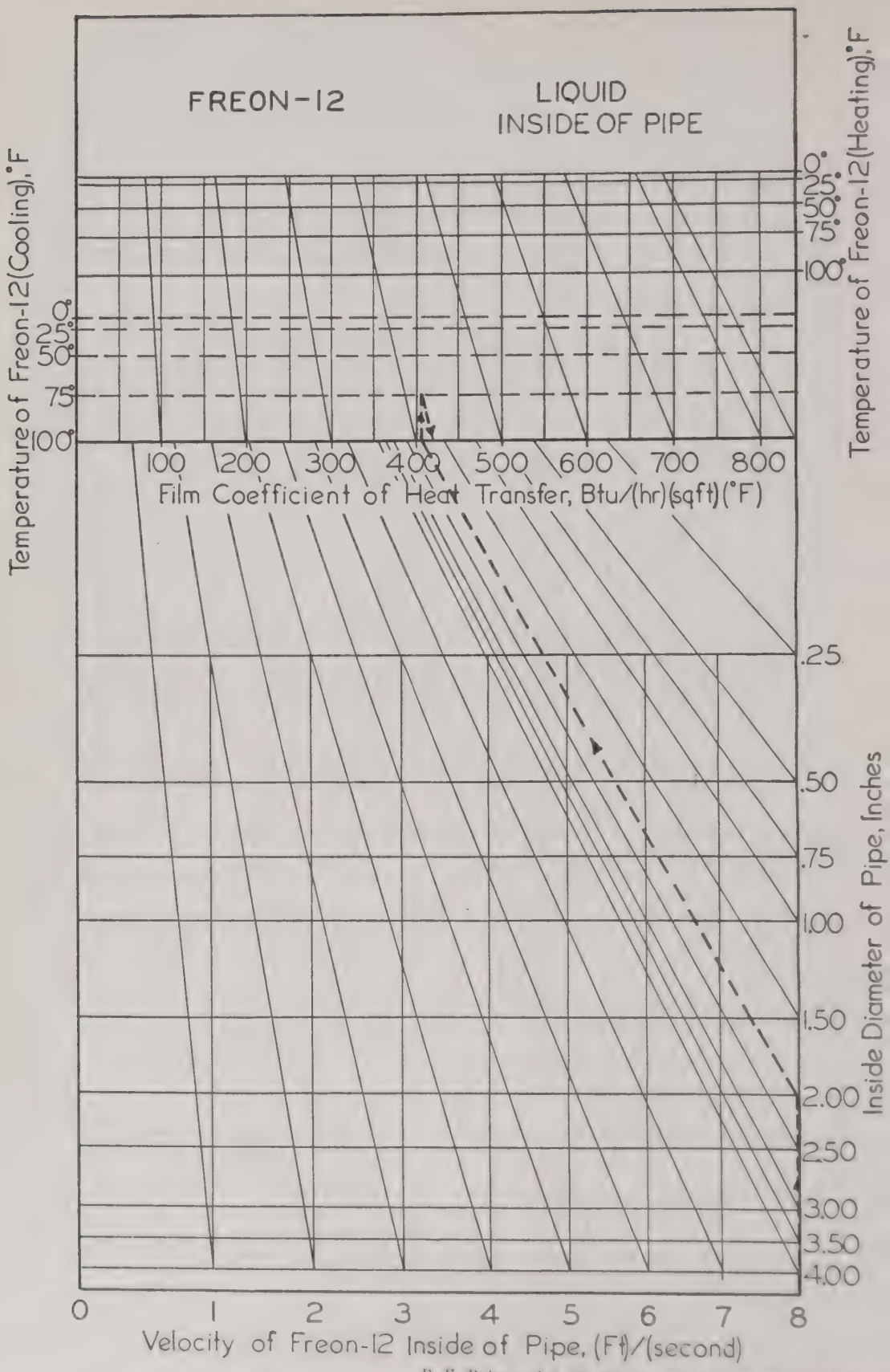

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Fig. 3. Film Coefficient for Liquid Freon-12 Flowing Inside a Pipe

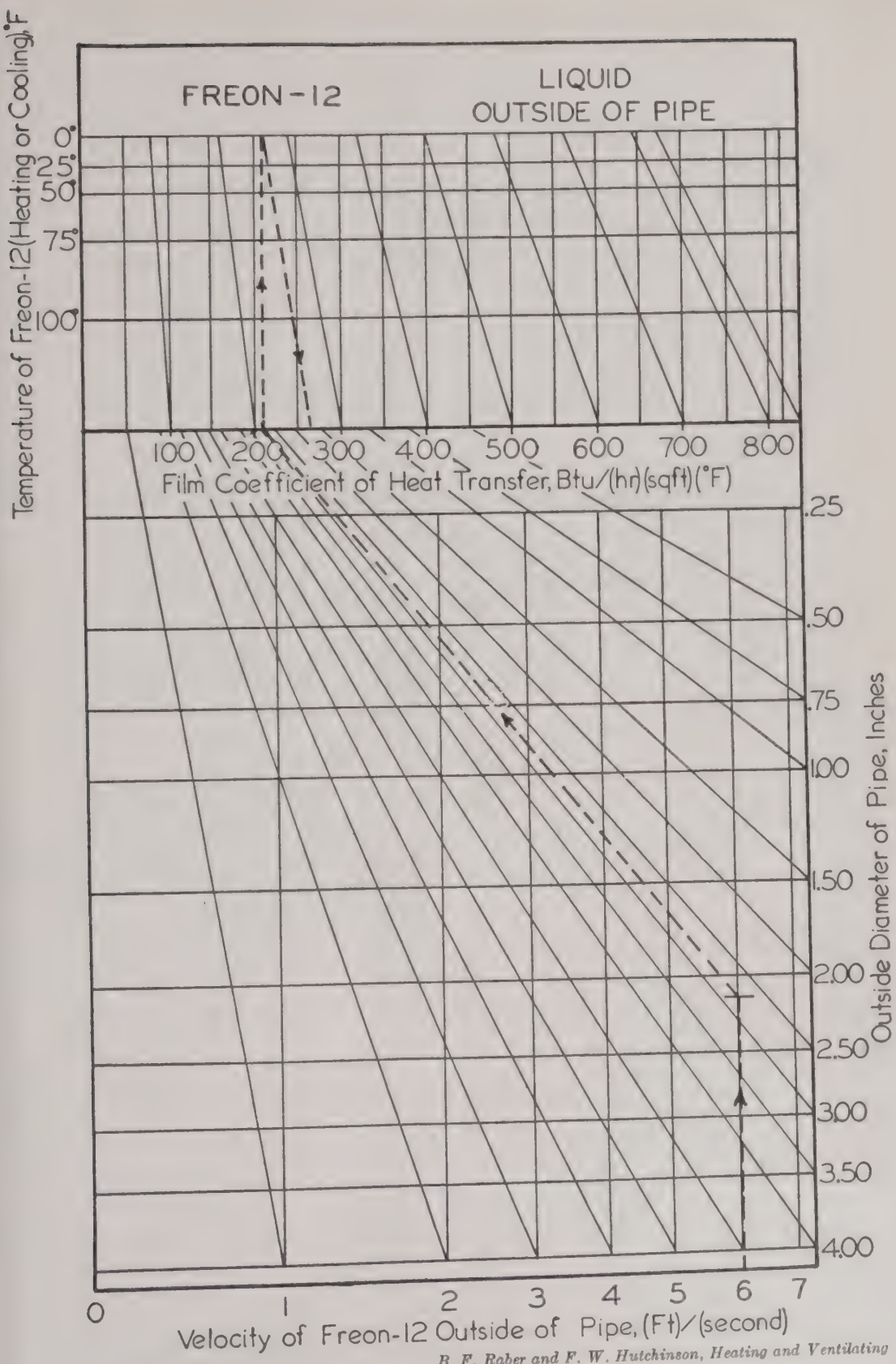

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Fig. 4. Film Coefficient for Liquid Freon-12 Flowing Outside of a Pipe

rection of heat flow, dA, and to the temperature difference existing between the fluids, Δt, or

$$dq = U dA \Delta t \tag{32}$$

where U, the proportionality factor, is the **overall coefficient of heat transfer.**

It is obvious that the value of U will depend upon where the measurement of area, dA, is taken (i.e., in a double-pipe interchanger, whether the inside or outside surface of the dividing wall is used.) Any surface can be used provided that the one selected is defined and used consistently.

Equation (32) represents the overall heat transfer; the flow through any single element of the path is given by an equation of the form

$$dq = h' dA' \Delta t' \tag{33}$$

in which dA' and $\Delta t'$ are the area of and temperature drop through the element being considered, and h' is called the **local individual coefficient of heat transfer.**

In the case of a double-tube interchanger, with fluids flowing through the center tube and in the annular space between the tubes, if one fluid is warmer than the other there will be a transfer of heat from the warm fluid through the wall to the cooler fluid. Then, from Equation (2) for a wall thickness x_w with a logarithmic mean area dA_w, the rate of conduction through the wall is

$$dq = \frac{k_w dA_w \Delta t_w}{x_w} \tag{34}$$

Frequently there will be scale or dirt adhering to one of the surfaces of the wall. The same amount of heat passing through the wall, dq, must then also pass through the scale deposit, which has a thickness of x_s, a thermal conductivity k_s, and which will be assumed to be on the inside wall, whose area is dA_i. Since scale deposits are usually thin, the area of the wall is used rather than the log mean area of the scale. So, for the scale,

$$dq = \frac{k_s dA_i \Delta t_s}{x_s} = h_s dA_i \Delta t_s \tag{35}$$

in which h_s is used for k_s/x_s. (Heat transfer coefficients for scale deposits may be obtained from **Table 1.**[11])

In addition to Equations (34) and (35), we may write equations for the boundary films between the outer fluid and the wall (local coefficient h_0, temperature drop Δt_0), and between the scale and the inner fluid (local coefficient h_i, temperature difference Δt_i).

We have then, for the individual temperature drops:

$$\Delta t_0 = dq/h_0 dA_0$$
$$\Delta t_w = dq\ x_w/k_w dA_w$$
$$\Delta t_s = dq/h_s dA_i$$
$$\Delta t_i = dq/h_i dA_i$$

Adding these four equations gives us the overall temperature difference, Δt:

$$\Delta t = dq \left(\frac{1}{h_0 dA_0} + \frac{x_w}{k_w dA_w} + \frac{1}{h_s dA_i} + \frac{1}{h_i dA_i} \right) \tag{36}$$

From Equation (32), $\Delta t/dq = 1/U\ dA$, so Equation (36) may be used to find U where dA is the area for which U is desired:

$$\frac{1}{U} = \frac{dA}{h_0 dA_0} + \frac{x_w dA}{k_w dA_w} + \frac{dA}{h_s dA_i} + \frac{dA}{h_i dA_i} \tag{37}$$

The most common use of this equation is the case where the wall thickness is small compared to the tube diameter, when all areas (dA_i, dA_0, etc.) are very nearly equal. Then

$$\frac{1}{U} = \frac{1}{h_0} + \frac{x_w}{k_w} + \frac{1}{h_s} + \frac{1}{h_i} \tag{38}$$

If there were scale deposits both inside and outside, and if the tube were made of two materials bonded together, this would become

$$\frac{1}{U} = \frac{1}{h_0} + \frac{1}{h_{s0}} + \frac{x_{w1}}{h_{w1}} + \frac{x_{w2}}{k_{w2}} + \frac{1}{h_{si}} + \frac{1}{h_i} \tag{39}$$

and since overall resistance R_T for an area dA is $R_T = 1/U$, and correspondingly, $R_0 = dA/h_0 dA_0$, etc., Equation (38) becomes

$$R_T = R_0 + R_w + R_s + R_i \tag{40}$$

which shows that when heat flows through a series of different resistances, the total resistance is the sum of all individual resistances.

Extended surfaces. Refrigerating engineers have made increasing use of all types of extended surfaces. A thorough analysis of various types of surfaces has been made by Gardner,[2] who produced curves shown in **Figs. 5** through **8.** Values

Table 1. Heat-Transfer Coefficients h_s for Scale Deposits from Water[1]

Temperature of heating medium	Up to 240 F		240–400 F	
Temperature of water	125 F or less		Above 125 F	
Water velocity, fps	3 and less	Over 3	3 and less	Over 3
Distilled	2,000	2,000	2,000	2,000
Sea water	2,000	2,000	1,000	1,000
Treated boiler feed water	1,000	2,000	500	1,000
Treated make-up for cooling tower	1,000	1,000	500	500
City, well, Great Lakes	1,000	1,000	500	500
Brackish, clean river water	500	1,000	330	500
River water, muddy, silty[2]	330	500	250	330
Hard (more than 15 g per gal)	330	330	200	200
Chicago Sanitary Canal	130	170	100	130

[1] From "Standards of Tubular Exchanger Manufacturers Association," 366 Madison Ave., New York, N. Y., 1941.
[2] Delaware, East River (New York), Mississippi, Schuylkill, and New York Bay.

of fin efficiency, ϕ, are given for the types of fins shown on each of the curves. However, it is necessary to convert fin efficiency into the actual total resistance of the fin and tube, R_m. Carrier and Anderson[1] give the following formula for obtaining the metal thermal resistance, R_m, in Btu/(hr) (sq ft of total outside surface) (deg F).

$$R_m = \frac{1}{h_a}\left[\frac{1-\phi}{\phi+\dfrac{S_p}{S_f}}\right] \tag{41}$$

where h_a is the outside air film coefficient, S_p is the outside exposed tube surface, sq ft per lineal ft of tube, and S_f is the fin surface, sq ft per lineal ft of tube. Carrier and Anderson[1] also point out that a reasonably close approximation of the efficiency of a square or rectangular plate fin is obtained by assuming a flat circular fin of equal area.

It is necessary, in using the familiar equation

$$q = UA\Delta t_m \tag{42}$$

to understand its limitations. It assumes steady state heat flow, constant specific heats, and adiabatic operation in parallel or counter-flow conditions, with U remaining constant. The mean temperature difference, Δt_m, is properly the logarithmic mean temperature difference (abbreviated log MTD, MED, LMTD, etc.) which is

$$\Delta t_m = \frac{\Delta t_1 - \Delta t_2}{\log_e \dfrac{\Delta t_1}{\Delta t_2}} = \frac{\Delta t_1 - \Delta t_2}{2.303 \log_{10} \dfrac{\Delta t_1}{\Delta t_2}} \tag{43}$$

When U varies with temperature we have the equation by Nikuradse,[7]

$$q = A\left[\frac{U_2\Delta t_1 - U_1\Delta t_2}{\log_e \dfrac{U_2\Delta t_1}{U_1\Delta t_2}}\right] \tag{44}$$

which, when $U_1 = U_2$, becomes equal to Equation (42).

Curves for determining mean temperature differences for reversed current and cross-flow exchangers of different types are given by Bowman, Mueller and Nagle, in Trans. ASME, vol. 62, pp. 283–294, 1940.

Radiation

6. The transmission of heat by radiation is a wave phenomenon, as is the transmission of light; the essential difference between the two is that the wave lengths involved in the transmission of heat are longer than those in light transmission. As the temperature of a body increases the proportionate amount of energy emitted as light increases, but even an incandescent lamp filament radiates only a few per cent of its total energy as light, the greatest portion of the energy being radiated as heat.

Materials differ in their ability to absorb and emit radiation, so a "black body" is used as a reference standard. Defined as a body which absorbs all radiation which hits it, reflecting none, it is said to have an **absorptivity** of one and a **reflectivity** of zero. For any body the ab-

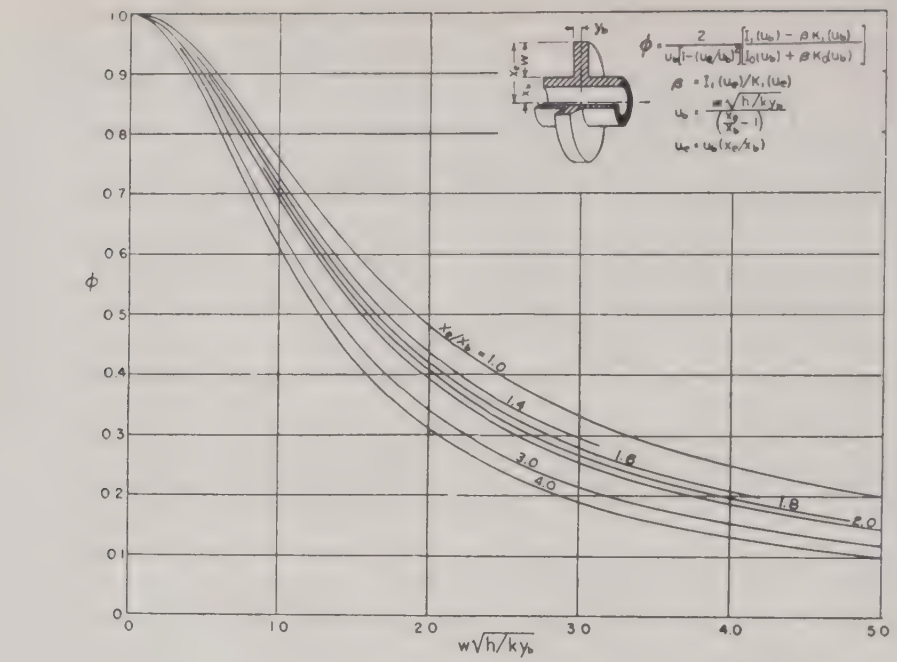

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Fig. 5. Efficiency of Annular Fins of Constant Thickness

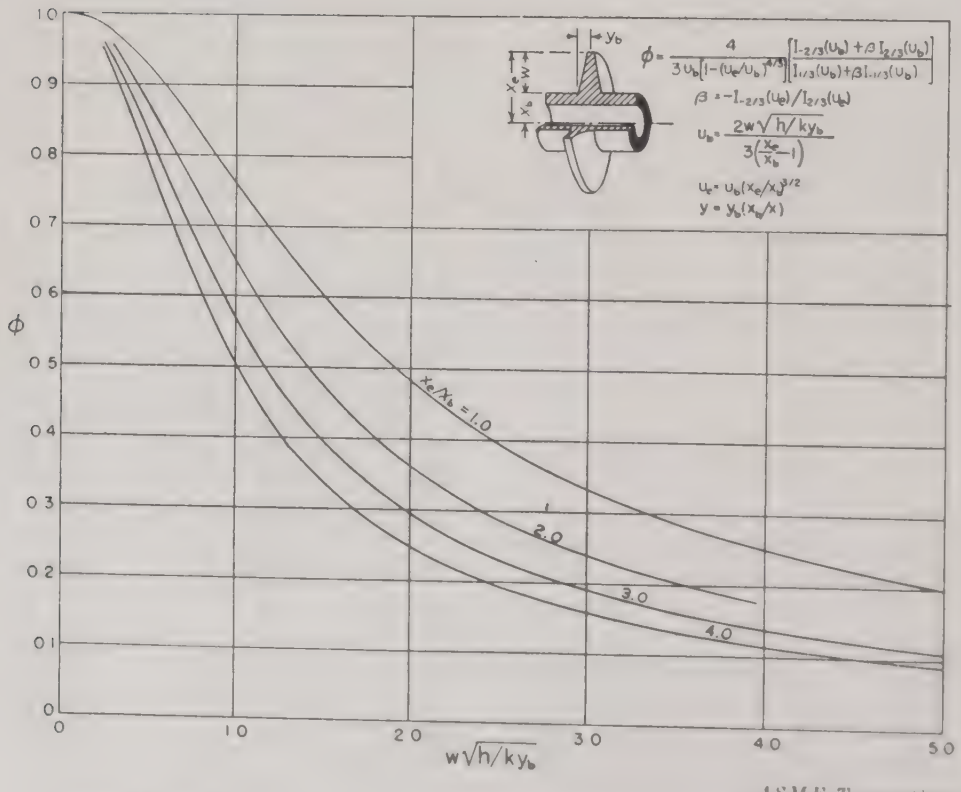

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Fig. 6. Efficiency of Annular Fins with Constant Metal Area for Heat Flow

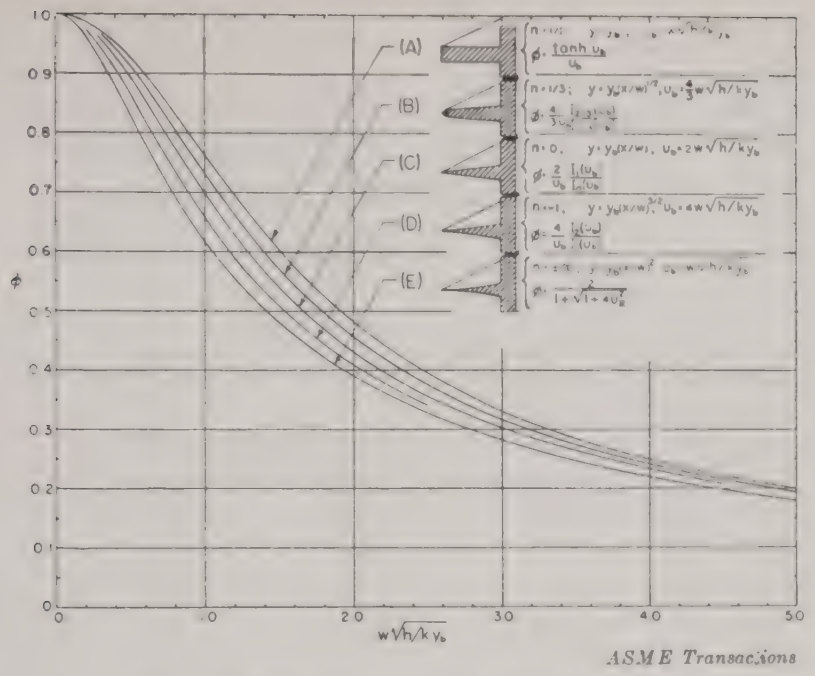

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Fig. 7. Fin Efficiency of Several Types of Straight Fin

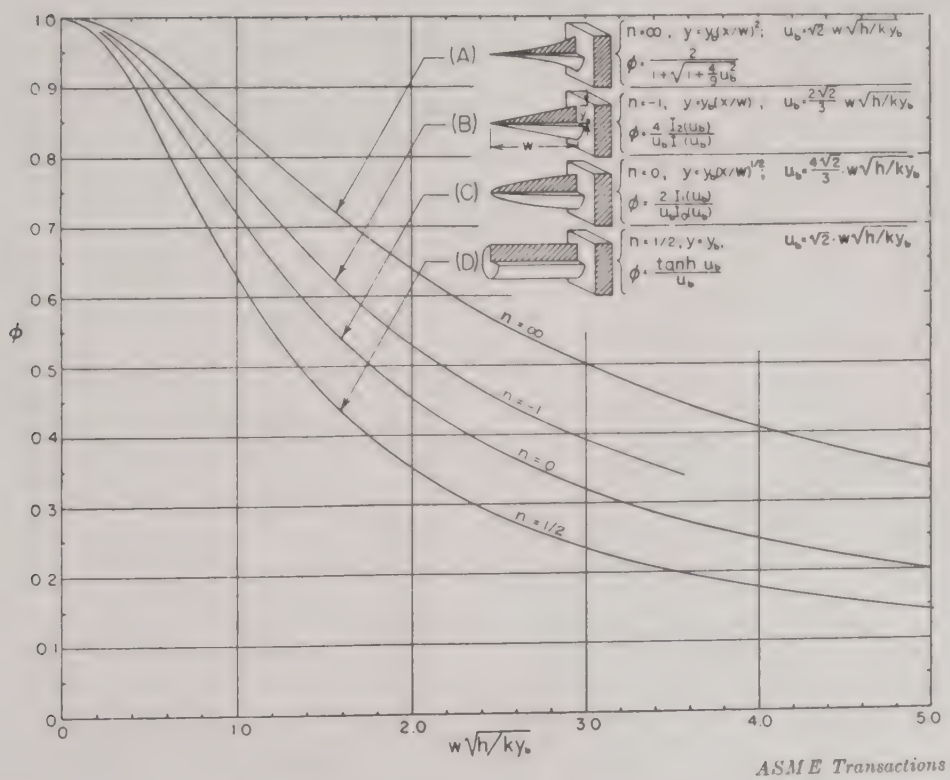

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Fig. 8. Efficiency Curves for Four Types of Spine

sorptivity is the fraction of impinging radiation absorbed, and the reflectivity is the fraction reflected. Neither can, of course, exceed one. The **emissivity,** ϵ, is the ratio of the radiating power to the radiating power of a black body at the same temperature. Kirchhoff's law shows that the emissivity of a body is the same as its absorptivity; thus a good radiator is a good absorber, and a good reflector is a poor radiator.

It was first proposed by Stefan in 1879 that the radiation from any body is proportional to the fourth power of its absolute temperature, and in 1894 Boltzmann proved Stefan's assumption for black bodies. The **Stefan-Boltzmann law** is

$$q_B = 17.23 \times 10^{-10} A_B T_B^4 \qquad (45)$$

where q_B is the total energy emitted from a black body, and A_B and T_B are the surface areas and absolute temperatures of the black body.

There are no perfect black bodies; actual surfaces usually absorb part of the impinging radiation and reflect some. If a surface reflects a constant proportion of the radiation at all wave lengths at a given temperature, the body is said to be **grey.**

Any body in a system is continually receiving as well as emitting radiation. The net change in energy is the difference between the heat gained and that lost. For two parallel planes of infinite extent, at absolute temperatures of T_1 and T_2, and separated by a vacuum, the heat lost by the hot surface is

$$q = \frac{1723A}{\frac{1}{\epsilon_1} + \frac{1}{\epsilon_2} - 1}\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right] \qquad (46)$$

where the subscripts 1 and 2 refer to the hot and cold surfaces respectively, ϵ_1 and ϵ_2 their emissivities, and T_1 and T_2 their absolute temperatures. This deduction from the Stefan-Boltzmann law also applies for the case where one body is entirely enclosed by the other, if they are of nearly equal size, in which case the area of the enclosed body should be used for A.

If the enclosed body is very small with respect to the outer body, the inner walls of the outer body act as black bodies irrespective of their surfaces, so $\epsilon_2 = 1$ and Equation (46) becomes

$$q = 1723\,\epsilon_1 A_1\left[\left(\frac{T_1}{1000}\right)^4 - \left(\frac{T_2}{1000}\right)^4\right] \qquad (47)$$

where ϵ_1 and A_1 refer to the enclosed body.

Sometimes it is convenient to use a simplified radiation equation

$$q_r = h_r A_s (t_s - t_e) \qquad (48)$$

in which h_r is defined as follows:

$$h_r = \frac{q_r}{A_s(t_s - t_e)} = \frac{1723\,\epsilon_s\left[(T_s/1000)^4 - (T_e/1000)^4\right]}{T_s - T_e} \qquad (49)$$

Where the subscripts s and e refer to the surface radiating and the enclosure, respectively, values of h_r may be found from **Fig. 9.**

At temperatures below 2,000 F most non-metallic surfaces have emissivities and absorptivities of about 0.90 to 0.95. Oxidized copper or brass will be about 0.60, polished steel 0.20, polished copper 0.10, and polished chromium 0.05.

Glass is almost transparent to solar radiation, but transmission can be reduced by painting the outside of the glass. At temperatures from 0 to 150 F, glass has a high emissivity (and absorptivity), thus making it suitable for thermometers.

At low temperatures, color has little effect on emissivity. White and black paints of the same degree of gloss have approx-

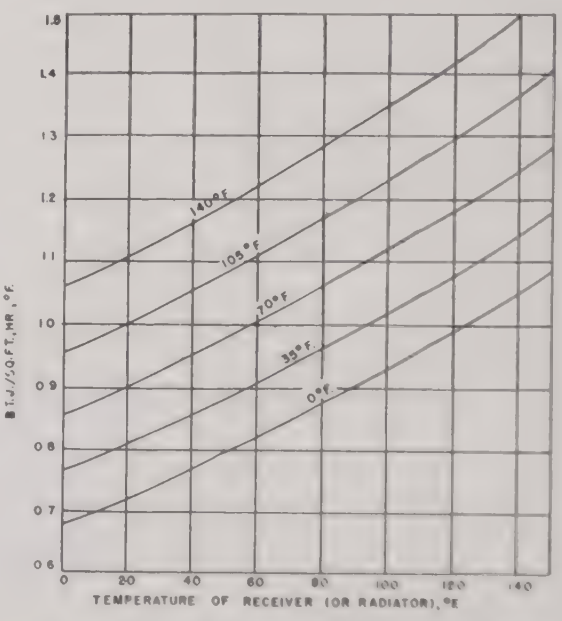

Fig. 9. Values of F_t, Btu per sq ft hr, F

imately equal values, although flat paints have higher emissivities than glossy surfaces. For solar radiation, color is important.[3] White paint absorbs the least, followed in increasing order by yellow, red, green, and black. Metallic paints have higher absorptivities of solar radiation than non-metallic paints (see Table 3, Chap. 12).

Table of Two-Thirds Powers and Three-Halves Powers

(To assist in locating the decimal point)

N	$N^{2/3}(=\sqrt[3]{N^2})$	$N^{3/2}(=\sqrt{N^3})$
.0001	.002154	.000001
.001	.01	.00003162
.01	.0464	.001
.1	.2154	.03162278
1.	1.	1.
10.	4.64	31.62278
100.	21.54	1000.
1000.	100.	31622.78
10000.	464.16	1000000.

Radiant energy is like light in that it travels in all directions from the surfaces of radiating bodies. All the energy radiated by one body is not necessarily received by another, and the surfaces are not always normal to the impinging rays. Thus, the transfer of energy is frequently very complex; only the more simple cases are treated here. For the general case of radiation between surfaces of finite areas, see "Radiant Heat Transmission" by H. C. Hottel, Proc. Third World Power Conference.

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6. THEORY OF FLOW AND VISCOSITY

Friction Loss in Pipes

1. The following symbols are used with the meanings given here:

A = cross sectional area in sq ft
A_i = cross sectional area in sq in.
U = wetted perimeter in ft
U_i = wetted perimeter in in.
D = pipe diameter in ft
d = pipe diameter in in.
m' = hydraulic radius in ft
m = hydraulic radius in in.
l = length of pipe in ft
g = 32.2 fps² gravitational acceleration
P and ΔP = absolute pressure and pressure drop in psf
p and Δp = absolute pressure and pressure drop in psi
$r = P_1/P_2 > 1$ = pressure ratio
$\rho = 1/\zeta = P_2/P_1 < 1$ = ratio of expansion
$\rho' = \gamma/g$ = mass density in slugs/cu ft
$\gamma = 62.4s$ = weight density in lb/cu ft
s = sp gr, relative to water = gr/cu cm
$V = 1/\gamma$ = specific volume in cu ft/lb
$\epsilon = (n-1)/n$ with $n = C_p/C_v$ the adiabatic exponent
v = velocity in fps
$h = v^2/2g$ = velocity head in ft

Flow

α = coefficient of discharge
G = flow in gpm; Q' in cfm; Q'' in cfs
$W' = Q' \cdot \gamma$ = flow in lb/min
W'' = flow in lb/sec

Head loss

ξ = general friction factor (for Darcy's equation)
$f = 4\xi$ friction factor for round pipes (Fanning's equation)
h' = head loss in ft, $\Delta p'$ = head loss in psi
h'_o and $\Delta p'_o$ the same for $l = 100$ ft of pipe
k = resistance factor

Viscosity

η = symbol for absolute viscosity:
η in poises $\quad \eta_c$ in centipoises

$$\eta' \text{ in } \frac{\text{lb sec}}{\text{sq ft}} \qquad \eta'_h \text{ in } \frac{\text{lb hr}}{\text{sq ft}}$$

μ = symbol for dynamic viscosity:

$$\mu' \text{ in } \frac{\text{lb}}{\text{ft sec}} \qquad \mu'_h \text{ in } \frac{\text{lb}}{\text{ft hr}}$$

ν = symbol for kinematic viscosity:
$\nu = \eta/s$ in stokes
ν_c in centistokes
ν' in sq ft/sec
ν'_h in sq ft/hr

Flow and Velocity in Conduits

2. The average velocity in a conduit is defined as

$$v = Q''/A = 144 \cdot (Q''/A_i) \text{ fps}$$

With the flow in cfm or gpm and the pipe diameter in inches, we can write:

$$\begin{aligned} v &= 144 \cdot (Q''/A_i) = 2.4 \cdot (Q'/A_i) \\ &= 0.321 \cdot (G/A_i) \text{ fps} \end{aligned} \tag{1}$$

also

$$\begin{aligned} v &= \frac{(183.35)}{d^2} \cdot Q'' = \frac{(3.056)}{d^2} \cdot Q' \\ &= \frac{(0.4085)}{d^2} \cdot G \text{ fps} \end{aligned} \tag{1a}$$

If the flow is given as W' lb/min, we substitute $Q' = W'/\gamma = W'/62.4\,s$.

Another useful relation is

$$1 \text{ cfs} = 60 \text{ cfm} = 448.8 \text{ gpm}$$
$$1 \text{ gpm} = 500\,s \cdot \text{lb/hr}$$

Basic Energy Equation of Flow; Flow through Orifices and Nozzles

3. The velocity of any fluid is the result of an acceleration process during which a part of the fluid's potential energy (i.e., energy due to pressure) is changed into kinetic energy. The decrement in potential energy, which is the work spent on acceleration, appears as increment in kinetic energy:

$$V \cdot (-dP) = -dP/\gamma = d(v^2/2g) \frac{\text{ft lb}}{\text{lb}} \tag{2}$$

Liquids. In the case of liquids the work spent in accelerating the fluid from $v = 0$ velocity is:

$$V \cdot (P_1 - P_2) = \frac{\Delta P}{\gamma} = \frac{144 \cdot \Delta p}{\gamma} = \left(\frac{v^2}{2g}\right) \frac{\text{ft lb}}{\text{lb}} \tag{3}$$

since for liquids the specific volume $V = 1/\gamma$ remains constant.

The pressure drop along the path (concurrent with the acceleration) is numerically equal to the pressure at the bottom of a real or imaginary column of liquid standing h ft high above the point where the acceleration ends, and having a uniform density equal to that of the liquid at flow condition. This height

$$h = \Delta P/\gamma = \frac{144 \cdot \Delta p}{\gamma} = \left(\frac{v^2}{2g}\right) = \left(\frac{v}{8.025}\right)^2 \text{ ft} \tag{4}$$

which is called the "velocity head," is identical with the height through which a free-falling body must drop to attain the velocity v fps.

Velocity of discharge. From Equation (4) we get

$$v = \sqrt{2gh} = 8.025\sqrt{h} \text{ fps} \qquad v = 96.3\sqrt{\Delta p/\gamma} = 12.19\sqrt{\Delta p/s} \text{ fps} \tag{5}$$

as the theoretical velocity. Because of friction in the orifice or nozzle, the actual velocity is somewhat less.

Volume discharged. Inertia forces in the jet emerging from a plain, sharp-edged orifice cause the jet to contract, until it reaches its maximum velocity and minimum diameter in the so-called "vena contracta," where all the fluid particles of the jet move in parallel; thus the natural contours of this free jet conform to a converging nozzle.

The smallest cross section of the jet is only a fraction of the actual orifice area; the velocity in it is slightly less than the theoretical value. Both these facts are accounted for by applying a so-called coefficient of discharge α, the value of which depends on the quality of the orifice edge and also on the magnitude of the approach-velocity. For a truly sharp orifice with a negligible approach velocity $\alpha = 0.62$. Dulling the edge and increasing the approach velocity increases the value of α.

The jet from a nozzle has attained its minimum cross section inside the nozzle; thus there is no contraction of the jet after leaving the nozzle and the factor α accounts chiefly for the loss in velocity. The value of α is about .97 to .99, depending on the quality of the nozzle.

If the actual area of the orifice or nozzle is A_i sq in., the volume discharged will be

$$Q'' = \frac{\alpha \cdot A_i}{144} \cdot v \text{ cfs} \tag{6}$$

with v in fps as in Equation (5).

In the more commonly used gpm units

$$G = 25 \cdot (\alpha \cdot A_i) \cdot \sqrt{h} \text{ gpm} \qquad G = 300 \cdot (\alpha \cdot A_i) \cdot \sqrt{\Delta p/\gamma} \text{ gpm} \qquad G = 38 \cdot (\alpha \cdot A_i) \cdot \sqrt{\Delta p/s} \text{ gpm} \tag{7}$$

Weight discharged may be expressed as $W'' = Q'' \cdot \gamma$ lb/sec. More explicitly, we can write

$$W' = 3.344\,(\alpha \cdot A_i)\gamma\sqrt{h} \text{ lb/min} \qquad W' = 40.125\,(\alpha \cdot A_i) \cdot \sqrt{\gamma \cdot \Delta p} \text{ lb/min} \qquad W' = 317\,(\alpha \cdot A_i) \cdot \sqrt{s \cdot \Delta p} \text{ lb/min} \tag{8}$$

In the equations given above the value of γ or s must be that existing at the actual conditions of the flow.

Gases and vapors. In the case of gases and vapors, a pressure drop is always accompanied by a corresponding increase of V, the specific volume; therefore the work spent on acceleration is

$$\left(\frac{v^2}{2g}\right) = \int_1^2 V \cdot (-dP) = P_1 \cdot V_1 \left[\frac{1 - \rho^\epsilon}{\epsilon}\right] \text{ ft lb/lb} \tag{9}$$

where $\rho = P_2/P_1$ is the expansion ratio and $\epsilon = (n-1)/n$, with n as the exponent in the $PV^n = \text{constant}$ equation for the adiabatic expansion line.

We should distinguish between three different cases:

If the **pressure drop is small** (less than 5% of the absolute pressure at the start), we can neglect the increase in volume and use the equations for liquids. We find the **velocity of discharge** to be

$$v = 96.3\sqrt{V_1 \Delta p} \text{ fps} \tag{10}$$

which is identical with Equation (5), since $V = 1/\gamma$. (It is more convenient to use V than γ when dealing with gases.)

The **volume discharged** will be

$$Q' = 40.125\,(\alpha \cdot A_i) \cdot \sqrt{V_1 \cdot \Delta p} \text{ cfm} \tag{11}$$

whereas the **weight discharged** will be

$$W' = 40.125\,(\alpha \cdot A_i) \cdot \sqrt{\Delta p/V_1} \text{ lb/min.} \tag{12}$$

Equations (10) and (11) give a smaller

figure, while Equation (12) gives a larger one, than the true value. For a $\Delta p = 5\%$ pressure drop, the error in v and Q' will be about -1.25%, whereas the error in W' will be about $+3.75\%$. However, we can increase the accuracy of these equations substantially by multiplying Equations (10) and (11) by the factor $(1 + 0.25\Delta p/p_1)$, and Equation (12) by the factor $[1 - j \cdot (\Delta p/p_1)]$, where $\Delta p/p_1 = (1 - \rho)$ is the relative pressure drop and j is a factor varying with the value of n, as given in **Table 1.**

By applying these correction factors, we can extend the use of these equations almost as far as the critical pressure drop, with only a negligible error.

For a **medium pressure drop** $(p_2 > p_c)$, we can either use the above equations with the proposed correction factors, or we can determine the correct values by means of the basic Equation (9) with $\rho = p_2/p_1 = (1 - \Delta p/p_1)$, finding the velocity to be

$$v = 96.3\sqrt{p_1 \cdot V_1} \cdot \left[\frac{1-\rho^\epsilon}{\epsilon}\right]^{1/2} \text{ fps} \quad (13)$$

and the flow to be

$$W' = 40.125\ (\alpha \cdot A_i) \cdot \sqrt{p_1/V_1} \times \left[\rho^{1/n}\left(\frac{1-\rho^\epsilon}{\epsilon}\right)^{1/2}\right] \text{ lb/min} \quad (14)$$

For a fixed $p_1 - V_1$ starting condition, the flow W' varies, as does the value

$$b = \rho^{1/n}\left(\frac{1-\rho^\epsilon}{\epsilon}\right)^{1/2}$$

which goes through a maximum at that critical expansion ratio, $\rho = \rho_c$, for which

$$\rho_c^\epsilon = \left(\frac{2}{n+1}\right) \quad (15)$$

and

$$b \text{ max} = \beta = \left(\frac{n}{n+1}\right)^{1/2} \cdot \left(\frac{2}{n+1}\right)^{1/(n-1)} \quad (16)$$

both of which are only a function of n, the exponent for the adiabatic expansion line.

If in Equation (13) we substitute ρ^ϵ as in Equation (15), we find

$$v_c = 96.3\sqrt{\left(\frac{n}{n+1}\right) \cdot p_1 \cdot V_1} \text{ fps} \quad (17)$$

as the velocity for the critical expansion ratio corresponding to the maximum flow

$$W_c' = 40.125\ (\alpha \cdot A_i) \cdot \beta\sqrt{p_1/V_1} \text{ lb/min} \quad (18)$$

with β as in Equation (16).

The corresponding values of β, ρ_c and $\epsilon = (n-1)/n$, for all feasible n values, are given in Table 1.

If the **pressure drop is large,** during the acceleration process the velocity, v, of the gas increases steadily, as does its volume, V. The **rate** of the velocity increase is greatest at the very start, diminishing as expansion progresses. The contrary is true for the rate of the volume increase. As a result the ratio V/v, which is a measure of the nozzle cross-section required, first decreases as expansion progresses, reaching its minimum at the critical expansion ratio in the vena contracta. After that it increases, indicating that the nozzle cross-section also must widen out, in order to accommodate the increasing volume as expansion proceeds (Laval nozzle). If such widening of the nozzle is not provided for,

Table 1. Data for Gas Flow

n	$1/n$	$\epsilon = \frac{n-1}{n}$	j	$\gamma_c = 1/\rho_c$	ρ_c	$\left(\frac{\Delta p}{p_1}\right)_c$	β
1.00	1.	0	.775	1.649	.6065	.3935	.429
1.05	.9524	.0476	.741	1.679	.5955	.4045	.4365
1.10	.9091	.0909	.710	1.709	.5850	.4150	.4440
1.15	.8696	.1304	.682	1.741	.5745	.4255	.4513
1.20	.8333	.1667	.655	1.771	.5645	.4355	.4584
1.25	.8000	.2000	.631	1.802	.5550	.4450	.4652
1.30	.7692	.2308	.608	1.832	.5460	.4540	.4717
1.35	.7407	.2593	.587	1.862	.5370	.4630	.4780
1.40	.7143	.2857	.567	1.894	.5280	.4720	.4840
1.45	.6896	.3104	.549	1.923	.5200	.4800	.4900
1.50	.6667	.3333	.532	1.953	.5120	.4880	.4960

Table 2. Viscosity of Freons and Methyl Chloride in Centipoises (η_c)

t F	Liquids at saturation pressure							Vapors at 1 atm						
	F-11	F-12	F-21	F-22	F-113	F-114	Methyl chloride	F-11	F-12	F-21	F-22	F-113	F-114	Methyl chloride
−40	.980	.423	.629	.351		.879	.349	.0088	.0106	.0094	.0105		.0095	.0086
−20	.801	.371	.547	.316	1.566	.711	.321	.0092	.0109	.0098	.0109	.0090	.0098	.0090
0	.677	.335	.484	.291	1.263	.598	.298	.0096	.0113	.0101	.0113	.0093	.0102	.0094
+20	.586	.308	.436	.271	1.043	.516	.279	.0099	.0116	.0105	.0118	.0095	.0106	.0098
40	.517	.286	.397	.256	.876	.454	.263	.0103	.0119	.0108	.0122	.0098	.0109	.0101
60	.461	.269	.364	.243	.747	.405	.249	.0106	.0123	.0111	.0126	.0101	.0112	.0105
80	.417	.255	.337	.232	.646	.366	.237	.0110	.0126	.0115	.0130	.0103	.0116	.0108
100	.380	.242	.314	.223	.564	.334	.226	.0113	.0129	.0118	.0133	.0106	.0119	.0111
120	.349	.232	.294	.214	.497	.307	.217	.0116	.0132	.0121	.0137	.0108	.0122	.0115
140	.323	.222	.277	.207	.442	.284	.208	.0120	.0135	.0124	.0141	.0111	.0125	.0118
160	.300	.214	.262	.201	.395		.200	.0123	.0138	.0127	.0145	.0113		.0122
180	.281	.207	.248	.195	.356		.193	.0126	.0140	.0130	.0148	.0116		.0125
200	.263	.200	.236		.322		.186	.0129	.0143	.0133	.0152	.0118		.0128
220	.248		.225		.293		.180	.0132	.0146	.0136	.0156	.0120		.0131
240	.232		.215		.268		.175	.0135	.0149	.0139	.0159	.0123		.0134

Data by Benning & Markwood, *Refrig. Eng.* April 1939.

the gas cannot expand beyond the critical ratio. Thus the velocity given in Equation (17) is the maximum attainable with converging nozzles or plain orifices.

After the jet emerges from the nozzle or the vena contracta, the gas is free to expand further. However, unguided by surrounding walls, the pressure drives in every direction, the motion becomes irregular, and the jet explodes.

The flow, as given in Equation (18), will be the maximum in all cases, even if the nozzle widens out, since widening the nozzle increases the velocity but not the flow, which is fixed by the cross sectional area of the throat.

Theory of Viscosity

4. Viscosity is the cause of the fluid friction, which appears whenever two adjacent layers of a fluid slide past each other. This frictional resistance is of the nature of a shearing stress, its magnitude depending on the rate of sliding as well as on the viscosity. For one square foot of sliding area the drag is

$$\tau = \eta' \cdot (dv/dx) \text{ psf} \qquad (19)$$

where η' is the absolute viscosity and dv/dx is the velocity variation normally across one unit thickness of the sliding layer in fps/ft = 1/sec units, (called velocity gradient). From Equation (19), the magnitude and dimension of the absolute viscosity in fps is

$$\eta' = \frac{\tau}{(dv/dx)} \text{ in } \frac{\text{lb sec}}{\text{sq ft}} \text{ units} \qquad (20)$$

In the scientific literature the absolute viscosity is given in "poises" (i.e., in dyne sec /sq cm units) and is designated by η; often it is given in centipoises (1/100th of a poise), and is designated by η_c (also by Z). Since 1 lb = 981 × 453.6 dyne and 1 sq ft = 929 sq cm, it is evident that one (lb sec /sq ft) unit = 479 poises. Thus η poises or η_c centipoises becomes,

$$\eta/479 = \eta_c/47900 = \eta' \text{ in } \frac{\text{lb sec}}{\text{sq ft}} \qquad (21)$$

The **Reynolds number,** Re, is an important criterion when analyzing flow problems. For round pipes,

$$Re = \frac{D \cdot v \cdot \rho}{\eta'} \qquad (22)$$

which is dimensionless and without any factor, if all components are given in consistent units. In this formula ρ is the mass-density (i.e., $\rho = \gamma/g$). Substituting this we get

$$Re = \frac{D \cdot v \cdot \gamma}{(\eta' \cdot g)} = \frac{D \cdot v \cdot \gamma}{\mu'} \qquad (22a)$$

This value $\eta' \cdot g = \mu'$ also is called viscosity, although it appears in lb/ft sec units. Often the two units are badly confused in engineering literature. For the sake of clarity let us call this unit "dynamic" viscosity. Its numerical relation with the poise and centipoise unit is

$$\mu' = 32.2\eta' = 32.2(\eta/479)$$
$$= \frac{\eta}{14.88} = \frac{\eta_c}{1488} \text{ in } \frac{\text{lb}}{\text{ft sec}} \qquad (23)$$

The ratio $\eta'/\rho' = \mu'/\gamma$ is called "kinematic" viscosity. Its dimension is sq ft/sec, and it is designated by ν'. With $\gamma = 62.4\ s$, from Equation (23) we derive

$$\nu' = \mu'/\gamma = \frac{\mu'}{62.4s} = \frac{(\eta/s)}{929} \qquad (24)$$
$$= \nu/929 \text{ in sq ft/sec}$$

where $(\eta/s) = \nu$ is the kinematic viscosity in sq cm/sec units, called "stokes." Analogously, the ratio $\eta_c/s = \nu_c$ is called centistoke (1/100th of the stoke).

In the study of heat transfer the viscosity units are based on the hour as the unit of time. With η the absolute viscosity in poises we find:

$$\eta'_h = \frac{\eta'}{3600} = \frac{\eta}{1.725 \cdot 10^6} \text{ in } \frac{\text{lb hr}}{\text{sq ft}} \text{ units}$$

$$\mu'_h = 3600 \cdot \mu' = 242 \cdot \eta \text{ in } \frac{\text{lb}}{\text{ft hr}} \text{ units}$$

$$\nu'_h = 3600 \cdot \nu' = 3.875\ (\eta/s) \text{ in } \frac{\text{sq ft}}{\text{hr}} \text{ units}$$

Thus we have three types of units for expressing viscosity: (1) abs unit = force · time /area (symbol η); (2) dynamic unit = force /length · time (symbol μ); and (3) kinematic unit = area/time (symbol ν).

When making computations involving viscosity, it is important to distinguish between these units in order to avoid utter chaos in our calculations.

The viscosity of liquids decreases rapidly with rising temperature, and vice versa. As a rough rule we may say that the logarithm of the absolute viscosity varies linearly with the reciprocal of the absolute temperature; that $\log \eta = a + b/T$, is a straight line in a semilog plot with η on the logarithmic scale.

The viscosity of gases, on the contrary, increases with the temperature.

Viscous Flow

5. The magnitude of the Reynolds number is a criterion of the type of flow there

Table 3. Viscosity of Liquid NH_3, Centipoises (η_c)

Pressure, psia	Temperature, F							Saturation	
	0°	10°	20°	30°	40°	50°	60°	η_c	temp, F
30	—	—	—	—	—	—	—	.2503	−.6
40	.2539	.2482	—	—	—	—	—	.2474	11.7
50	.2562	.2509	.2444	—	—	—	—	.2432	21.7
60	.2585	.2533	.2474	.2398	—	—	—	.2398	30.2
70	.2605	.2556	.2499	.2427	—	—	—	.2362	37.7
80	.2624	.2576	.2521	.2553	.2369	—	—	.2328	44.4
90	.2643	.2595	.2541	.2476	.2397	.2301	—	.2295	50.5
100	.2661	.2613	.2559	.2496	.2420	.2331	—	.2264	56.0
120	.2691	.2645	.2594	.2534	.2461	.2376	.2277	.2204	66.0
140	.2718	.2675	.2625	.2567	.2495	.2413	.2318		
160	.2740	.2699	.2651	.2594	.2526	.2445	.2355		
180	.2760	.2720	.2674	.2617	.2549	.2472	.2387		
200	.2778	.2740	.2695	.2639	.2571	.2496	.2510		
220	.2794	.2756	.2712	.2657	.2590	.2515	.2430		
240	.2808	.2770	.2727	.2673	.2607	.2532	.2447		
260	.2820	.2783	.2740	.2687	.2623	.2542	.2463		
280	.2833	.2796	.2752	.2699	.2636	.2560	.2477		
300	.2844	.2807	.2763	.2710	.2647	.2572	.2489		
320	.2853	.2817	.2773	.2719	.2657	.2584	.2501		
340	.2861	.2825	.2782	.2728	.2666	.2594	.2510		
360	.2868	.2833	.2789	.2736	.2675	.2603	.2520		

Table 4. Viscosity of Liquid Methyl Chloride, Centipoises (η_c)

Pressure, psia	Temperature, F									Saturated liquid	
	0°	10°	20°	30°	40°	50°	60°	70°	80°	η_c	temp, F
20	.3083	—	—	—	—	—	—	—	—	.3064	2.9
30	.3133	.3064	.2993	—	—	—	—	—	—	.2975	22.3
40	.3177	.3112	.3043	.2965	—	—	—	—	—	.2901	36.6
50	.3210	.3145	.3079	.3002	.2911	—	—	—	—	.2826	48.4
60	.3239	.3176	.3109	.3033	.2943	.2846	—	—	—	.2760	58.6
70	.3265	.3202	.3136	.3061	.2971	.2877	.2778	—	—	.2704	67.2
80	.3283	.3222	.3154	.3079	.2994	.2903	.2805	.2701	—	.2634	76.1
90	.3297	.3238	.3170	.3095	.3011	.2921	.2824	.2719	.2607	.2581	82.2
100	.3303	.3248	.3181	.3107	.3023	.2933	.2835	.2731	.2617	.2505	88.7

will be in conduit. If $Re<1200$, the flow is always viscous; this flow is characterized by the fact that the fluid particles move past each other in parallel lines without mixing. The velocity distribution over the cross section depends on the shape of the conduit.

In round pipes viscous flow is coexistent with a parabolic velocity distribution over the cross section. The maximum velocity in the pipe center is twice the average velocity expressed as $v=Q''/A$. Right at the pipe surface the velocity is nil, but at that point the velocity gradient (dv/dx) has its maximum value of $8v/D$. The force required to move the contents of an l-ft long straight pipe against this fluid friction at the wall is

$$\Delta P \cdot A = U \cdot l \cdot \eta' \cdot (dv/dx) = 8\pi\eta' v l \text{ lb} \quad (25)$$

with ΔP as the pressure drop in psf over the l ft of pipe and $(dv/dx)=8v/D$ right at the inside surface.

With the pipe diameter in inches, the viscosity in poises, and the flow in gpm or cfm, we find the pressure drop for $l=100$ ft in psi as

$$\Delta p'_0 = \frac{6.68v\eta}{d^2} = \frac{2.73G\eta}{d^4} = \frac{20.42Q'\eta}{d^4} \text{ psi} \quad (26)$$

or in ft of the fluid flowing

$$h'_0 = \frac{15.42v\nu}{d^2} = \frac{6.3G\nu}{d^4} = \frac{47.11Q'\nu}{d^4} \text{ ft} \quad (27)$$

where $\nu=\eta/s$ is the kinematic viscosity of the fluid in stokes with η and s for the condition existing at the flow.

(If the viscosity is taken in centipoises $(\eta_c=100\eta)$, or in centistokes $(\nu_c=100\nu)$, Equations (26) and (27) must be divided by 100.)

Equation (27) is used with some modifications for determining ν, the kinematic viscosity of fluids. The most common method for this determination is to meas-

Table 5. Viscosity of Liquid Sulfur Dioxide, Centipoises (η_c)

Pressure, psia	Temperature, F								Saturated liquid	
	0°	10°	20°	30°	40°	50°	60°	70°	η_c	temp, F
10	—	—	—	—	—	—	—	—	.4790	−1.34
20	.4840	.4544	.4232	—	—	—	—	—	.4013	26.44
30	.4921	.4629	.4324	.4003	—	—	—	—	.3470	44.75
40	.4980	.4695	.4393	.4074	.3735	.3366	—	—	.3040	58.33
50	.5028	.4747	.4451	.4134	.3792	.3431	.3050	.2665	.2548	70.40
60	.5065	.4794	.4499	.4184	.3841	.3484	.3108	.2731	—	—
70	.5099	.4834	.4541	.4225	.3885	.3529	.3159	.2789		
80	.5125	.4868	.4577	.4261	.3921	.3564	.3198	.2835		
90	.5147	.4894	.4609	.4292	.3950	.3595	.3233	.2874		
100	.5165	.4917	.4634	.4319	.3975	.3621	.3265	.2909		
110	.5180	.4937	.4655	.4340	.3997	.3643	.3293	.2940		

Table 6. Viscosity of Refrigerants in Centipoises (η_c)

t °C	NH_3			CO_2			SO_2			Butane	Propane	Ethane	Ethylene	t °C
	Liquid	Vapor		Liquid	Vapor		Liquid	Vapor		Vapor at 1 atm				
	at ps	at 1 atm	at ps	at ps	at 1 atm	at ps	at ps	at 1 atm	at ps					
−60	.380	.0071			.0108								.00735	−60
−40	.275	.0078			.0118								.00805	−40
−20	.255	.0086	.0109	.120	.0128		.465		.0106				.00875	−20
−10	.245		.0113	.111		.0167	.412		.0113					−10
0	.240	.0093	.0118	.087	.0138	.0174	.368	.0116	.0123	.0069	.0075	.0086	.00940	0
+10	.230		.0124	.071		.0183	.333		.0135					+10
20	.220	.0100	.0129	.048	.0148	.0203	.304	.0126	.0151	.0074	.0081	.0092	.01005	20
30				.032		.0235	.279		.0169					30
50		.0111			.0162			.0140		.0082	.0088	.0101	.01100	50
100		.0130			.0185			.0163		.0095	.0101	.0115	.01260	100
150		.0148			.0208			.0186		.0110	.0113	.0128	.01400	150
200		.0166			.0229			.0207			.0125	.0142	.0154	200
250		.0184			.0249			.0227			.0136	.0154	.0166	250
300		.0202			.0268			.0246			.0144			300

Data from Henning, Wärmetechnische Richtwerte. VDI—1938.

Table 7. Kinematic Viscosity of Water (ν_c in centistokes)

t, F	ν_c	t, F	ν_c	t, F	ν_c	t, F	ν_c
32	1.794	80	.865	130	.517	180	.357
40	1.546	90	.768	140	.478	190	.336
50	1.310	100	.687	150	.441	200	.316
60	1.130	110	.622	160	.410	210	.299
70	.984	120	.565	170	.383	220	.283

Data from Smithsonian tables.

ure the time (t) as the number of seconds required by a fixed volume of the fluid to pass by gravity through a capillary of strictly fixed dimensions. The time, found with the Saybolt universal viscometer, is expressed as "Universal Saybolt seconds." The relation between this time and the kinematic viscosity in centistokes is

$$\nu_c = \eta_c/s = \left(0.220t - \frac{180}{t}\right) \text{ centistokes} \quad (28)$$

(For $\nu_c = 1$, the kinematic viscosity of water at 68.4 F, the Saybolt time is 31 seconds.)

From the discussion above, it becomes apparent that citing viscosity data without the corresponding temperatures is meaningless.

Turbulent Flow and Reynolds Number (Re)

6. Liquids. Whenever the Reynolds number $Re > 2500$, the flow will be turbulent; the frictional resistance increases almost with the square of the velocity as compared to the linear increase during viscous flow.

The basic relation for the head loss from frictional resistance is Darcy's equation

$$h' = \xi \cdot l/m' \cdot \left(\frac{v^2}{2g}\right) \text{ ft} \quad (29)$$

which gives the head loss as a multiple of the velocity head in feet of the fluid flowing. In this equation, ξ is the general friction factor (for any type of conduit), and $m' = (A/U)$ is the hydraulic radius in ft for the conduit in question.

For round pipes $m' = D/4$. Thus $\xi/m' = 4\xi/D = f/D$ and we get

$$h' = f \cdot \frac{l}{D} \cdot \left(\frac{v^2}{2g}\right) = f \cdot \frac{12l}{d} \cdot \left(\frac{v^2}{2g}\right) \text{ ft} \quad (30)$$

which is called **Fanning's equation,** and is useful only for round pipes with $f = 4\xi$ as the friction factor.

It is customary to write up the friction loss equation for $l = 100$ ft of actual of equivalent length of pipe. With the hy-

draulic radius as $m=(A_i/U_i)$ in inches, and the diameter also in inches, by combining all constants into one factor we get

$$h'_0=18.63\ (\xi/m)\cdot v^2=18.63\ (f/d)\cdot v^2 \text{ ft} \quad (31)$$

or

$$\begin{aligned}\Delta p'_0&=8.075s\cdot(\xi/m)\cdot v^2\\&=8.075s\cdot(f/d)\cdot v^2 \text{ psi}\end{aligned} \quad (32)$$

When selecting a proper value for the friction factor, care must be taken to determine whether it is a ξ value or an $f=4\xi$ value, and to use the proper equation.

With the flow in gpm, cfm, or in lb/min, we can modify Fanning's equation for $l=100$ ft of round pipe into

$$h'_0=3.11f\cdot\frac{G^2}{d^5}=174f\cdot\frac{(Q')^2}{d^5} \text{ ft} \quad (33)$$

or

$$\Delta p'_0=1.35fs\cdot\frac{G^2}{d^5}=75.4fs\cdot\frac{(Q')^2}{d^5} \text{ psi} \quad (34)$$

and

$$\Delta p'_0=1.208f\cdot\frac{(W')^2}{\gamma\cdot d^5}=\frac{1.936}{100}\cdot f\cdot\frac{(W')^2}{s\cdot d^5} \text{ psi} \quad (35)$$

By coordinating the results of a great many tests of turbulent flow, it has been found that the friction factor can be plotted as a function of the Reynolds number *Re*. It is usually plotted on a logarithmic scale.

McAdams gives such a plot; the lower line is for drawn, smooth tubes, whereas the upper line gives good average values for standard commercial steel pipes.[1]

This empirical $\xi=f(Re)$ relation can be expressed also by an equation; the values for smooth tubes follow the equation

$$1000\xi=1.4+125/Re^{0.32}$$

whereas those for standard steel pipes are

$$1000\xi=3.07+188.6/Re^{0.38}$$

Kemler and Pigott have also analyzed a great number of tests and segregated the friction factor data into different pipe diameter groups, which they plotted separately against *Re*, the Reynolds number. These plots reflect the influence of the relative roughness of the pipe on the magnitude of the friction factor.

Moody has correlated Reynolds number, diameter, roughness and friction factor by means of the Colebrook function,

$$\frac{1}{\sqrt{f}}=-2\log\left(\frac{\epsilon/D}{3.7}+\frac{2.51}{R\sqrt{f}}\right)$$

where ϵ is a factor, expressed in feet, representative of the roughness of the pipe, which for any type of pipe is constant irrespective of diameter. Stickney has expressed this in chart form, **Fig. 1,** in which read from diameter at the top, down to type of pipe, across to *Re*, and interpolate for *f*. Note that $f=4\xi$.

When figuring a case of pipe friction loss, one must know the numerical magnitude of all the variables given in the equations. With these figures known, one first calculates the Reynolds number *Re* pertaining to the specific case. This value of *Re* not only determines the magnitude of the friction factor, but it is also the criterion of what type of flow will exist. If $Re<1200$, the flow is definitely viscous; if $Re>2500$, it will be turbulent. For *Re* values in between, the flow is uncertain.

While the above equations serve primarily for finding the friction loss in turbulent flow, they can be used also for viscous flow simply by substituting $\xi=16/Re$, or $f=64/Re$.

The Reynolds number, which obviously plays an important role, can be expressed in different ways. Its basic form is given by Equations (22) and (22a); other forms are (with η in poises and $\nu=(\eta/s)$ in stokes)

$$Re=1.24\ \frac{dv\gamma}{\eta}=77.42\ \frac{dv}{\nu} \quad (36)$$

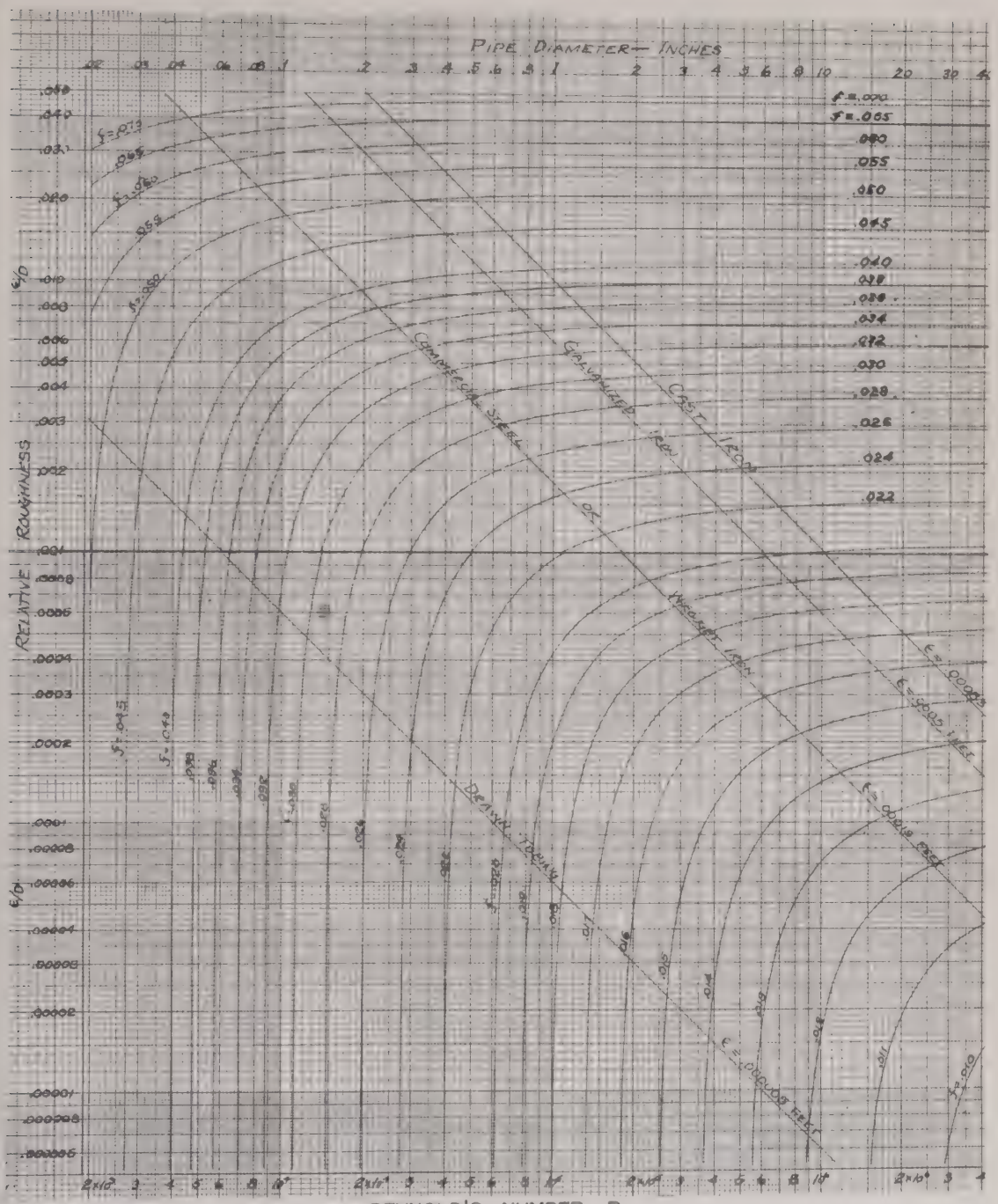
and with the flow in gpm, cfm or lb/min,

$$\begin{aligned}Re&=0.5066\ \frac{G\cdot\gamma}{d\cdot\eta}=31.62\ \frac{G}{d\cdot\nu}\\Re&=3.79\cdot\frac{Q'\cdot\gamma}{d\cdot\eta}=236.5\ \frac{Q'}{d\cdot\nu}\\Re&=3.79\ \frac{W'}{d\cdot\eta}\end{aligned} \quad (37)$$

(If the viscosity is taken in centipoises ($\eta_c=100\eta$) or in centistokes ($\nu_c=100\nu$), then Equations (36) and (37) must be multiplied by 100 to get *Re* correctly.)

For "not-round" ducts, we substitute in Equation (36), but not in Equation (37),

[1] These are ξ values; multiply by four to get *f* values.



f = FRICTION FACTOR ($f = 4\mathfrak{f}$)

$\frac{\epsilon}{D}$ = RELATIVE ROUGHNESS

ϵ AND D IN SAME UNITS (inches, feet)

ϵ = ABSOLUTE ROUGHNESS

D = PIPE DIAMETER

$R_e = \frac{VD}{\nu'} = 7740\frac{\mathfrak{f}V}{\nu_c} = 3162\frac{G}{\mathfrak{f}\nu_c}$

V = FLUID VELOCITY ft/sec

$\mathfrak{f}$ = PIPE DIAMETER inches

D = " " feet

ν' = KINEMATIC VISCOSITY sq ft/sec

ν_c = " " Centistoke

G = GALLONS PER MINUTE

Fig. 1. Friction Factors

the diameter with its equivalent in terms of hydraulic radius: $d = 4m = (A_i/U_i)$, with the velocity figured according to Equation (1).

For annular cross sections (double-pipe heat exchangers) $m = (d_1 - d_2)/4$, with d_1 the inside diameter of the outside pipe and d_2 the outside diameter of the inside pipe in inches.

Gases and vapors. All our formulae for friction loss hold good to a limited degree also for gas and vapor lines, provided the pressure drop is less than 10% of the initial absolute pressure. Most friction loss problems encountered by the refrigerating engineer in connection with gas and vapor lines fall well within this restriction, and thus can be handled with the simple equations for liquids. However, for easier handling one may substitute $\gamma = 1/V$

$$\Delta p'_0 = .1294f \frac{v^2}{V \cdot d} = 1.208f \cdot \frac{(Q')^2}{V \cdot d^5} = 1.208f \cdot \frac{V \cdot (W')^2}{d^5} \text{ psi} \qquad (38)$$

for $l = 100$ ft of pipe, with $V = 1/\gamma$ as the specific gas volume in cu ft/lb (for the p_1 starting pressure).

If greater accuracy is desired, one may first figure a preliminary $\Delta p'$ value, and then with this calculate the correction factor $(1 + \Delta p'/2 \cdot p_1)$ as a multiplier for the erstwhile $\Delta p'$.

The friction factor f in Equation (38) is a function of the Reynolds number Re. McAdams and Sherwood found that for air, gas, and steam flow one may write

$$f = 4\xi = 0.0216 + 186/Re \qquad (39)$$

with Re as per any of its equations.

However, for gas flow the most convenient procedure is to express Re with the flow as W' in lb/min. Since for gases η varies only with the temperature, Re and f also remain constant during the flow, if the temperature stays constant. Both the velocity and gas density change during the flow, but in such a manner that $v \cdot \gamma = v/V$ remains constant.

In his many tests with air, gas, and saturated and superheated steam, Fritzsche found that the friction factor f, primarily a function of the fluid weight passing through the pipe, depends only slightly on the diameter. His test results can be expressed as

$$f = \frac{3.30}{100} \cdot \frac{d^{.027}}{(W')^{.143}} = \frac{3.45}{100 \cdot (W')^{.143}} \qquad (40)$$

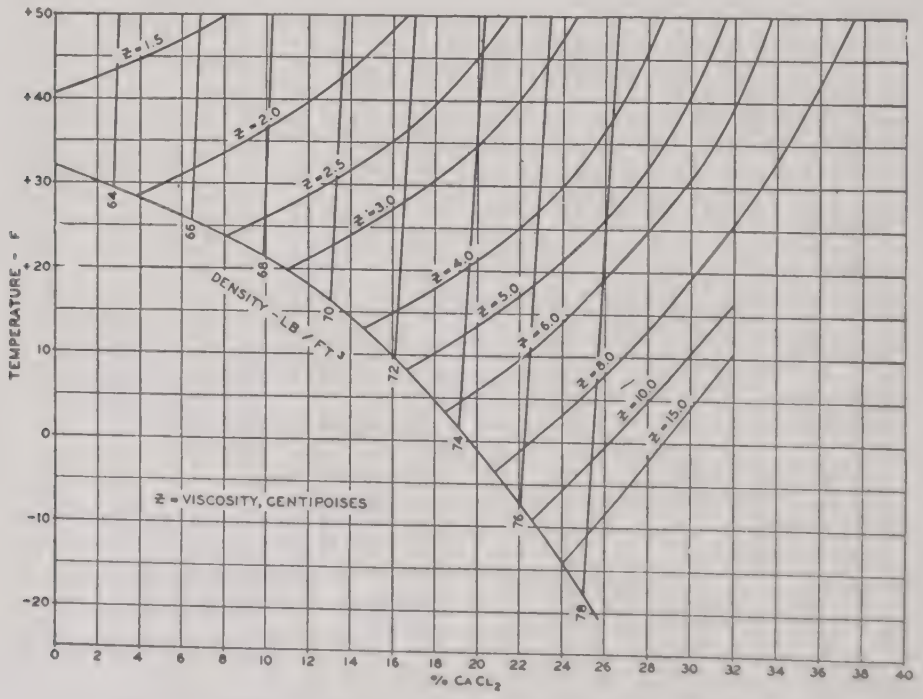


Fig. 2. Diagram of Viscosity Values of Calcium Chloride Brine

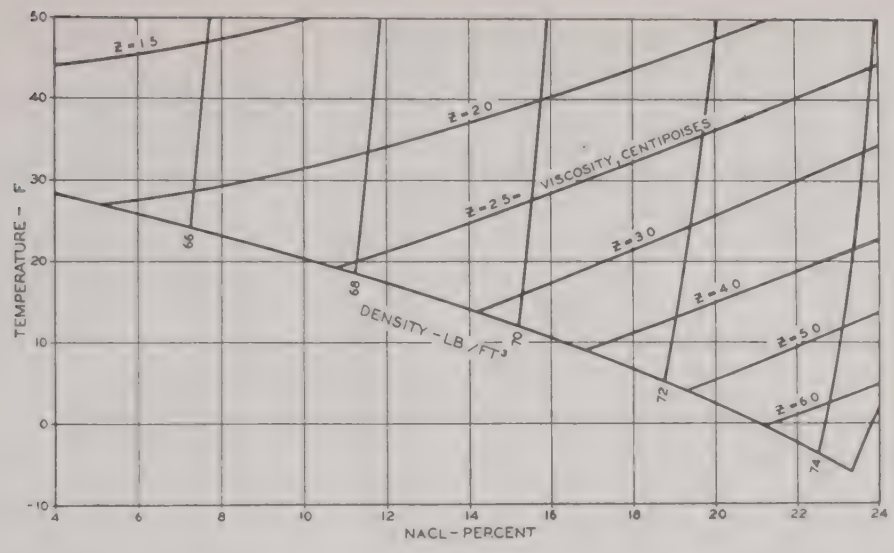


Fig. 3. Viscosity and Density of Sodium Chloride

The left side of Equation (40) is the original equation, while the right side is a simplification; since $d^{.027}$ varies only slightly with d, an average value, say $5^{.027} = 1.0444$, can be substituted for it, resulting in 3.45, as shown above. This equation of Fritzsche, which may be used very conveniently, gives f values close to those found in Equation (39).

Miscellaneous Losses

7. Losses due to obstructions in the line. Elbows, fittings, and valves in the line disturb the flow pattern, causing an additional head loss, which can be expressed either by a resistance factor k (giving this head loss as a multiple of the velocity head: $h' = k(v^2/2g)$ ft), or by an equivalent length of pipe (l_e) of the same diameter, which may be added to the actual pipe length when figuring the total friction loss.

The Crane Company, in its "Flow of Fluids," has published a nomogram **(Fig. 6)** which gives the equivalent length for almost any available fitting.

The resistance of **bends** and **welding elbows** is less than that of screw elbows, depending on the (R/d) relative radius of the bend; 90° bends show a minimum of resistance, when (R/d) is around 3. The equivalent length for 90° bends, expressed in pipe diameters, are given in the Tube-Turns[11] catalog as follows:

R/d	l_e (in. diam)
.5	40
1.0	18.5
1.5	12.2
2	10
2.5	9.2
3	9
3.5	9.2
4	10
4.5	11

For 45° bends take 64% and for 180° bends 134% of the given l_e values.

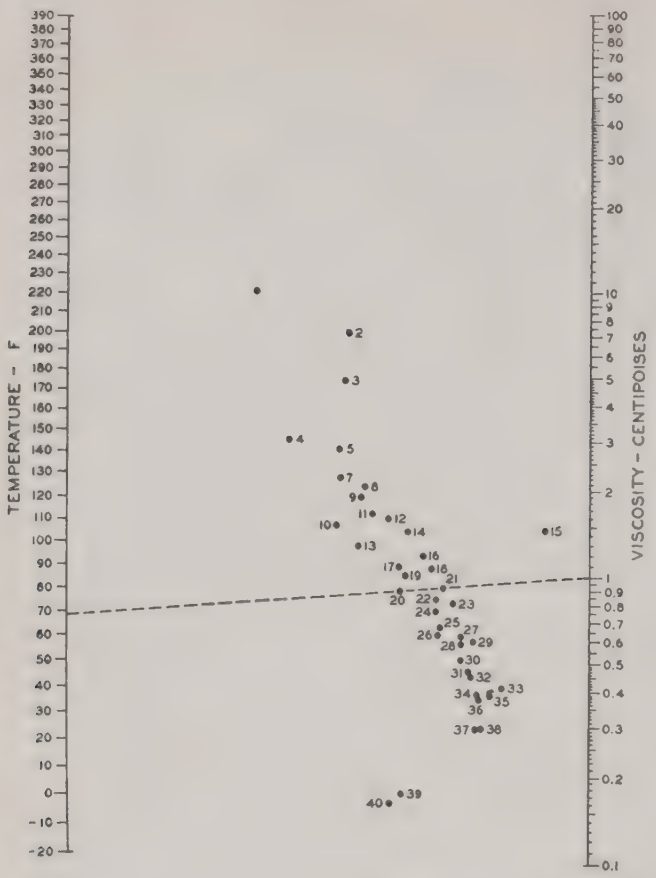
The resistance of **single-weld mitre bends** varies with δ, the angle of the deviation. According to Kirchbach (Hutte: 26th ed):

δ	22.5°	30°	45°	60°	90°
k	.11	.17	.32	.68	1.27

Where k is a multiplying factor of $v^2/2g$.

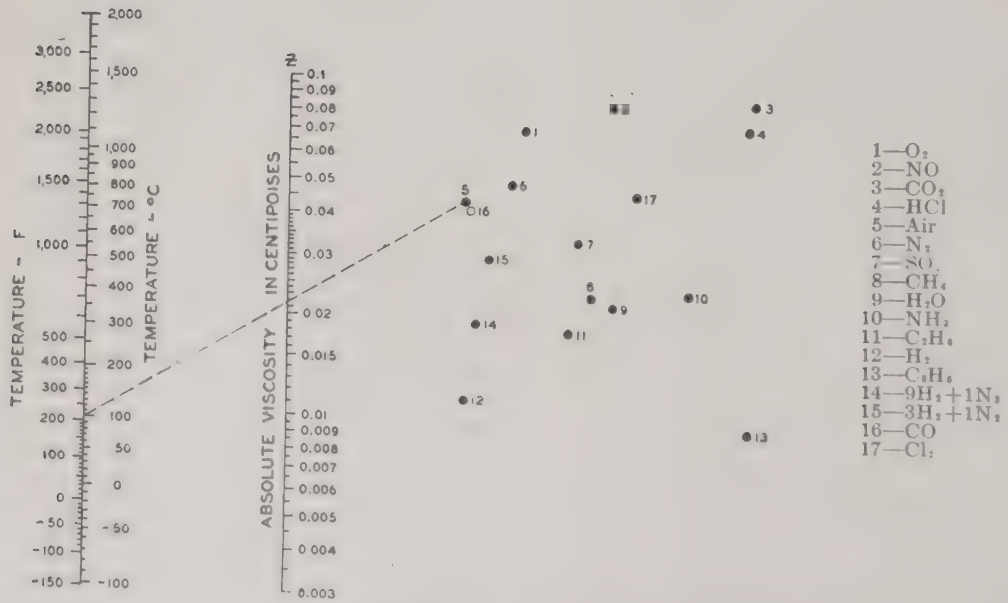
Entrance loss. Whenever a fluid enters a pipe and the jet suffers a contraction at the entrance, there will be a loss of head. The jet is first speeded up to the greater velocity v_0 in the vena contracta, from which it slows down again to the lesser velocity v in the pipe; during this slowdown the kinetic energy of the jet is never fully reconverted into static head (pressure), since part of it is lost in impact, turbulence and eddies.

The head consumed in the speed-up process is $h_0 = v_0^2/2g = (1+k) \cdot v^2/2g$ ft, of which $v^2/2g$ remains in the fluid and



18	Acetic Acid 100%
12	Acetic Acid 70%
34	Acetone
17	Allyl Alcohol
39	Ammonia
8	Aniline
26	Benzene
11	Butyl Alcohol
40	Carbon Dioxide
33	Carbon Disulfide
21	Carbon Tetrachloride
22	Chlorbenzene
29	Chloroform
37	Ether
19	Ethyl Alcohol 100%
10	Ethyl Alcohol 40%
30	Ethyl Acetate
23	Ethylene Chloride
4	Ethylene Glycol
1	Glycerol 100%
7	Glycerol 50%
31	Heptane
36	Hexane (N)
15	Mercury
26	Methyl Alcohol 100%
24	Methyl Alcohol 90%
13	Methyl Alcohol 30%
32	Methyl Acetate
9	Naphthalene
14	Nitrobenzene
28	Octane
38	Pentane
5	Phenol
35	Sulfur Dioxide
2	Sulfuric Acid 111%
3	Sulfuric Acid 98%
6	Sulfuric Acid 60%
27	Toluene
16	Turpentine
20	Water

Fig. 4. Viscosity of Various Liquids[6]



1—O_2
2—NO
3—CO_2
4—HCl
5—Air
6—N_2
7—SO_2
8—CH_4
9—H_2O
10—NH_3
11—C_2H_4
12—H_2
13—C_6H_6
14—$9H_2+1N_2$
15—$3H_2+1N_2$
16—CO
17—Cl_2

Fig. 5. Viscosity of Gases at Atmospheric Pressure[6]

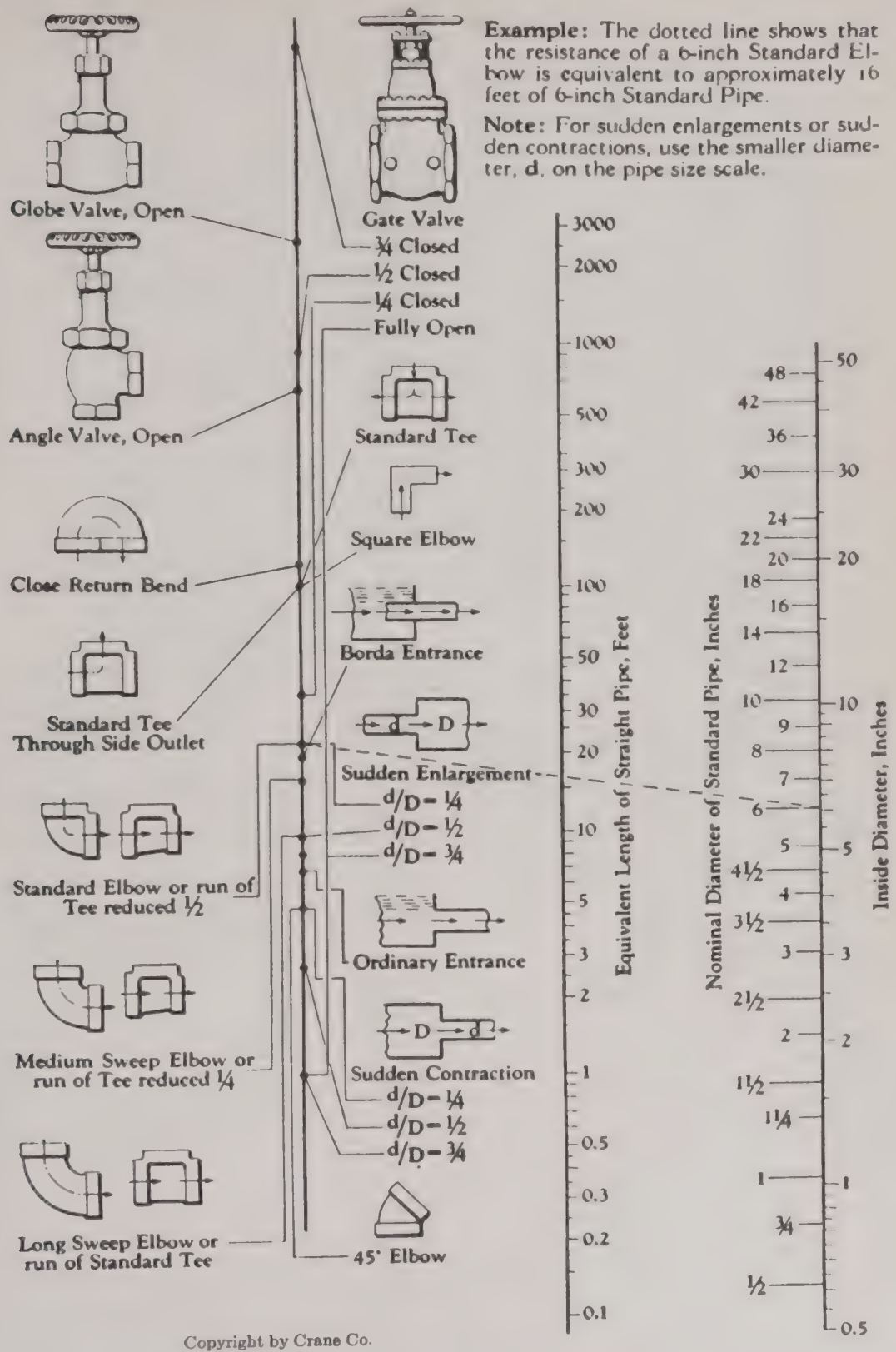


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Fig. 6. Resistance of Valves and Fittings to Flow of Fluids

$k \cdot v^2/2g$ is lost. The greater the contraction of the jet, the larger is k.

If the **pipe inlet is flush** with the face of the wall and well rounded, there will be no contraction, and thus no entrance loss: $k = .06$ or less and covers only the friction loss proper. If the pipe edge is only slightly rounded, $k = 0.25$; and if the edge is a sharp 90°, then $k = 0.50$.

If the **pipe protrudes** beyond the face of the wall (Borda entrance), the contraction increases, and thus the k value is increased. For standard pipes with a sharp 90° inside edge, $k = 0.80$.

The so-called **"reversal loss"** in waterboxes of multipass coolers and condensers is the same $h_0 = (1+k) \cdot v^2/2g = h_r'$ ft head loss for each pass; the fluid must be reaccelerated during its entrance to each successive pass. The Heat Exchange Institute has published data for reversal losses which follow the equation

$$h_r' = \frac{8.35}{100} \cdot v^{1.46} \text{ ft} \qquad (41)$$

for each pass with v the velocity in the tubes in fps.

Loss from sudden enlargements. In this case the fluid jet, going from the smaller pipe with the velocity v_1 into the larger one, plunges into a turbulent mass of fluid, from which it finally emerges with the velocity v_2 fixed by the flow and the pipe size. The loss of energy during this impact is expressed by the Borda-Carnot equation as

$$h' = (v_1 - v_2)^2/2g = \frac{v_1^2}{2g}\left(1 - \frac{v_2}{v_1}\right)^2 \text{ ft}$$

To avoid head loss in pipe transitions, they should be made as slender cones with an included angle of not more than 8° to 10° and with smooth joints.

Loss from sudden contraction. This is essentially of the same nature as the entrance loss. However, the fluid approaching the reduced inlet already has the velocity v_1. Thus the contraction at the new entrance is less and k correspondingly less, the larger the velocity ratio v_1/v_2, with v_2 as the velocity in the following smaller pipe. O'Brien and Hickox give the following data:

v_1/v_2	k
0.1	.362
0.2	.338
0.3	.308
0.4	.267
0.5	.221
0.6	.164
0.7	.105
0.8	.053
0.9	.015

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PART II

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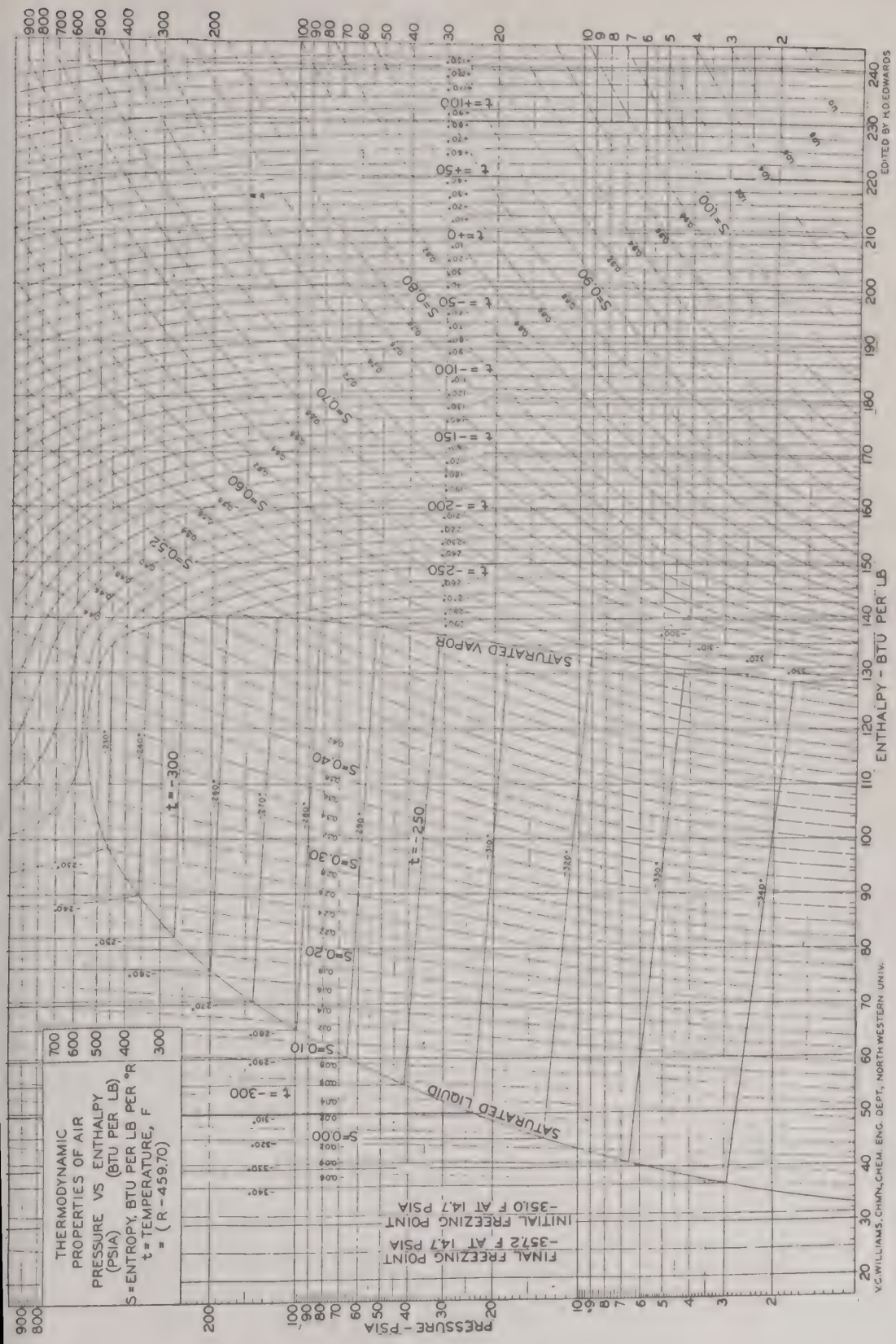
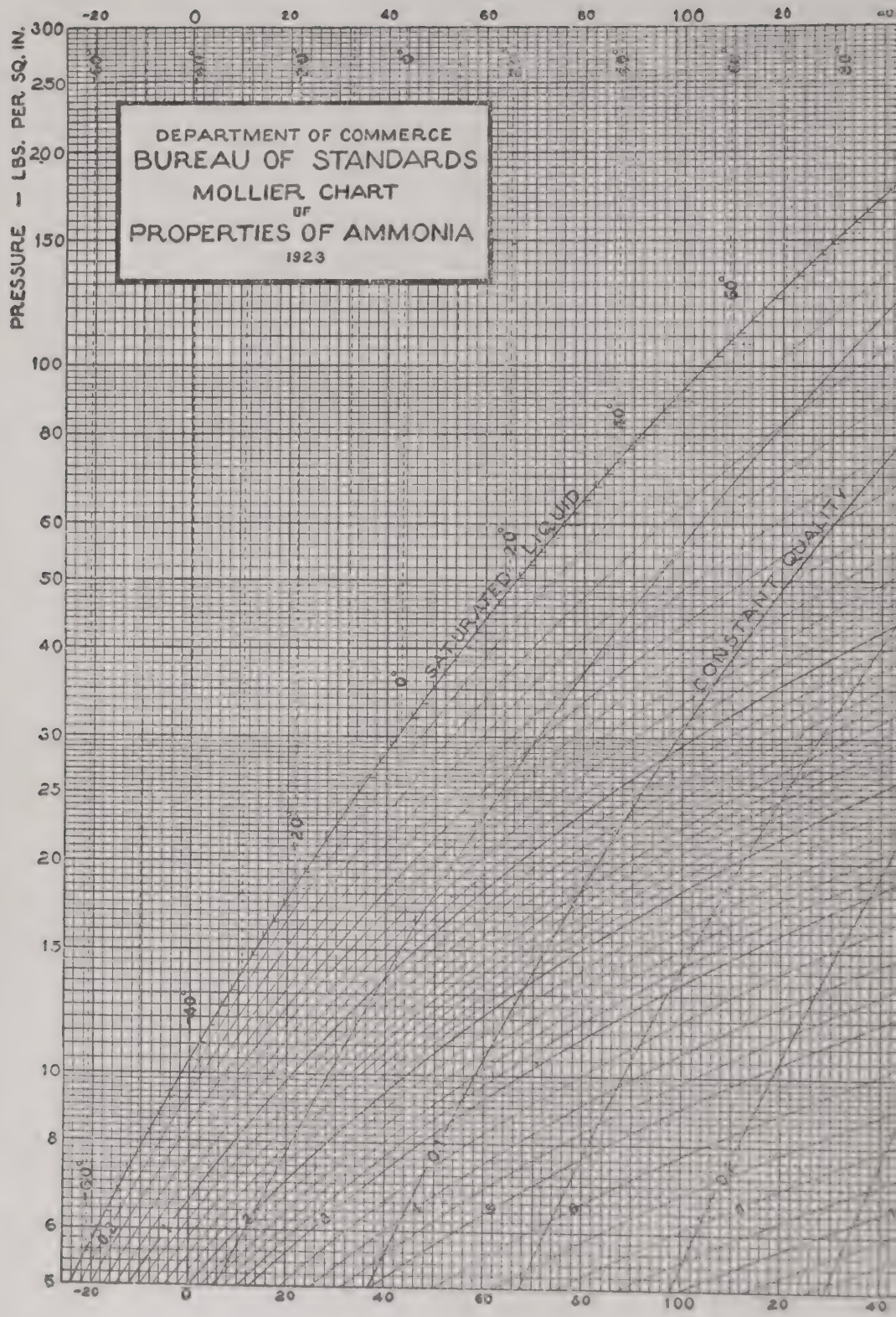

THERMODYNAMIC PROPERTIES OF AIR
PRESSURE VS ENTHALPY
(PSIA) (BTU PER LB)
S=ENTROPY, BTU PER LB PER °R
t=TEMPERATURE, F
=(R-459.70)
FINAL FREEZING POINT
-352 F AT 14.7 PSIA
INITIAL FREEZING POINT
-351.0 F AT 14.7 PSIA
SATURATED LIQUID
SATURATED VAPOR
PRESSURE - PSIA
ENTHALPY - BTU PER LB
V.C. WILLIAMS, CHEM. ENG. DEPT, NORTH WESTERN UNIV.
EDITED BY H.O. EDWARDS

Chart 1

Table 1. Ammonia—Properties of Liquid and Saturated Vapor

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−105	1.00	*27.9	45.71	223.14	−66.6	569.1	−0.1717	1.6211
−104	1.04	27.8	45.67	214.23	−65.6	569.6	− .1689	1.6174
−103	1.08	27.7	45.63	205.90	−64.5	570.1	− .1661	1.6136
−102	1.14	27.7	45.59	197.70	−63.5	570.6	− .1634	1.6100
−101	1.19	27.5	45.55	190.08	−62.5	571.0	− .1606	1.6062
−100	1.24	*27.4	45.51	182.90	−61.5	571.4	−0.1579	1.6025
− 99	1.29	27.3	45.47	175.42	−60.5	571.9	− .1552	1.5988
− 98	1.35	27.2	45.43	168.48	−59.5	572.3	− .1525	1.5952
− 97	1.41	27.0	45.40	161.98	−58.5	572.7	− .1498	1.5915
− 96	1.47	26.9	45.36	155.92	−57.5	573.2	− .1471	1.5878
− 95	1.52	*26.8	45.32	150.30	−56.5	573.6	−0.1444	1.5842
− 94	1.59	26.7	45.28	144.68	−55.5	574.1	− .1417	1.5806
− 93	1.66	26.5	45.24	139.27	−54.5	574.5	− .1390	1.5771
− 92	1.73	26.4	45.20	134.06	−53.4	575.0	− .1363	1.5736
− 91	1.79	26.2	45.16	129.06	−52.4	575.5	− .1336	1.5702
− 90	1.86	*26.1	45.12	124.28	−51.4	575.9	−0.1309	1.5667
− 89	1.94	26.0	45.08	119.75	−50.4	576.3	− .1281	1.5634
− 88	2.02	25.8	45.04	115.37	−49.4	576.7	− .1254	1.5600
− 87	2.11	25.6	45.00	111.31	−48.4	577.2	− .1227	1.5566
− 86	2.18	25.5	44.96	107.39	−47.4	577.6	− .1199	1.5533
− 85	2.27	*25.3	44.92	103.63	−46.4	578.0	−0.1171	1.5499
− 84	2.36	25.1	44.88	99.87	−45.4	578.4	− .1144	1.5466
− 83	2.46	24.9	44.84	96.28	−44.3	578.9	− .1117	1.5433
− 82	2.55	24.7	44.80	92.86	−43.3	579.3	− .1090	1.5401
− 81	2.65	24.5	44.76	89.65	−42.3	579.7	− .1063	1.5368
− 80	2.74	*24.3	44.73	86.54	−41.3	580.1	−0.1036	1.5336
− 79	2.85	24.1	44.68	83.50	−40.3	580.5	− .1009	1.5304
− 78	2.96	23.9	44.64	80.61	−39.2	581.0	− .0983	1.5273
− 77	3.07	23.6	44.60	77.90	−38.2	581.4	− .0956	1.5241
− 76	3.19	23.4	44.56	75.30	−37.2	581.8	− .0930	1.5210
− 75	3.30	*23.2	44.52	72.80	−36.2	582.2	−0.0903	1.5178
− 74	3.43	22.9	44.48	70.35	−35.2	582.7	− .0877	1.5148
− 73	3.56	22.7	44.44	68.01	−34.2	583.1	− .0800	1.5117
− 72	3.69	22.4	44.40	65.78	−33.1	583.6	− .0824	1.5087
− 71	3.82	22.2	44.36	63.70	−32.1	584.0	− .0797	1.5056
− 70	3.94	*21.9	44.32	61.65	−31.1	584.4	−0.0771	1.5026
− 69	4.09	21.6	44.28	59.60	−30.1	584.8	− .0745	1.4997
− 68	4.24	21.3	44.24	57.64	−29.4	585.3	− .0719	1.4969
− 67	4.39	21.0	44.19	55.78	−28.0	585.8	− .0694	1.4940
− 66	4.54	20.7	44.15	54.01	−27.0	586.2	− .0668	1.4911
− 65	4.69	*20.4	44.11	52.34	−26.0	586.6	−0.0642	1.4833
− 64	4.86	20.1	44.07	50.79	−25.0	587.0	− .0616	1.4856
− 63	5.03	19.6	44.03	49.26	−23.9	587.5	− .0590	1.4829
− 62	5.20	19.3	43.99	47.74	−22.9	588.0	− .0565	1.4802
− 61	5.38	18.9	43.95	46.23	−21.9	588.4	− .0539	1.4774
− 60	5.55	*18.6	43.91	44.73	−20.9	588.8	−0.0514	1.4747
− 59	5.74	18.2	43.87	43.37	−19.8	589.3	− .0487	1.4720
− 58	5.93	17.8	43.83	42.05	−18.8	589.7	− .0460	1.4694
− 57	6.13	17.4	43.78	40.79	−17.8	590.2	− .0434	1.4667
− 56	6.33	17.0	43.74	39.56	−16.7	590.6	− .0407	1.4640
− 55	6.54	*16.6	43.70	38.38	−15.7	591.0	−0.0381	1.4614
− 54	6.75	16.2	43.66	37.24	−14.6	591.5	− .0355	1.4588
− 53	6.97	15.7	43.62	36.15	−13.6	591.9	− .0330	1.4563
− 52	7.20	15.3	43.58	35.09	−12.6	592.3	− .0305	1.4537
− 51	7.43	14.8	43.54	34.06	−11.5	592.8	− .0279	1.4513

* Inches of mercury below one standard atmosphere (29.92 in.).


DEPARTMENT OF COMMERCE
BUREAU OF STANDARDS
MOLLIER CHART
OF
PROPERTIES OF AMMONIA
1923
PRESSURE — LBS. PER SQ. IN.
300
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100
80
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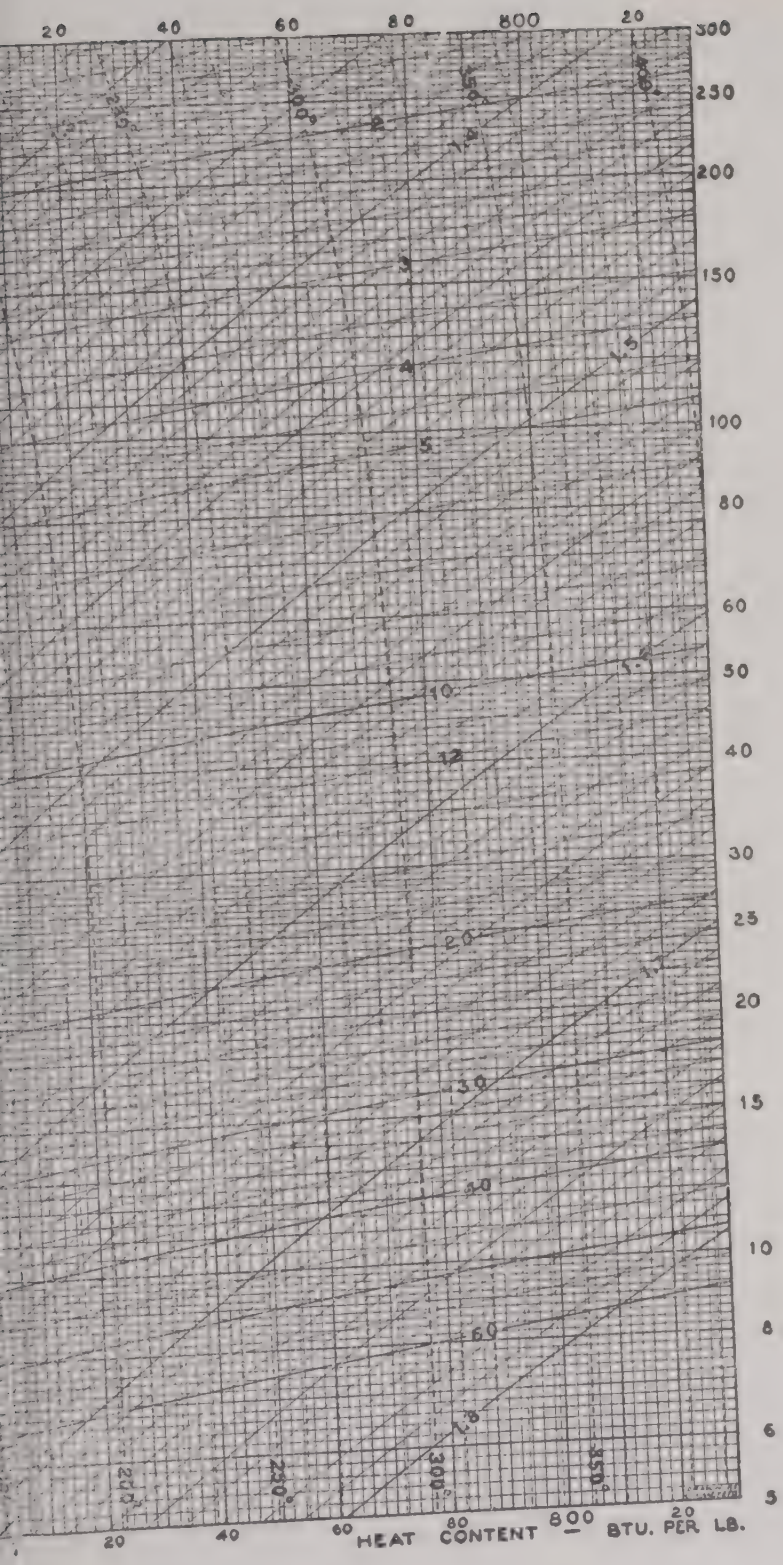
140
SATURATED LIQUID
CONSTANT QUALITY



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. datum −40 F
ı per lb F

Vapor s_v
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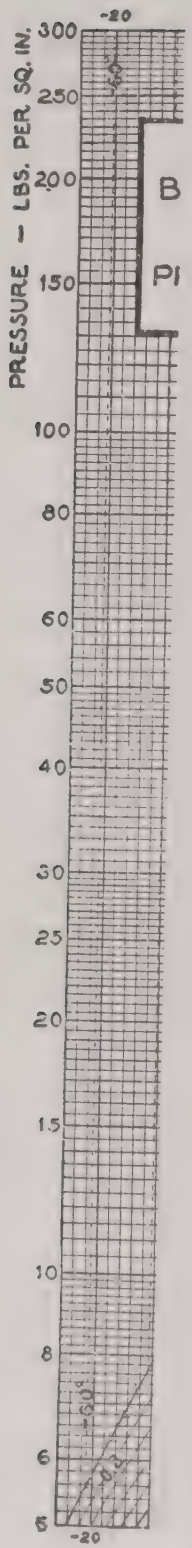
PRESSURE — LBS. PER SQ. IN.
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. datum −40 F
ı per lb F

Vapor s_v
1.4487
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1.4413
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1.4363
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1.4314
1.4290
1.4266
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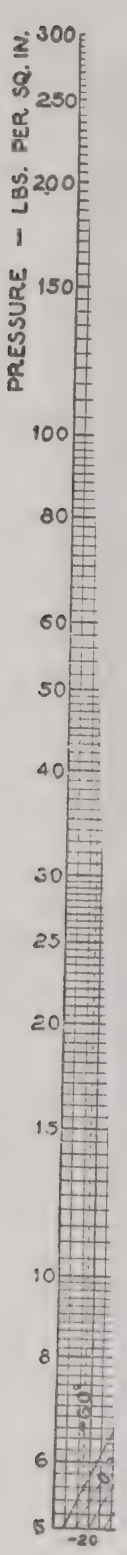
PRESSURE — LBS. PER SQ. IN.
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Table 1. Ammonia—Properties of Liquid and Saturated Vapor (Continued)

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per hr		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb cu ft $1/v_f$	cu ft lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
− 50	7.67	*14.3	43.49	33.08	−10.5	593.2	−0.0254	1.4487
− 49	7.91	13.8	43.45	32.12	− 9.4	593.7	− .0229	1.4463
− 48	8.16	13.3	43.41	31.20	− 8.4	594.2	− .0203	1.4438
− 47	8.42	12.8	43.37	30.31	− 7.3	594.6	− .0178	1.4413
− 46	8.68	12.2	43.33	29.45	− 6.3	595.0	− .0153	1.4388
− 45	8.95	*11.7	43.28	28.62	− 5.3	595.4	−0.0128	1.4363
− 44	9.23	11.1	43.24	27.82	− 4.3	595.9	− .0102	1.4338
− 43	9.51	10.6	43.20	27.04	− 3.2	596.4	− .0076	1.4314
− 42	9.81	10.0	43.16	26.29	− 2.1	596.8	− .0051	1.4290
− 41	10.10	9.3	43.12	25.56	− 1.1	597.2	− .0025	1.4266
− 40	10.41	*8.7	43.08	24.86	0.0	597.6	0.0000	1.4242
− 39	10.72	8.1	43.04	24.18	1.1	598.0	.0025	1.4217
− 38	11.04	7.4	42.99	23.53	2.1	598.3	.0051	1.4193
− 37	11.37	6.8	42.95	22.89	3.2	598.7	.0076	1.4169
− 36	11.71	6.1	42.90	22.27	4.3	599.1	.0101	1.4144
− 35	12.05	*5.4	42.86	21.68	5.3	599.5	0.0126	1.4120
− 34	12.41	4.7	42.82	21.10	6.4	599.9	.0151	1.4096
− 33	12.77	3.9	42.78	20.54	7.4	600.2	.0176	1.4072
− 32	13.14	3.2	42.73	20.00	8.5	600.6	.0201	1.4048
− 31	13.52	2.4	42.69	19.48	9.6	601.0	.0226	1.4025
− 30	13.90	*1.6	42.65	18.97	10.7	601.4	0.0250	1.4001
− 29	14.30	0.8	42.61	18.48	11.7	601.7	.0275	1.3978
− 28	14.71	0.0	42.57	18.00	12.8	602.1	.0300	1.3955
− 27	15.12	0.4	42.54	17.54	13.9	602.5	.0325	1.3932
− 26	15.55	0.8	42.48	17.09	14.9	602.8	.0350	1.3909
− 25	15.98	1.3	42.44	16.66	16.0	603.2	0.0374	1.3886
− 25	16.42	1.7	42.40	16.24	17.1	603.6	.0399	1.3863
− 23	16.88	2.2	42.35	15.83	18.1	603.9	.0423	1.3840
− 22	17.34	2.6	42.31	15.43	19.2	604.3	.0488	1.3818
− 21	17.81	3.1	42.26	15.05	20.3	604.6	.0472	1.3796
− 20	18.30	3.6	42.22	14.68	21.4	605.0	0.0497	1.3774
− 19	18.79	4.1	42.18	14.32	22.4	605.3	.0521	1.3752
− 18	19.30	4.6	42.13	13.97	23.5	605.7	.0545	1.3729
− 17	19.81	5.1	42.09	13.62	24.6	606.1	.0570	1.3708
− 16	20.34	5.6	42.04	13.29	25.6	606.4	.0594	1.3686
− 15	20.88	6.2	42.00	12.97	26.7	606.7	0.0618	1.3664
− 14	21.43	6.7	41.96	12.66	27.8	607.1	.0642	1.3643
− 13	21.99	7.8	41.91	12.36	28.9	607.5	.0666	1.3621
− 12	22.56	7.9	41.87	12.06	30.0	607.8	.0690	1.3600
− 11	23.15	8.5	41.82	11.78	31.0	608.1	.0714	1.3579
− 10	23.74	9.0	41.78	11.50	32.1	608.5	0.0738	1.3558
− 9	24.35	9.7	41.74	11.23	33.2	608.8	.0762	1.3537
− 8	24.97	10.3	41.69	10.97	34.3	609.2	.0768	1.3516
− 7	25.61	10.9	41.65	10.71	35.4	609.5	.0809	1.3495
− 6	26.26	11.6	41.60	10.47	36.4	609.8	.0833	1.3474
− 5	26.92	12.2	41.56	10.23	37.5	610.1	0.0857	1.3454
− 4	27.59	12.9	41.52	9.991	38.6	610.5	.0880	1.3433
− 3	28.28	13.6	41.47	9.763	39.7	610.8	.0904	1.3413
− 2	28.98	14.3	41.43	9.541	40.7	611.1	.0928	1.3393
− 1	29.69	15.0	41.38	9.326	14.8	611.4	.0951	1.3372
0	30.42	15.7	41.34	9.116	42.9	611.8	0.0975	1.3352
1	31.16	16.5	41.29	8.912	44.0	612.1	0.0998	1.3332
2	31.92	17.2	41.25	8.714	45.1	612.4	.1022	1.3312
3	32.69	18.0	41.20	8.521	46.2	612.7	.1045	1.3292
4	33.47	18.8	41.16	8.333	47.2	613.0	.1069	1.3273
5	34.27	19.6	41.11	8.150	48.3	613.3	.1092	1.3253
6	35.09	20.4	41.07	7.971	49.4	613.6	0.1115	1.3234
7	35.92	21.2	41.01	7.798	50.5	613.9	.1138	1.3214
8	36.77	22.1	40.98	8.629	51.6	614.3	.1162	1.3195
9	37.63	22.9	40.93	7.464	52.7	614.6	.1185	1.3176
10	38.51	23.8	40.89	7.304	53.8	614.9	.1208	1.3157
11	39.40	24.7	40.84	7.148	54.9	615.2	0.1231	1.3137
12	40.31	25.6	40.80	6.996	56.0	615.5	.1254	1.3118
13	41.24	26.5	40.75	6.847	57.1	615.8	.1277	1.3099
14	42.18	27.5	40.71	6.703	58.2	616.1	.1300	1.3081
15	43.14	28.4	40.66	6.562	59.2	616.3	.1323	1.3062

* Inches of mercury below one standard atmosphere (29.92 in.).

Table 1. Ammonia—Properties of Liquid and Saturated Vapor (Continued)

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
16	44.12	29.4	40.61	6.425	60.3	616.6	0.1346	1.3043
17	45.12	30.4	40.57	6.291	61.4	616.9	.1369	1.3025
18	46.13	31.4	40.52	6.161	62.5	617.2	.1392	1.3006
19	47.16	32.5	40.48	6.034	63.6	617.5	.1415	1.2988
20	48.21	33.5	40.43	5.910	64.7	617.8	.1437	1.2969
21	49.28	34.6	40.38	5.789	65.8	618.0	0.1460	1.2951
22	50.36	35.7	40.34	5.671	66.9	618.3	.1483	1.2933
23	51.47	36.8	40.29	5.556	68.0	618.6	.1505	1.2951
24	52.59	37.9	40.25	5.443	69.1	618.9	.1528	1.2897
25	53.78	39.0	40.20	5.334	70.2	619.1	.1551	1.2879
26	54.90	40.2	40.15	5.227	71.3	619.4	0.1573	1.2861
27	56.08	41.4	40.10	5.123	72.4	619.7	.1596	1.2843
28	57.28	42.6	40.06	5.021	73.5	619.9	.1618	1.2825
29	58.50	43.8	40.01	4.922	74.6	620.2	.1641	1.2808
30	59.74	45.0	39.96	4.825	75.7	620.5	.1663	1.2790
31	61.00	46.3	39.91	4.730	76.8	620.7	0.1686	1.2773
32	62.29	47.6	39.86	4.637	77.9	621.0	.1708	1.2755
33	63.59	48.9	39.82	4.547	79.0	621.2	.1730	1.2738
34	64.91	50.2	39.77	4.459	80.1	621.5	.1753	1.2731
35	66.26	52.6	39.72	4.373	81.2	621.7	.1775	1.2704
36	67.63	52.9	39.67	4.289	82.3	622.0	0.1797	1.2686
37	69.02	54.3	39.63	4.207	83.4	622.2	.1819	1.2669
38	70.43	55.7	39.58	4.126	84.6	622.5	.1841	1.2652
39	71.87	57.2	39.54	4.048	85.7	622.7	.1863	1.2635
40	73.32	58.6	39.49	3.971	86.8	623.0	.1885	1.2618
41	74.80	60.1	39.44	3.897	87.9	623.2	0.1908	1.2602
42	76.31	61.6	39.39	3.823	89.0	623.4	.1930	1.2585
43	77.83	63.1	39.34	3.752	90.1	623.7	.1952	1.2568
44	79.38	64.7	39.29	3.682	91.2	623.9	.1974	1.2552
45	80.96	66.3	39.24	3.614	92.3	624.1	.1996	1.2535
46	82.55	67.9	39.19	3.547	93.5	624.4	0.2018	1.2519
47	84.18	69.5	39.14	3.481	94.6	624.6	.2040	1.2502
48	85.82	71.1	39.10	3.418	95.7	624.8	.2062	1.2486
49	87.49	72.8	39.05	3.355	96.8	625.0	.2083	1.2469
50	89.19	74.5	39.00	3.294	97.9	625.2	.2105	1.2453
51	90.91	76.2	38.95	3.234	99.1	625.5	0.2127	1.2437
52	92.66	78.0	38.90	3.176	100.2	625.7	.2149	1.2421
53	94.43	79.7	38.85	3.119	101.3	625.9	.2171	1.2405
54	96.23	81.5	38.80	3.063	102.4	626.1	.2192	1.2389
55	98.06	83.4	38.75	3.008	103.5	626.3	.3214	1.2373
56	99.91	85.2	38.70	2.954	104.7	626.5	0.2236	1.2357
57	101.8	87.1	38.65	2.902	105.8	626.7	.2257	1.2341
58	103.7	89.0	38.60	2.851	106.9	626.9	.2279	1.2325
59	105.6	90.9	38.55	2.800	108.1	627.1	.2301	1.2310
60	107.6	92.9	38.50	2.751	109.2	627.3	.2322	1.2294
61	109.6	94.9	38.45	2.703	110.3	627.5	0.2344	1.2278
62	111.6	96.9	38.40	2.656	111.5	627.7	.2365	1.2262
63	113.6	98.9	38.35	2.610	112.6	627.9	.2387	1.2247
64	115.7	101.0	38.30	2.565	113.7	628.0	.2408	1.2231
65	117.8	103.1	38.25	2.520	114.8	628.2	.2430	1.2216
66	120.0	105.3	38.20	2.477	116.0	628.4	0.2451	1.2201
67	122.1	107.4	38.15	2.435	117.1	628.6	.2473	1.2186
68	124.3	109.6	38.10	2.393	118.3	628.8	.2494	1.2170
69	126.5	111.8	38.05	2.352	119.4	628.9	.2515	1.2155
70	128.8	114.1	38.00	2.312	120.5	629.1	.2537	1.2140
71	131.1	116.4	37.95	2.273	121.7	629.3	0.2558	1.2125
72	133.4	118.7	37.90	2.235	122.8	629.4	.2579	1.2110
73	135.7	121.0	37.84	2.197	124.0	629.6	.2601	1.2095
74	138.1	123.4	37.79	2.161	125.1	629.8	.2622	1.2080
75	140.5	125.8	37.74	2.125	126.2	629.9	.2643	1.2065
76	143.0	128.3	37.69	2.089	127.4	630.1	0.2664	1.2050
77	145.4	130.7	37.64	2.055	128.5	630.2	.2685	1.2035
78	147.9	133.2	37.58	2.021	129.7	630.4	.2706	1.2020
79	150.5	135.8	37.53	1.988	130.8	630.5	.2728	1.2006
80	153.0	138.3	37.48	1.955	132.0	630.7	.2749	1.1991

Table 1. Ammonia—Properties of Liquid and Saturated Vapor (Concluded)

Temp F t	Pressure		Liquids, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per hr		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
81	155.6	140.9	37.43	1.923	133.1	630.8	0.2769	1.1976
82	158.3	143.6	37.37	1.892	134.3	631.0	.2791	1.1962
83	161.0	146.3	37.32	1.861	135.4	631.1	.2812	1.1947
84	163.6	149.0	37.26	1.831	136.6	631.3	.2833	1.1933
85	166.4	151.7	37.21	1.801	137.8	631.4	.2854	1.1918
86	169.2	154.5	37.16	1.772	138.9	631.5	0.2875	1.1904
87	172.0	157.3	37.11	1.744	140.1	631.7	.2895	1.1889
88	174.8	160.1	37.05	1.716	141.2	631.8	.2917	1.1875
89	177.7	163.0	37.00	1.688	142.4	631.9	.2937	1.1860
90	180.6	165.9	36.95	1.661	143.5	632.0	.2958	1.1846
91	183.6	168.9	36.89	1.635	144.7	632.1	0.2979	1.1832
92	186.6	171.9	36.84	1.609	145.8	632.3	.3000	1.1818
93	189.6	174.9	36.78	1.584	147.0	632.3	.3021	1.1804
94	192.7	178.0	36.73	1.559	148.2	632.5	.3041	1.1789
95	195.8	181.1	36.67	1.534	149.4	632.6	.3062	1.1775
96	198.9	184.2	36.62	1.510	150.5	632.6	0.3083	1.1761
97	202.1	187.4	36.56	1.487	151.7	632.8	.3104	1.1747
98	205.3	190.6	36.51	1.464	152.9	632.9	.3125	1.1733
99	208.6	193.9	36.45	1.441	154.0	632.9	.3145	1.1719
100	211.9	197.2	36.40	1.419	155.2	633.0	.3166	1.1705
101	215.2	200.5	36.34	1.397	156.4	633.1	0.3187	1.1691
102	218.6	203.9	36.29	1.375	157.6	633.2	.3207	1.1677
103	222.0	207.3	36.23	1.354	158.7	633.3	.3228	1.1663
104	224.4	210.7	36.18	1.334	159.9	633.4	.3248	1.1649
105	228.9	214.2	36.12	1.313	161.1	633.4	.3269	1.1635
106	232.5	217.8	36.06	1.293	162.3	633.5	0.3289	1.1621
107	236.0	221.3	36.01	1.274	163.5	633.6	.3310	1.1607
108	239.7	225.0	35.95	1.254	164.6	633.6	.3330	1.1593
109	243.3	228.6	35.90	1.235	165.8	633.7	.3351	1.1580
110	247.0	232.3	35.84	1.217	167.0	633.7	.3372	1.1566
111	250.8	236.1	35.78	1.198	168.2	633.8	0.3392	1.1552
112	354.5	239.8	35.72	1.180	169.4	633.8	.3413	1.1538
113	258.4	243.7	35.67	1.163	170.6	633.9	.3433	1.1524
114	262.2	247.5	35.61	1.145	171.8	633.9	.3453	1.1510
115	266.2	251.5	35.55	1.128	173.0	633.9	.3474	1.1497
116	270.1	255.4	35.49	1.112	174.2	634.0	0.3495	1.1483
118	274.1	259.4	35.43	1.095	175.4	634.0	.3515	1.1469
118	278.2	263.5	35.38	1.079	176.6	634.0	.3515	1.1455
119	282.3	267.6	35.32	1.063	177.8	634.0	.3556	1.1441
120	286.4	271.7	35.26	1.047	179.0	634.0	.3576	1.1427
121	290.6	275.9	35.20	1.032	180.2	634.0	0.3597	1.1414
122	294.8	280.1	35.14	1.017	181.4	634.0	.3618	1.1400
123	299.1	284.4	35.08	1.002	182.6	634.0	.3638	1.1386
124	303.4	288.7	35.02	0.987	183.9	634.0	.3659	1.1372
125	307.8	293.1	34.96	0.973	185.1	634.0	.3679	1.1358

Table 2. Ammonia—Specific Volume of Superheated Vapor (cu ft per lb)

Temp, deg F	Pressure in psia							Saturation temperature in italics (deg F)						
	5	6	7	8	9	10	11	12	13	14	15	16	17	1
	−63.11	*−57.64*	*−52.88*	*−48.64*	*−44.38*	*−41.34*	*−38.14*	*−35.16*	*−32.37*	*−29.76*	*−27.29*	*−24.95*	*−22.73*	*−2*
(*at sat*)	(*49.31*)	(*41.59*)	(*36.01*)	(*31.79*)	(*28.48*)	(*25.81*)	(*23.61*)	(*21.77*)	(*20.20*)	(*18.85*)	(*17.67*)	(*16.64*)	(*15.72*)	(*14*
−50	51.05	42.44	36.29	—	—	—	—	—	—	—	—	—	—	
−40	52.36	43.55	37.25	32.52	28.85	25.90	—	—	—	—	—	—	—	
−30	53.67	44.64	38.19	33.36	29.59	26.58	24.12	22.07	20.33	—	—	—	—	
−20	54.97	45.73	39.13	34.19	30.34	27.26	24.74	22.64	20.86	19.33	18.01	16.86	15.83	14.
−10	56.26	46.82	40.07	35.01	31.07	27.92	25.35	23.20	21.38	19.82	18.47	17.29	16.24	15.
0	57.55	47.90	41.00	35.83	31.80	28.58	25.95	23.75	21.90	20.30	18.92	17.72	16.65	15.
10	58.84	48.98	41.93	36.64	32.53	29.24	26.55	24.31	22.41	20.78	19.37	18.14	17.05	16.
20	60.12	50.05	42.85	37.45	33.26	29.90	27.15	24.86	22.92	21.26	19.82	18.56	17.45	16.
30	61.41	51.12	43.77	38.26	33.98	30.55	27.74	25.41	23.43	21.73	20.26	18.97	17.84	16.
40	62.69	52.19	44.69	39.07	34.70	31.20	28.34	25.95	23.93	22.20	20.70	19.39	18.23	17.
50	63.96	53.26	45.61	39.88	35.42	31.85	28.93	26.49	24.43	22.67	21.14	19.80	18.62	17.
60	65.24	54.32	46.53	40.68	36.13	32.49	29.52	27.03	24.94	23.14	21.58	20.21	19.01	17.
70	66.51	55.39	47.44	41.48	36.85	33.14	30.10	27.57	25.43	23.60	22.01	20.62	19.39	18.
80	67.79	56.45	48.36	42.28	37.56	33.78	30.69	28.11	25.93	24.06	22.44	21.03	19.78	18.
90	69.06	57.51	49.27	43.08	38.27	34.42	31.28	28.65	26.43	24.53	22.88	21.43	20.16	19.
100	70.33	58.58	50.18	43.88	38.98	35.07	31.86	29.19	26.93	24.99	23.31	21.84	20.54	19.
110	71.60	59.64	51.09	44.68	39.70	35.71	32.44	29.72	27.42	25.45	23.74	22.24	20.92	19.
120	72.87	60.70	52.00	45.48	40.40	36.35	33.03	30.26	27.92	25.91	24.17	22.65	21.30	20.
130	74.14	61.76	52.91	46.27	41.11	36.99	33.61	30.79	28.41	26.37	24.60	23.05	21.68	20.
140	75.41	62.82	53.82	47.07	41.82	37.62	34.19	31.33	28.90	26.85	25.03	23.45	22.06	20.
150	76.68	63.87	54.73	47.87	42.53	38.26	34.77	31.86	29.40	27.29	25.46	23.86	22.44	21.
160	77.95	64.93	55.63	48.66	43.24	38.90	35.35	32.39	29.89	27.74	25.88	24.26	22.82	21.
170	79.21	65.99	56.54	49.46	43.95	39.54	35.93	32.92	30.38	28.20	26.31	24.66	23.20	21.
180	80.48	67.05	57.45	50.25	44.65	40.17	36.51	33.46	30.87	28.66	26.74	25.06	23.58	22.
190	—	—	—	—	—	40.81	37.09	33.99	31.36	29.11	27.16	25.46	23.95	22.
200	—	—	—	—	—	41.45	37.67	34.52	31.85	29.57	27.59	25.86	24.33	22.
220	—	—	—	—	—	—	—	—	—	—	28.44	26.66	25.08	23.

Temp, deg F	Pressure in psia							Saturation temperature in italics (deg F)						
	18	19	20	21	22	23	24	25	26	27	28	29	30	31
	−20.61	*−18.58*	*−16.64*	*−14.78*	*−12.98*	*−11.25*	*−9.58*	*−7.96*	*−6.39*	*−4.87*	*−3.40*	*−1.97*	*−0.57*	*+0.*
(*at sat*)	(*14.90*)	(*14.17*)	(*13.50*)	(*12.90*)	(*12.35*)	(*11.85*)	(*11.39*)	(*10.98*)	(*10.56*)	(*10.20*)	(*9.853*)	(*9.534*)	(*9.236*)	(*8.9*
−20	14.93	—	—	—	—	—	—	—	—	—	—	—	—	
−10	15.32	14.49	13.74	13.06	12.45	11.89	—	—	—	—	—	—	—	
0	15.70	14.85	14.09	13.40	12.77	12.20	11.67	11.19	10.74	10.33	9.942	9.584	9.250	
10	16.08	15.21	14.44	13.73	13.09	12.50	11.96	11.47	11.01	10.59	10.20	9.834	9.492	9.
20	16.46	15.57	14.78	14.06	13.40	12.80	12.25	11.75	11.28	10.85	10.45	10.08	9.731	9.
30	16.83	15.93	15.11	14.38	13.71	13.10	12.54	12.03	11.55	11.11	10.70	10.32	9.966	9.
40	17.20	16.28	15.45	14.70	14.02	13.40	12.82	12.30	11.81	11.37	10.95	10.56	10.20	9.
50	17.57	16.63	15.78	15.02	14.32	13.69	13.11	12.57	12.08	11.62	11.19	10.80	10.43	10.
60	17.94	16.98	16.12	15.34	14.63	13.98	13.39	12.84	12.34	11.87	11.44	11.03	10.65	10.
70	18.30	17.33	16.45	15.65	14.93	14.27	13.66	13.11	12.59	12.12	11.68	11.26	10.88	10.
80	18.67	17.67	16.78	15.97	15.23	14.56	13.94	13.37	12.85	12.37	11.92	11.50	11.10	10.
90	19.03	18.02	17.10	16.28	15.53	14.84	14.22	13.64	13.11	12.61	12.15	11.73	11.33	10.
100	19.39	18.36	17.43	16.59	15.83	15.13	14.49	13.90	13.36	12.86	12.39	11.96	11.55	11.
110	19.75	18.70	17.76	16.90	16.12	15.41	14.76	14.17	13.61	13.10	12.63	12.18	11.77	11.
120	20.11	19.04	18.08	17.21	16.42	15.70	15.04	14.43	13.87	13.34	12.86	12.41	11.99	11.
130	20.47	19.38	18.41	17.52	16.72	15.98	15.31	14.69	14.12	13.59	13.10	12.64	12.21	11.
140	20.83	19.72	18.73	17.83	17.01	16.26	15.58	14.95	14.37	13.83	13.33	12.86	12.43	12.
150	21.19	20.06	19.05	18.14	17.31	16.55	15.85	15.21	14.62	14.07	13.56	13.09	12.65	12.
160	21.54	20.40	19.37	18.44	17.60	16.83	16.12	15.47	14.87	14.31	13.80	13.31	12.87	12.
170	21.90	20.74	19.70	18.75	17.89	17.11	16.39	15.73	15.12	14.55	14.03	13.54	13.08	12.
180	22.26	21.08	20.02	19.06	18.19	17.39	16.66	15.99	15.37	14.79	14.26	13.76	13.30	12.
190	22.61	21.42	20.34	19.36	18.48	17.67	16.93	16.25	15.62	15.03	14.49	13.99	13.52	13.
200	22.97	21.75	20.66	19.67	18.77	17.95	17.20	16.50	15.86	15.27	14.72	14.21	13.73	13.
220	23.68	22.43	21.30	20.28	19.35	18.51	17.73	17.02	16.36	15.75	15.18	14.65	14.16	13.
240	—	—	21.94	20.89	19.94	19.07	18.27	17.53	16.85	16.23	15.64	15.10	14.59	14.
260	—	—	—	—	—	—	—	18.04	17.35	16.70	16.10	15.54	15.02	14.
280	—	—	—	—	—	—	—	—	—	—	—	—	15.45	14.

Table 2. Ammonia—Specific Volume of Superheated Vapor (cu ft per lb) (Continued)

mp, F	Pressure in psia							Saturation temperature in italics (deg F)						
	31	32	33	34	35	36	37	38	39	40	42	44	46	48
	0.79	*2.11*	*3.40*	*4.66*	*5.89*	*7.09*	*8.27*	*9.42*	*10.55*	*11.66*	*13.81*	*15.88*	*17.87*	*19.80*
sat)	(8.955)	(8.698)	(8.445)	(8.211)	(7.991)	(7.782)	(7.584)	(7.396)	(7.217)	(7.047)	(6.731)	(6.442)	(6.177)	(5.934)
0	9.173	8.874	8.592	8.328	8.078	7.842	7.619	7.407	—	—	—	—	—	—
0	9.405	9.099	8.812	8.542	8.287	8.046	7.819	7.603	7.398	7.203	6.842	6.513	6.213	5.937
0	9.633	9.321	9.028	8.753	8.493	8.247	8.015	7.795	7.586	7.387	7.019	6.683	6.377	6.096
0	9.858	9.540	9.242	8.960	8.695	8.445	8.208	7.983	7.770	7.568	7.192	6.850	6.538	6.251
0	10.08	9.757	9.452	9.166	8.895	8.640	8.398	8.170	7.952	7.746	7.363	7.014	6.696	6.404
0	10.30	9.972	9.661	9.369	9.093	8.833	8.587	8.353	8.132	7.922	7.531	7.176	6.851	6.554
0	10.52	10.18	9.868	9.570	9.289	9.024	8.773	8.535	8.310	8.096	7.697	7.336	7.005	6.702
0	10.74	10.40	10.07	9.770	9.484	9.214	8.958	8.716	8.486	8.268	7.862	7.494	7.157	6.848
0	10.96	10.61	10.28	9.969	9.677	9.402	9.142	8.895	8.661	8.439	8.026	7.650	7.308	6.993
0	11.17	10.81	10.48	10.17	9.869	9.589	9.324	9.073	8.835	8.609	8.188	7.806	7.457	7.137
0	11.38	11.02	10.68	10.36	10.06	9.775	9.506	9.250	9.008	8.777	8.349	7.960	7.605	7.280
0	11.60	11.23	10.88	10.56	10.25	9.961	9.686	9.426	9.179	8.945	8.510	8.114	7.753	7.421
0	11.81	11.44	11.08	10.75	10.44	10.15	9.866	9.602	9.351	9.112	8.669	8.267	7.899	7.562
0	12.02	11.64	11.28	10.95	10.63	10.33	10.05	9.776	9.521	9.278	8.828	8.419	8.045	7.702
0	12.23	11.85	11.48	11.14	10.82	10.51	10.22	9.950	9.691	9.444	8.986	8.570	8.190	7.842
0	12.44	12.05	11.68	11.33	11.00	10.69	10.40	10.12	9.860	9.609	9.144	8.721	8.335	7.981
0	12.66	12.26	11.88	11.53	11.19	10.88	10.58	10.30	10.03	9.774	9.301	8.871	8.479	8.119
0	12.87	12.46	12.08	11.72	11.38	11.06	10.76	10.47	10.20	9.938	9.458	9.021	8.623	8.257
0	13.07	12.66	12.27	11.91	11.56	11.24	10.93	10.64	10.36	10.10	9.614	9.171	8.766	8.395
0	13.28	12.86	12.47	12.10	11.75	11.42	11.11	10.81	10.53	10.27	9.770	9.320	8.909	8.532
0	13.70	13.27	12.86	12.48	12.12	11.78	11.46	11.16	10.87	10.59	10.08	9.617	9.194	8.805
0	14.12	13.67	13.26	12.86	12.49	12.14	11.81	11.50	11.20	10.92	10.39	9.913	9.477	9.077
0	14.53	14.08	13.65	13.24	12.86	12.50	12.16	11.84	11.53	11.24	10.70	10.21	9.760	9.348
0	14.95	14.48	14.04	13.62	13.23	12.86	12.51	12.18	11.86	11.56	11.01	10.50	10.04	9.619
0	—	—	—	—	—	—	—	—	—	11.88	11.31	10.80	10.32	9.888

mp, g F	Pressure in psia							Saturation temperature in italics (deg F)						
	48	50	52	54	56	58	60	62	64	66	68	70	75	80
	19.80	*21.67*	*23.48*	*25.28*	*26.94*	*28.59*	*30.21*	*31.78*	*33.31*	*34.81*	*36.27*	*37.70*	*41.13*	*44.40*
sat)	(5.934)	(5.710)	(5.502)	(5.309)	(5.129)	(4.962)	(4.805)	(4.658)	(4.519)	(4.389)	(4.267)	(4.151)	(3.887)	(3.655)
20	5.937	—	—	—	—	—	—	—	—	—	—	—	—	—
30	6.096	5.838	5.599	5.378	5.172	4.981	—	—	—	—	—	—	—	—
40	6.251	5.988	5.744	5.519	5.310	5.115	4.933	4.762	4.602	4.452	4.310	4.177	—	—
50	6.404	6.135	5.887	5.657	5.444	5.245	5.060	4.886	4.723	4.570	4.426	4.290	3.982	3.712
60	6.554	6.280	6.027	5.793	5.576	5.373	5.184	5.007	4.842	4.686	4.539	4.401	4.087	3.812
70	6.702	6.423	6.165	5.927	5.706	5.499	5.307	5.127	4.958	4.799	4.650	4.509	4.189	3.909
80	6.848	6.564	6.302	6.059	5.834	5.624	5.428	5.244	5.072	4.910	4.758	4.615	4.289	4.005
90	6.993	6.704	6.437	6.190	5.960	5.746	5.547	5.360	5.185	5.020	4.865	4.719	4.388	4.098
00	7.137	6.843	6.571	6.319	6.085	5.868	5.665	5.474	5.296	4.129	4.971	4.822	4.485	4.190
10	7.280	6.980	6.704	6.447	6.209	5.988	5.781	5.588	5.406	5.236	5.075	4.924	4.581	4.281
20	7.421	7.117	6.835	6.575	6.333	6.107	5.897	5.700	5.516	5.342	5.179	5.025	4.676	4.371
30	7.562	7.252	6.966	6.701	6.455	6.226	6.012	5.811	5.624	5.447	5.281	5.125	4.770	4.460
40	7.702	7.387	7.096	6.827	6.576	6.343	6.126	5.922	5.731	5.552	5.383	5.224	4.863	4.548
50	7.842	7.521	7.225	6.952	6.697	6.460	6.239	6.032	5.838	5.656	5.484	5.323	4.956	4.635
60	7.981	7.655	7.354	7.076	6.817	6.577	6.352	6.142	5.944	5.759	5.585	5.420	5.048	4.722
70	8.119	7.788	7.483	7.200	6.937	6.692	6.464	6.250	6.050	5.862	5.685	5.518	5.139	4.808
80	8.257	7.921	7.611	7.323	7.056	6.808	6.576	6.359	6.155	5.964	5.784	5.615	5.230	4.893
90	8.395	8.053	7.738	7.446	7.175	6.923	6.687	6.467	6.260	6.066	5.883	5.711	5.320	4.978
00	8.532	8.185	7.865	7.569	7.294	7.037	6.787	6.574	6.364	6.167	5.982	5.807	5.410	5.063
10	8.669	8.317	7.992	7.691	7.412	7.151	6.909	6.681	6.468	6.268	6.080	5.902	5.500	5.147
20	8.805	8.448	8.118	7.813	7.529	7.265	7.019	6.788	6.572	6.369	6.179	5.998	5.589	5.231
30	8.941	8.579	8.244	7.935	7.647	7.379	7.129	6.895	6.675	6.470	6.275	6.093	5.678	5.315
40	9.077	8.710	8.370	8.056	7.764	7.492	7.238	7.001	6.778	6.570	6.373	6.187	5.767	5.398
50	9.213	8.840	8.496	8.177	7.881	7.605	7.338	7.107	6.881	6.670	6.470	6.282	5.855	5.482
60	9.348	8.970	8.621	8.298	7.998	7.718	7.437	7.213	6.984	6.769	6.567	6.376	5.943	5.565
80	9.619	9.230	8.871	8.539	8.230	7.943	7.676	7.424	7.188	6.968	6.760	6.563	6.119	5.730
00	9.888	9.489	9.120	8.779	8.462	8.167	7.892	7.634	7.392	7.165	6.952	6.750	6.294	5.894

Table 2. Ammonia—Specific Volume of Superheated Vapor (cu ft per lb) (Continued)

Temp, deg F	Pressure in psia						Saturation temperature in italics (deg F)							
	80	85	90	95	100	105	110	115	120	125	130	135	140	145
	44.40	*47.50*	*50.47*	*53.32*	*56.05*	*58.67*	*61.21*	*63.65*	*66.02*	*68.31*	*70.53*	*72.69*	*74.79*	*76.8*
(at sat)	(3.655)	(3.449)	(3.266)	(3.101)	(2.952)	(2.817)	(2.693)	(2.580)	(2.476)	(2.380)	(2.291)	(2.209)	(2.132)	(2.06
50	3.712	3.473	—	—	—	—	—	—	—	—	—	—	—	—
60	3.812	3.569	3.353	3.160	2.985	2.828	—	—	—	—	—	—	—	—
70	3.909	3.662	3.442	3.245	3.068	2.907	2.761	2.628	2.505	2.392	—	—	—	—
80	4.005	3.753	3.529	3.329	3.149	2.985	2.837	2.701	2.576	2.461	2.355	2.257	2.166	2.08
90	4.098	3.842	3.614	3.411	3.227	3.061	2.910	2.772	2.645	2.528	2.421	2.321	2.228	2.14
100	4.190	3.930	3.698	3.491	3.304	3.135	2.981	2.841	2.712	2.593	2.484	2.382	2.288	2.20
110	4.281	4.016	3.780	3.570	3.380	3.208	3.051	2.909	2.778	2.657	2.546	2.442	2.347	2.25
120	4.371	4.101	3.862	3.647	3.454	3.279	3.120	2.975	2.842	2.719	2.606	2.501	2.404	2.31
130	4.460	4.186	3.942	3.724	3.527	3.350	3.188	3.040	2.905	2.780	2.665	2.559	2.460	2.36
140	4.548	4.269	4.021	3.799	3.600	3.419	3.255	3.105	2.967	2.840	2.724	2.615	2.515	2.42
150	4.635	4.352	4.100	3.874	3.672	3.488	3.321	3.168	3.029	2.900	2.781	2.671	2.569	2.47
160	4.722	4.434	4.178	3.949	3.743	3.556	3.386	3.231	3.089	2.958	2.838	2.726	2.622	2.52
170	4.808	4.515	4.255	4.022	3.813	3.623	3.451	3.294	3.149	3.016	2.894	2.780	2.675	2.57
180	4.893	4.596	4.332	4.096	3.883	3.690	3.515	3.355	3.209	3.074	2.949	2.834	2.727	2.62
190	4.978	4.677	4.408	4.168	3.952	3.757	3.579	3.417	3.268	3.131	3.004	2.887	2.779	2.67
200	5.063	4.757	4.484	4.241	4.021	3.823	3.642	3.477	3.326	3.187	3.059	2.940	2.830	2.72
210	5.147	4.836	4.560	4.313	4.090	3.888	3.705	3.538	3.385	3.243	3.113	2.992	2.880	2.77
220	5.231	4.916	4.635	4.384	4.158	3.954	3.768	3.598	3.442	3.299	3.167	3.044	2.931	2.82
230	5.315	4.995	4.710	4.455	4.226	4.019	3.830	3.658	3.500	3.354	3.220	3.096	2.981	2.87
240	5.398	5.074	4.785	4.526	4.294	4.083	3.892	3.717	3.557	3.409	3.273	3.147	3.030	2.92
250	5.482	5.152	4.859	4.597	4.361	4.148	3.954	3.776	3.614	3.464	3.326	3.198	3.080	2.96
260	5.565	5.230	4.933	4.668	4.428	4.212	4.015	3.835	3.671	3.519	3.379	3.249	3.129	3.01
270	5.648	5.308	5.007	4.738	4.495	4.276	4.076	3.894	3.727	3.573	3.431	3.300	3.179	3.06
280	5.730	5.386	5.081	4.808	4.562	4.340	4.137	3.952	3.783	3.627	3.483	3.350	3.227	3.11
290	5.812	5.464	5.155	4.878	4.629	4.403	4.198	4.011	3.839	3.681	3.535	3.400	3.275	3.15
300	5.894	5.541	5.228	4.947	4.695	4.466	4.259	4.069	3.895	3.735	3.587	3.450	3.323	3.20
320	—	—	—	—	—	—	—	—	—	3.842	3.690	3.550	3.420	3.29

Table 2. Ammonia—Specific Volume of Superheated Vapor (cu ft per lb) (Concluded)

Temp, deg F	Pressure in psia						Saturation temperature in italics (deg F)										
	145	150	160	170	180	190	200	210	220	230	240	250	260	270	280	290	300
	76.83	*78.81*	*82.64*	*86.29*	*89.78*	*93.13*	*96.34*	*99.43*	*102.42*	*105.30*	*108.09*	*110.80*	*113.42*	*115.97*	*118.45*	*120.86*	*123.21*
(at sat)	(2.061)	(1.994)	(1.872)	(1.764)	(1.667)	(1.581)	(1.502)	(1.431)	(1.367)	(1.307)	(1.253)	(1.202)	(1.155)	(1.112)	(1.072)	(1.034)	(0.999)
80	2.080	2.001	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
90	2.141	2.061	1.914	1.784	1.668	—	—	—	—	—	—	—	—	—	—	—	—
100	2.200	2.118	1.969	1.837	1.720	1.615	1.520	1.433	—	—	—	—	—	—	—	—	—
110	2.257	2.174	2.023	1.889	1.770	1.663	1.567	1.480	1.400	1.328	1.261	—	—	—	—	—	—
120	2.313	2.228	2.075	1.939	1.818	1.710	1.612	1.524	1.443	1.370	1.302	1.240	1.182	1.128	1.078	—	—
130	2.368	2.281	2.125	1.988	1.865	1.755	1.656	1.566	1.485	1.410	1.342	1.278	1.220	1.166	1.115	1.068	1.023
140	2.421	2.334	2.175	2.035	1.910	1.799	1.698	1.608	1.525	1.449	1.380	1.316	1.257	1.202	1.151	1.103	1.058
150	2.474	2.385	2.224	2.081	1.955	1.842	1.740	1.648	1.564	1.487	1.416	1.352	1.292	1.236	1.184	1.136	1.091
160	2.526	2.435	2.272	2.127	1.999	1.884	1.780	1.687	1.601	1.524	1.452	1.386	1.326	1.269	1.217	1.168	1.123
170	2.577	2.485	2.319	2.172	2.042	1.925	1.820	1.725	1.638	1.559	1.487	1.420	1.359	1.302	1.249	1.199	1.153
180	2.627	2.534	2.365	2.216	2.084	1.966	1.859	1.762	1.675	1.594	1.521	1.453	1.391	1.333	1.279	1.229	1.183
190	2.677	2.583	2.411	2.260	2.126	2.005	1.897	1.799	1.710	1.629	1.554	1.486	1.422	1.364	1.309	1.259	1.211
200	2.727	2.631	2.457	2.303	2.167	2.045	1.935	1.836	1.745	1.663	1.587	1.518	1.453	1.394	1.339	1.287	1.239
210	2.776	2.679	2.502	2.346	2.208	2.084	1.972	1.872	1.780	1.696	1.619	1.549	1.484	1.423	1.367	1.315	1.267
220	2.825	2.726	2.547	2.389	2.248	2.123	2.009	1.907	1.814	1.729	1.651	1.580	1.514	1.452	1.396	1.343	1.294
230	2.873	2.773	2.591	2.431	2.288	2.161	2.046	1.942	1.848	1.762	1.683	1.610	1.543	1.481	1.424	1.370	1.320
240	2.921	2.820	2.635	2.473	2.328	2.199	2.082	1.977	1.881	1.794	1.714	1.640	1.572	1.509	1.451	1.397	1.346
250	2.969	2.866	2.679	2.514	2.467	2.236	2.118	2.011	1.914	1.826	1.745	1.670	1.601	1.537	1.478	1.423	1.372
260	3.017	2.912	2.723	2.555	2.407	2.274	2.154	2.046	1.947	1.857	1.775	1.699	1.630	1.565	1.505	1.449	1.397
270	3.064	2.958	2.766	2.596	2.446	2.311	2.189	2.080	1.980	1.889	1.805	1.729	1.658	1.592	1.532	1.475	1.422
280	3.111	3.004	2.809	2.637	2.484	2.348	2.225	2.113	2.012	1.920	1.835	1.758	1.686	1.620	1.558	1.501	1.447
290	3.158	3.049	2.852	2.678	2.523	2.384	2.260	2.147	2.044	1.951	1.865	1.786	1.714	1.646	1.584	1.526	1.472
300	3.205	3.095	2.895	2.718	2.561	2.421	2.295	2.180	2.076	1.982	1.895	1.815	1.741	1.673	1.610	1.551	1.496
320	3.298	3.185	2.980	2.798	2.637	2.493	2.364	2.246	2.140	2.043	1.954	1.872	1.796	1.726	1.661	1.601	1.544
340	—	—	3.064	2.878	2.713	2.565	2.432	2.312	2.203	2.103	2.012	1.928	1.850	1.778	1.712	1.650	1.592
360	—	—	—	—	—	—	2.500	2.377	2.265	2.163	2.069	1.983	1.904	1.830	1.762	1.698	1.639
380	—	—	—	—	—	—	2.568	2.442	2.327	2.222	2.126	2.038	1.957	1.881	1.811	1.747	1.686
400	—	—	—	—	—	—	—	—	—	—	—	2.093	2.009	1.932	1.861	1.794	1.732

Table 3. Ammonia—Enthalpy of Superheated Vapor (Btu per lb)

Temp, deg F	Pressure in psia						Saturation temperature in italics (deg F)							
	5	6	7	8	9	10	11	12	13	14	15	16	17	18
	−63.11	*−57.64*	*−52.88*	*−48.64*	*−44.88*	*−41.34*	*−38.14*	*−35.16*	*−32.37*	*−29.76*	*−27.29*	*−24.95*	*−22.73*	*−20*
(*at sat*)	(*588.3*)	(*590.6*)	(*592.5*)	(*594.2*)	(*595.7*)	(*597.1*)	(*598.3*)	(*599.4*)	(*600.5*)	(*601.4*)	(*602.4*)	(*603.2*)	(*604.0*)	(*604*
−50	595.2	594.6	594.0	—	—	—	—	—	—	—	—	—	—	—
−40	600.3	599.8	599.3	598.8	598.3	597.8	—	—	—	—	—	—	—	—
−30	605.4	604.9	604.5	604.1	603.6	603.2	602.7	602.3	601.8	—	—	—	—	—
−20	610.4	610.0	609.6	609.3	608.9	608.5	608.1	607.7	607.2	606.8	606.4	606.0	605.6	605
−10	615.4	615.1	614.7	614.4	614.0	613.7	613.3	613.0	612.6	612.2	611.9	611.5	611.1	610
0	620.4	620.1	619.8	619.5	619.2	618.9	618.5	618.2	617.9	617.6	617.2	616.9	616.6	616
10	625.4	625.2	624.9	624.6	624.3	624.0	623.7	623.4	623.1	622.8	622.5	622.2	621.9	621
20	630.4	630.2	629.9	629.7	629.4	629.1	628.9	628.6	628.3	628.0	627.8	627.5	627.2	626
30	635.4	635.2	635.0	634.7	634.5	634.2	634.0	633.7	633.5	633.2	633.0	632.7	632.5	632
40	640.4	640.2	640.0	639.8	639.5	639.3	639.1	638.9	638.6	638.4	638.2	638.0	637.7	637
50	645.5	645.2	645.0	644.8	644.6	644.4	644.2	644.0	643.8	643.6	643.4	643.2	642.9	642
60	650.5	650.3	650.1	649.9	649.7	649.5	649.3	649.1	648.9	648.7	648.5	648.3	648.1	647
70	655.5	655.3	655.2	655.0	654.8	654.6	654.4	654.3	654.1	653.9	653.7	653.5	653.3	653
80	660.6	660.4	660.2	660.1	659.9	659.7	659.6	659.4	659.2	659.0	658.9	658.7	658.5	658
90	665.6	665.5	665.3	665.2	665.0	664.8	664.7	664.5	664.4	664.2	664.0	663.9	663.7	663
100	670.7	670.6	670.4	670.3	670.1	670.0	669.8	669.7	669.5	669.4	669.2	669.1	668.9	668
110	675.8	675.7	675.5	675.4	675.3	675.1	675.0	674.8	674.7	674.5	674.4	674.3	674.1	674
120	680.9	680.8	680.7	680.5	680.4	680.3	680.1	680.0	679.9	679.7	679.6	679.5	679.3	679
130	686.1	685.9	685.8	685.7	685.6	685.4	685.3	685.2	685.1	684.9	684.8	684.7	684.5	684
140	691.2	691.1	691.0	690.9	690.7	690.6	690.5	690.4	690.3	690.1	690.0	689.9	689.8	689
150	696.4	696.3	696.2	696.1	695.9	695.8	695.7	695.6	595.5	695.4	695.3	695.1	695.0	694
160	701.6	701.5	701.4	701.3	701.2	701.1	700.9	700.8	700.7	700.6	700.5	700.4	700.3	700
170	706.8	706.7	706.6	706.5	706.4	706.3	706.2	706.1	706.0	705.9	705.8	705.7	705.6	705
180	712.1	712.0	711.9	711.8	711.7	711.6	711.5	711.4	711.3	711.2	711.1	711.0	710.9	710
190	—	—	—	—	—	716.9	716.8	716.7	716.6	716.5	716.4	716.3	716.2	716
200	—	—	—	—	—	722.2	722.1	722.0	721.9	721.8	721.7	721.6	721.5	721
220	—	—	—	—	—	—	—	—	—	—	—	732.3	732.2	732

Temp, deg F	Pressure in psia							Saturation temperature in italics (deg F)						
	18	19	20	21	22	23	24	25	26	27	28	29	30	31
	−20.61	*−18.58*	*−16.64*	*−14.78*	*−12.98*	*−11.25*	*−9.58*	*−7.96*	*−6.39*	*−4.87*	*−3.40*	*−1.97*	*−0.57*	*+*
(*at sat*)	(*604.8*)	(*605.5*)	(*606.2*)	(*606.8*)	(*607.4*)	(*608.1*)	(*608.6*)	(*609.1*)	(*609.7*)	(*610.2*)	(*610.7*)	(*611.1*)	(*611.6*)	(*612*
−20	605.1	—	—	—	—	—	—	—	—	—	—	—	—	—
−10	610.7	610.3	610.0	609.6	609.2	608.8	—	—	—	—	—	—	—	—
0	616.2	615.9	615.5	615.2	614.8	614.5	614.1	613.8	613.4	613.0	612.7	612.3	611.9	—
10	621.6	621.3	621.0	620.7	620.4	620.0	619.7	619.4	619.1	618.8	618.4	618.1	617.8	617
20	626.9	626.7	626.4	626.1	625.8	625.5	625.2	625.0	624.7	624.4	624.1	623.8	623.5	623
30	632.2	632.0	631.7	631.5	631.2	630.9	630.7	630.4	630.2	629.9	629.6	629.4	629.1	628
40	637.5	637.3	637.0	636.8	636.6	636.3	636.1	635.8	635.6	635.4	635.1	634.9	634.6	634
50	642.7	642.5	642.3	642.1	641.9	641.6	641.4	641.2	641.0	640.8	640.5	640.3	640.1	639
60	647.9	647.7	647.5	647.3	647.1	646.9	646.7	646.5	646.3	646.1	645.9	645.7	645.5	645
70	653.1	653.0	652.8	652.6	652.4	652.2	652.0	651.8	651.6	651.5	651.2	651.1	650.9	650
80	658.4	658.2	658.0	657.8	657.7	657.5	657.3	657.1	656.9	656.8	656.6	656.4	656.2	656
90	663.6	663.4	663.2	663.1	662.9	662.7	662.6	662.4	662.2	662.1	661.9	661.7	661.6	661
100	668.8	668.6	668.5	668.3	668.1	668.0	667.8	667.7	667.5	667.4	667.2	667.1	666.9	666
110	674.0	673.8	673.7	673.5	673.4	673.2	673.1	673.0	672.8	672.7	672.5	672.4	672.2	672
120	679.2	679.1	678.9	678.8	678.6	678.5	678.4	678.2	678.1	678.0	677.8	677.7	677.5	677
130	684.4	684.3	684.9	684.0	683.9	683.8	683.6	683.5	683.4	683.3	683.1	683.0	682.9	682
140	689.7	689.5	689.4	689.3	689.2	689.0	688.9	688.8	688.7	688.6	688.4	688.3	688.2	688
150	694.9	694.8	694.7	694.6	694.4	694.3	694.2	694.1	694.0	693.9	693.7	693.6	693.5	693
160	700.2	700.1	700.0	699.8	699.7	699.6	699.5	699.4	699.3	699.2	699.1	699.0	698.8	698
170	705.5	705.4	705.3	705.1	705.0	704.9	704.8	704.7	704.6	704.5	704.4	704.3	704.2	704
180	710.8	710.7	710.6	710.5	710.4	710.3	710.2	710.1	710.0	709.9	709.8	709.7	709.6	709
190	716.1	716.0	715.9	715.8	715.7	715.6	715.5	715.4	715.3	715.2	715.1	715.0	714.9	714
200	721.4	721.3	721.2	721.1	721.1	721.0	720.9	720.8	720.7	720.6	720.5	720.4	720.3	720
220	732.2	732.1	732.0	731.9	731.8	731.7	731.7	731.6	731.5	731.4	731.3	731.2	731.1	731
240	—	—	742.8	742.7	742.7	742.6	742.6	742.5	742.4	742.3	742.2	742.2	742.0	742
260	—	—	—	—	—	—	—	753.4	753.3	753.2	753.2	753.1	753.0	752
280	—	—	—	—	—	—	—	—	—	—	—	—	764.1	764

Table 3. Ammonia—Enthalpy of Superheated Vapor (Btu per lb) (Continued)

mp, F	Pressure in psia							Saturation temperature in italics (deg F)						
	31	32	33	34	35	36	37	38	39	40	42	44	46	48
	0.79	*2.11*	*3.40*	*4.66*	*5.89*	*7.09*	*8.27*	*9.42*	*10.55*	*11.66*	*13.81*	*15.88*	*17.87*	*19.80*
sat)	(612.0)	(612.4)	(612.8)	(613.2)	(613.6)	(614.0)	(614.3)	(614.7)	(615.0)	(615.4)	(616.0)	(616.6)	(617.2)	(617.7)
0	617.4	617.1	616.8	616.4	616.1	615.7	615.4	615.0	—	—	—	—	—	—
0	623.2	622.9	622.6	622.3	622.0	621.7	621.4	621.0	620.7	620.4	619.8	619.1	618.5	617.8
0	628.8	628.5	628.3	628.0	627.7	627.4	627.2	626.9	626.6	626.3	625.8	625.2	624.6	624.0
0	634.4	634.1	633.9	633.6	633.4	633.1	632.9	632.6	632.4	632.1	631.6	631.1	630.5	630.0
0	639.9	639.6	639.4	639.2	638.9	638.7	638.5	638.3	638.0	637.8	637.3	636.8	636.4	635.9
0	645.3	645.1	644.9	644.7	644.4	644.2	644.0	643.8	643.6	643.4	643.0	642.5	642.1	741.6
0	650.7	650.5	650.3	650.1	649.9	649.7	649.5	649.3	649.1	648.9	648.5	648.1	647.7	647.3
0	565.1	655.9	655.7	655.5	655.3	655.2	655.0	654.8	654.6	654.4	654.1	653.7	653.3	652.9
0	661.4	661.2	661.1	660.9	660.7	660.6	660.4	660.2	660.1	659.9	659.5	659.2	658.9	658.5
0	666.7	666.6	666.4	666.3	666.1	666.0	665.8	665.6	665.5	665.3	665.0	664.7	664.4	664.0
0	672.1	671.9	671.8	671.6	671.5	671.3	671.2	671.0	670.9	670.7	670.4	670.1	669.8	669.5
0	677.4	677.3	667.1	677.0	676.8	676.7	676.6	676.4	676.3	676.1	675.9	675.6	675.3	675.0
0	682.7	682.6	682.5	682.3	682.2	682.1	681.9	681.8	681.7	681.5	681.3	681.0	680.7	680.5
0	688.1	687.9	687.8	687.7	687.6	687.4	687.3	687.2	687.1	686.9	686.7	686.4	686.2	685.9
0	693.4	693.3	693.2	693.0	692.9	692.8	692.7	692.6	692.5	692.3	692.1	691.9	691.6	691.4
0	698.7	698.6	698.5	698.4	698.3	698.2	698.1	698.0	697.8	697.7	697.5	697.3	697.1	696.8
0	704.1	704.0	703.9	703.8	703.7	703.6	703.5	703.3	703.2	703.1	702.9	702.7	702.5	702.3
0	709.5	709.4	709.3	709.2	709.1	709.0	708.9	708.7	708.6	708.5	708.3	708.1	707.9	707.7
0	714.8	714.7	714.6	714.5	714.5	714.4	714.3	714.2	714.1	714.0	713.8	713.6	713.4	713.2
0	720.2	720.1	720.0	720.0	719.9	719.8	719.7	719.6	719.5	719.4	719.2	719.0	718.8	718.7
0	731.1	731.0	730.9	730.8	730.7	730.6	730.6	730.5	730.4	730.3	730.1	730.0	729.8	729.6
0	742.0	741.9	741.8	741.7	741.7	741.6	741.5	741.4	741.3	741.3	741.1	741.0	740.8	740.6
0	752.9	752.9	752.8	752.7	752.7	752.6	752.5	752.4	752.4	752.3	752.2	752.0	751.9	751.7
0	764.0	763.9	763.9	763.8	763.7	763.7	763.6	763.5	763.5	763.4	763.3	763.1	763.0	762.9
0	—	—	—	—	—	—	—	—	—	774.6	774.5	774.3	774.2	774.1

mp, F	Pressure in psia							Saturation temperature in italics (deg F)						
	48	50	52	54	56	58	60	62	64	66	68	70	75	80
	19.80	*21.67*	*23.48*	*25.23*	*26.94*	*28.59*	*30.21*	*31.78*	*33.31*	*34.81*	*36.27*	*37.70*	*41.13*	*44.40*
sat)	(617.7)	(618.2)	(618.7)	(619.2)	(619.7)	(620.1)	(620.5)	(620.9)	(621.3)	(621.7)	(622.0)	(622.4)	(623.2)	(624.0)
0	617.8	—	—	—	—	—	—	—	—	—	—	—	—	—
0	624.0	623.4	622.8	622.2	621.6	621.0	—	—	—	—	—	—	—	—
0	630.0	629.5	629.0	628.4	627.9	627.3	626.8	626.2	625.6	625.1	624.5	623.9	—	—
0	635.9	635.4	634.9	634.4	633.9	633.4	632.9	632.4	631.9	631.4	630.9	630.4	629.1	627.7
0	641.6	641.2	640.8	640.3	639.9	639.4	639.0	638.5	638.0	637.6	637.1	636.6	635.5	634.3
0	647.3	646.9	646.5	646.1	645.7	645.3	644.9	644.4	644.0	643.6	643.2	642.7	641.7	640.6
0	652.9	652.6	652.2	651.8	651.4	651.1	650.7	650.3	649.9	649.5	649.1	648.7	647.7	646.7
0	658.5	658.2	657.8	657.5	657.1	656.8	656.4	656.0	655.7	655.3	655.0	645.6	653.7	652.8
0	664.0	663.7	663.4	663.1	662.7	662.4	662.1	661.7	661.4	661.1	660.7	660.6	659.6	658.7
0	669.5	669.2	668.9	668.6	668.3	668.0	667.7	667.4	667.1	666.8	666.5	666.1	665.4	664.6
0	675.0	674.7	674.4	674.2	673.9	673.6	673.3	673.0	672.7	672.4	672.1	671.8	671.1	670.4
0	680.5	680.2	679.9	679.7	679.4	679.1	678.9	678.6	678.3	678.0	677.8	677.5	676.8	676.1
0	685.9	685.7	685.4	685.2	684.9	684.7	684.4	684.2	683.9	683.6	683.4	683.1	682.5	681.8
0	691.4	691.1	690.9	690.7	690.4	690.2	689.9	689.7	689.5	689.2	689.0	688.7	688.1	687.5
0	696.8	696.6	696.4	696.1	695.9	695.7	695.5	695.2	695.0	694.8	694.5	694.3	693.7	693.2
0	702.3	702.1	701.8	701.6	701.4	701.2	701.0	700.8	700.5	700.3	700.1	699.9	699.3	698.8
0	707.3	707.5	707.3	707.1	706.9	706.7	706.5	706.3	706.1	705.9	705.7	705.5	704.9	704.4
0	713.2	713.0	712.8	712.6	712.4	712.2	712.0	711.8	711.6	711.4	711.2	711.0	710.5	710.0
0	718.7	718.5	718.3	718.1	717.9	717.7	717.5	717.3	717.2	717.0	716.8	716.6	716.1	715.6
0	724.2	724.0	723.8	723.6	723.4	723.2	723.1	722.9	722.7	722.5	722.3	722.2	721.7	721.3
0	729.6	729.4	729.3	729.1	728.9	728.8	728.6	728.4	728.3	728.1	727.9	727.7	727.3	726.9
0	735.1	735.0	734.8	734.7	734.5	734.4	734.1	734.0	733.8	733.7	733.5	733.3	732.9	732.5
0	740.6	740.5	740.3	740.2	740.0	739.9	739.7	739.5	739.4	739.2	739.1	738.9	738.5	738.1
0	746.2	745.6	745.9	745.8	745.6	745.5	745.3	745.1	745.0	744.8	744.7	744.5	744.1	743.8
0	751.7	751.6	751.4	751.3	751.1	751.0	750.9	750.7	750.6	750.4	750.3	750.1	749.8	749.4
0	757.3	757.2	756.0	756.9	756.7	756.6	756.5	756.3	756.2	756.1	756.8	757.8	755.5	755.1
0	762.9	762.7	762.6	762.5	762.3	762.2	762.1	761.9	761.8	761.7	761.5	761.4	761.1	760.7
0	768.5	768.4	768.2	768.1	768.0	767.8	767.7	767.6	767.5	767.4	767.2	767.1	766.8	766.4
0	774.1	774.0	773.8	773.7	773.6	773.5	773.3	773.2	773.1	773.0	772.8	772.7	772.4	772.1

Table 3. Ammonia—Enthalpy of Superheated Vapor (Btu per lb) (Continued)

Temp, deg F	Pressure in psia						Saturation temperature in italics (deg F)							
	80	85	90	95	100	105	110	115	120	125	130	135	140	145
	44.40	*47.50*	*50.47*	*53.32*	*56.05*	*58.67*	*61.21*	*63.65*	*66.02*	*68.31*	*70.53*	*72.69*	*74.79*	*76.83*
(*at sat*)	(624.0)	(624.7)	(625.3)	(625.9)	(626.5)	(627.0)	(627.5)	(628.0)	(628.4)	(628.8)	(629.2)	(629.6)	(629.9)	(630.3
50	627.7	626.4	—	—	—	—	—	—	—	—	—	—	—	—
60	634.3	633.0	631.8	630.5	629.3	627.9	—	—	—	—	—	—	—	—
70	640.6	639.5	638.3	637.2	636.0	634.9	633.7	632.5	631.3	630.0	—	—	—	—
80	646.7	645.7	644.7	643.6	642.6	641.5	640.5	639.4	638.3	637.2	636.0	634.9	633.8	632.6
90	652.8	651.8	650.9	649.9	649.0	648.0	647.0	646.0	645.0	644.0	643.0	642.0	640.9	639.9
100	658.7	657.8	657.0	656.1	655.2	654.3	653.4	652.5	651.6	650.7	649.7	648.8	647.8	646.9
110	664.6	663.8	663.0	662.1	661.3	660.5	659.7	658.8	658.0	657.1	656.3	654.4	654.5	653.6
120	670.4	669.6	668.9	668.1	667.3	666.6	665.8	665.0	664.2	663.5	662.7	661.9	661.1	660.2
130	676.1	675.4	674.7	674.0	673.3	672.6	671.9	671.1	670.4	669.7	668.9	668.2	667.4	666.7
140	681.8	681.2	680.5	679.8	679.2	678.5	677.8	677.2	676.5	675.8	675.1	674.4	673.7	673.0
150	687.5	686.9	686.3	685.6	685.0	684.4	683.7	683.1	682.5	681.8	681.2	680.5	679.9	679.2
160	693.2	692.6	692.0	691.4	690.8	690.2	689.6	689.0	688.4	687.8	687.2	686.6	686.0	685.4
170	698.8	698.2	697.7	697.1	696.6	696.0	695.4	694.9	694.3	693.7	693.2	692.6	692.0	691.4
180	704.4	703.9	703.4	702.8	702.3	701.8	701.2	700.7	700.2	699.6	699.1	698.6	698.0	697.5
190	710.0	709.5	709.0	708.5	708.0	707.5	707.0	706.5	706.0	705.5	705.0	704.5	704.0	703.4
200	715.6	715.2	714.7	714.2	713.7	713.3	712.8	712.3	711.8	711.3	710.9	710.4	709.9	709.4
210	721.3	720.8	720.4	719.9	719.4	719.0	718.5	718.1	717.6	717.2	716.7	716.2	715.8	715.3
220	726.9	726.4	726.0	725.6	725.1	724.7	724.3	723.8	723.4	723.0	722.5	722.1	721.6	721.2
230	732.5	732.1	731.7	731.3	730.8	730.4	730.0	729.6	729.2	728.8	728.3	727.9	727.5	727.1
240	738.1	737.7	737.3	736.9	736.5	736.1	735.7	735.3	734.9	734.5	734.1	733.7	733.3	732.9
250	743.8	743.4	743.0	742.6	742.2	741.9	741.5	741.1	740.7	740.3	739.9	739.6	739.2	738.8
260	749.4	749.0	748.7	748.3	747.9	747.6	747.2	746.8	746.5	746.1	745.7	745.4	745.0	744.6
270	755.1	754.7	754.4	754.0	753.6	753.3	752.9	752.6	752.2	751.9	751.5	751.2	750.8	750.5
280	760.7	760.4	760.0	759.7	759.4	759.0	758.7	758.4	758.0	757.7	757.3	757.0	756.7	756.3
290	766.4	766.1	765.8	765.8	765.1	764.8	764.5	764.1	763.8	763.5	763.1	762.8	762.5	762.2
300	772.1	771.8	771.5	771.2	770.8	770.5	770.2	769.9	769.6	769.3	769.0	768.6	768.3	768.0
320	—	—	—	—	—	—	—	—	—	780.9	780.6	780.3	780.0	779.7

Table 3. Ammonia—Enthalpy of Superheated Vapor (Btu per lb) (Concluded)

Temp, deg F	Pressure in psia								Saturation temperature in italics (deg F)									
	145	150	160	170	180	190	200	210	220	230	240	250	260	270	280	290	300
	76.83	*78.81*	*82.64*	*86.29*	*89.78*	*93.13*	*96.34*	*99.43*	*102.42*	*105.30*	*108.09*	*110.80*	*113.42*	*115.97*	*118.45*	*120.86*	*123.21*
(*at sat*)	(*630.2*)	(*630.5*)	(*631.1*)	(*631.6*)	(*632.0*)	(*632.4*)	(*632.7*)	(*633.0*)	(*633.2*)	(*633.4*)	(*633.6*)	(*633.8*)	(*633.9*)	(*633.9*)	(*634.0*)	(*634.0*)	(*634.0*)
80	632.6	631.4	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
90	639.9	638.8	636.6	634.4	632.2	—	—	—	—	—	—	—	—	—	—	—	—
100	646.9	645.9	643.9	641.9	639.9	637.8	635.6	633.5	—	—	—	—	—	—	—	—	—
110	653.6	652.8	651.0	649.1	647.3	645.4	643.4	641.5	639.4	637.4	635.3	—	—	—	—	—	—
120	660.2	659.4	657.8	656.1	654.4	652.6	650.9	649.1	647.3	645.4	643.5	641.5	639.5	637.5	635.4	—	—
130	666.7	665.9	664.4	662.8	661.3	659.7	658.1	656.4	654.8	653.1	651.3	649.6	647.8	645.9	644.0	642.1	640.1
140	673.0	672.3	670.9	669.4	668.0	666.5	665.0	663.5	662.0	660.4	658.8	657.2	655.6	653.9	652.2	650.5	648.7
150	679.2	678.6	677.2	675.9	674.6	673.2	671.8	670.4	669.0	667.6	666.1	664.6	663.1	661.6	660.1	658.5	656.9
160	685.4	684.8	683.5	682.3	681.0	679.7	678.4	677.1	675.8	674.5	673.1	671.8	670.4	669.0	667.6	666.1	664.7
170	691.4	690.9	689.7	688.5	687.3	686.1	684.9	683.7	682.5	681.3	680.0	678.7	677.5	676.2	674.9	673.5	672.2
180	697.5	696.9	695.8	694.7	693.6	692.5	691.3	690.2	689.1	687.9	686.7	685.5	684.4	683.2	681.9	680.7	679.5
190	703.4	702.9	701.9	700.8	699.8	698.7	697.7	696.6	695.5	694.4	693.3	692.2	691.1	690.0	688.9	687.7	686.5
200	709.4	708.9	707.9	706.9	705.9	704.9	703.9	702.9	701.9	700.9	699.8	698.8	697.7	696.7	695.6	694.6	693.5
210	715.3	714.8	713.9	713.0	712.0	711.1	710.1	709.2	708.2	707.2	706.2	705.3	704.3	703.3	702.3	701.3	700.3
220	721.2	720.7	719.9	719.0	718.1	717.2	716.3	715.3	714.4	713.5	712.6	711.7	710.7	709.8	708.8	707.9	706.9
230	727.1	726.6	725.8	724.9	724.1	723.2	722.4	721.5	720.6	719.8	718.9	718.0	717.1	716.2	715.3	714.4	713.5
240	732.9	732.5	731.7	730.9	730.1	729.3	728.4	727.6	726.8	726.0	725.1	724.3	723.4	722.6	721.8	720.9	720.0
250	738.8	738.4	737.6	736.8	736.1	735.3	734.5	733.7	732.9	732.1	731.3	730.5	729.7	728.9	728.1	727.3	726.5
260	744.6	744.3	743.5	742.8	742.0	741.3	740.5	739.8	739.0	738.3	737.5	736.7	736.0	735.2	734.4	733.7	732.9
270	750.5	750.1	749.4	748.7	748.0	747.3	746.5	745.8	745.1	744.4	743.6	742.9	742.2	741.4	740.7	740.0	739.2
280	756.3	756.0	755.3	754.6	753.9	753.2	752.5	751.8	751.1	750.5	749.8	749.1	748.4	747.7	747.0	746.3	745.5
290	762.2	761.8	761.2	760.5	759.9	759.2	758.5	757.9	757.2	756.5	755.9	755.2	754.5	753.9	753.2	752.5	751.8
300	768.0	767.7	767.1	766.4	765.8	765.2	764.5	763.9	763.2	762.6	762.0	761.3	760.7	760.0	759.4	758.7	758.1
320	779.7	779.4	778.9	778.3	777.7	777.1	776.5	775.9	775.3	774.7	774.1	773.5	772.9	772.3	771.7	771.1	770.5
340	—	791.2	790.7	790.1	789.6	789.0	788.5	787.9	787.4	786.8	786.3	785.7	785.2	784.6	784.0	783.5	782.9
360	—	—	—	—	—	—	800.5	800.0	799.5	798.9	798.4	797.9	797.4	796.9	796.3	795.8	795.3
380	—	—	—	—	—	—	812.5	812.0	811.6	811.1	810.6	810.1	809.6	809.1	808.7	808.2	807.7
400	—	—	—	—	—	—	—	—	—	—	—	822.3	821.9	821.4	821.0	820.5	820.1

Table 4. Ammonia—Entropy of Superheated Vapor (Btu/lb F from −40 F)

Temp, deg F	Pressure in psia							Saturation temperature in italics (deg F)						
	5	6	7	8	9	10	11	12	13	14	15	16	17	18
	−63.11	*−57.64*	*−52.88*	*−48.64*	*−44.88*	*−41.34*	*−38.14*	*−35.16*	*−32.37*	*−29.76*	*−27.29*	*−24.96*	*−22.73*	*−20*
(*at sat*)	(*1.4857*)	(*1.4703*)	(*1.4574*)	(*1.4462*)	(*1.4363*)	(*1.4276*)	(*1.4196*)	(*1.4124*)	(*1.4057*)	(*1.3996*)	(*1.3938*)	(*1.3885*)	(*1.3835*)	(*1.37*
−50	1.5025	1.4803	1.4611	—	—	—	—	—	—	—	—	—	—	—
−40	1.5149	1.4928	1.4739	1.4573	1.4426	1.4293	—	—	—	—	—	—	—	—
−30	1.5269	1.5049	1.4861	1.4697	1.4551	1.4420	1.4300	1.4190	1.4088	—	—	—	—	—
−20	1.5385	1.5166	1.4979	1.4816	1.4672	1.4542	1.4423	1.4314	1.4213	1.4119	1.4031	1.3948	1.3870	1.3
−10	1.5498	1.5280	1.5094	1.4932	1.4788	1.4659	1.4542	1.4434	1.4334	1.4241	1.4154	1.4072	1.3994	1.3
0	1.5608	1.5391	1.5206	1.5044	1.4902	1.4773	1.4656	1.4549	1.4450	1.4358	1.4272	1.4191	1.4114	1.4
10	1.5716	1.5499	1.5314	1.5154	1.5012	1.4884	1.4768	1.4661	1.4563	1.4472	1.4386	1.4306	1.4230	1.4
20	1.5821	1.5605	1.5421	1.5261	1.5119	1.4992	1.4876	1.4770	1.4672	1.4582	1.4497	1.4417	1.4342	1.4
30	1.5925	1.5708	1.5525	1.5365	1.5224	1.5097	1.4982	1.4877	1.4779	1.4688	1.4604	1.4525	1.4450	1.4
40	1.6026	1.5810	1.5627	1.5467	1.5327	1.5200	1.5085	1.4980	1.4883	1.4793	1.4709	1.4630	1.4556	1.4
50	1.6125	1.5910	1.5727	1.5568	1.5427	1.5301	1.5187	1.5082	1.4985	1.4896	1.4812	1.4733	1.4659	1.4
60	1.6223	1.6008	1.5825	1.5666	1.5526	1.5400	1.5286	1.5182	1.5085	1.4996	1.4912	1.4834	1.4761	1.4
70	1.6319	1.6104	1.5921	1.5763	1.5623	1.5497	1.5383	1.5279	1.5183	1.5094	1.5011	1.4933	1.4860	1.4
80	1.6413	1.6199	1.6016	1.5858	1.5718	1.5593	1.5479	1.5375	1.5279	1.5191	1.5108	1.5030	1.4957	1.4
90	1.6506	1.6292	1.6110	1.5952	1.5812	1.5687	1.5573	1.5470	1.5374	1.5285	1.5203	1.5125	1.5052	1.4
100	1.6598	1.6384	1.6202	1.6044	1.5904	1.5779	1.5666	1.5562	1.5467	1.5378	1.5296	1.5218	1.5146	1.5
110	1.6689	1.6474	1.6292	1.6135	1.5995	1.5870	1.5757	1.5654	1.5558	1.5470	1.5388	1.5310	1.5238	1.5
120	1.6778	1.6563	1.6382	1.6224	1.6085	1.5960	1.5847	1.5744	1.5649	1.5560	1.5478	1.5401	1.5328	1.5
130	1.6865	1.6651	1.6470	1.6312	1.6173	1.6049	1.5936	1.5833	1.5737	1.5649	1.5567	1.5490	1.5418	1.5
140	1.6952	1.6738	1.6557	1.6399	1.6260	1.6136	1.6023	1.5920	1.5825	1.5737	1.5655	1.5578	1.5506	1.5
150	1.7038	1.6824	1.6643	1.6485	1.6346	1.6222	1.6109	1.6006	1.5911	1.5824	1.5742	1.5665	1.5593	1.5
160	1.7122	1.6909	1.6727	1.6570	1.6431	1.6307	1.6194	1.6092	1.5997	1.5909	1.5827	1.5750	1.5678	1.5
170	1.7206	1.6992	1.6811	1.6654	1.6515	1.6391	1.6278	1.6176	1.6081	1.5993	1.5911	1.5835	1.5763	1.5
180	1.7289	1.7075	1.6894	1.6737	1.6598	1.6474	1.6362	1.6259	1.6164	1.6076	1.5995	1.5918	1.5846	1.5
190	—	—	—	—	—	1.6556	1.6444	1.6341	1.6246	1.6159	1.6077	1.6001	1.5929	1.5
200	—	—	—	—	—	1.6637	1.6525	1.6422	1.6328	1.6240	1.6158	1.6082	1.6010	1.5
210	—	—	—	—	—	—	—	—	—	—	1.6238	1.6162	1.6090	1.5
220	—	—	—	—	—	—	—	—	—	—	1.6318	1.6242	1.6170	1.6

Temp, deg F	Pressure in psia							Saturation temperature in italics (deg F)						
	18	19	20	21	22	23	24	25	26	27	28	29	30	31
	−20.61	*−18.58*	*−16.64*	*−14.78*	*−12.98*	*−11.25*	*−9.58*	*−7.96*	*−6.39*	*−4.87*	*−3.40*	*−1.97*	*−0.57*	*+0*
(*at sat*)	(*1.3787*)	(*1.3742*)	(*1.3700*)	(*1.3659*)	(*1.3621*)	(*1.3584*)	(*1.3549*)	(*1.3515*)	(*1.3482*)	(*1.3451*)	(*1.3421*)	(*1.3392*)	(*1.3364*)	(*1.33*
−20	1.3795	—	—	—	—	—	—	—	—	—	—	—	—	—
−10	1.3921	1.3851	1.3784	1.3720	1.3659	1.3600	—	—	—	—	—	—	—	—
0	1.4042	1.3973	1.3907	1.3844	1.3784	1.3726	1.3670	1.3616	1.3564	1.3513	1.3465	1.3417	1.3371	
10	1.4158	1.4090	1.4025	1.3962	1.3903	1.3846	1.3791	1.3738	1.3686	1.3637	1.3589	1.3542	1.3497	1.34
20	1.4270	1.4203	1.4138	1.4077	1.4018	1.3961	1.3907	1.3855	1.3804	1.3755	1.3708	1.3662	1.3618	1.35
30	1.4380	1.4312	1.4248	1.4187	1.4129	1.4073	1.4019	1.3967	1.3917	1.3869	1.3822	1.3777	1.3733	1.36
40	1.4486	1.4419	1.4356	1.4295	1.4237	1.4181	1.4128	1.4077	1.4027	1.3979	1.3933	1.3888	1.3845	1.38
50	1.4590	1.4523	1.4460	1.4400	1.4342	1.4287	1.4234	1.4183	1.4134	1.4087	1.4041	1.3996	1.3953	1.39
60	1.4691	1.4625	1.4562	1.4502	1.4445	1.4390	1.4337	1.4287	1.4238	1.4191	1.4145	1.4101	1.4059	1.40
70	1.4790	1.4724	1.4662	1.4602	1.4545	1.4491	1.4438	1.4388	1.4339	1.4292	1.4247	1.4204	1.4161	1.41
80	1.4887	1.4822	1.4760	1.4700	1.4643	1.4589	1.4537	1.4487	1.4439	1.4392	1.4347	1.4304	1.4261	1.42
90	1.4983	1.4918	1.4856	1.4796	1.4740	1.4686	1.4634	1.4584	1.4536	1.4489	1.4445	1.4401	1.4359	1.43
100	1.5077	1.5012	1.4950	1.4891	1.4834	1.4780	1.4729	1.4679	1.4631	1.4585	1.4540	1.4497	1.4456	1.44
110	1.5169	1.5104	1.5042	1.4983	1.4927	1.4873	1.4822	1.4772	1.4725	1.4679	1.4634	1.4591	1.4550	1.45
120	1.5260	1.5195	1.5133	1.5075	1.5019	1.4965	1.4914	1.4864	1.4817	1.4771	1.4726	1.4684	1.4642	1.46
130	1.5349	1.5285	1.5223	1.5165	1.5109	1.5055	1.5004	1.4954	1.4907	1.4861	1.4817	1.4775	1.4733	1.46
140	1.5438	1.5373	1.5312	1.5253	1.5197	1.5144	1.5093	1.5043	1.4996	1.4950	1.4906	1.4864	1.4823	1.47
150	1.5525	1.5460	1.5399	1.5340	1.5285	1.5231	1.5180	1.5131	1.5084	1.5038	1.4994	1.4952	1.4911	1.48
160	1.5610	1.5546	1.5485	1.5426	1.5371	1.5317	1.5266	1.5217	1.5170	1.5125	1.5081	1.5039	1.4998	1.49
170	1.5695	1.5631	1.5569	1.5510	1.5456	1.5402	1.5352	1.5303	1.5256	1.5210	1.5167	1.5124	1.5083	1.50
180	1.5778	1.5714	1.5653	1.5595	1.5539	1.5486	1.5436	1.5387	1.5340	1.5295	1.5251	1.5209	1.5168	1.51
190	1.5861	1.5797	1.5736	1.5678	1.5622	1.5569	1.5518	1.5470	1.5423	1.5378	1.5334	1.5292	1.5251	1.52
200	1.5943	1.5878	1.5817	1.5759	1.5704	1.5651	1.5600	1.5552	1.5505	1.5460	1.5416	1.5374	1.5334	1.52
210	1.6023	1.5959	1.5898	1.5839	1.5785	1.5732	1.5681	1.5633	1.5586	1.5541	1.5497	1.5455	1.5415	1.53
220	1.6103	1.6039	1.5978	1.5920	1.5865	1.5812	1.5761	1.5713	1.5666	1.5621	1.5578	1.5536	1.5495	1.54
230	—	—	1.6057	1.5999	1.5944	1.5891	1.5840	1.5792	1.5745	1.5680	1.5657	1.5615	1.5574	1.55
240	—	—	1.6135	1.6077	1.6022	1.5969	1.5919	1.5870	1.5824	1.5779	1.5736	1.5694	1.5653	1.56
250	—	—	—	—	—	—	—	1.5948	1.5901	1.5856	1.5813	1.5771	1.5731	1.56
260	—	—	—	—	—	—	—	1.6025	1.5978	1.5933	1.5890	1.5848	1.5808	1.57
270	—	—	—	—	—	—	—	—	—	—	—	—	1.5884	1.58
280	—	—	—	—	—	—	—	—	—	—	—	—	1.5960	1.59

Table 4. Ammonia—Entropy of Superheated Vapor (Btu/lb F from −40 F) (Continued)

np, F	Pressure in psia								Saturation temperature in italics (deg F)					
	31	32	33	34	35	36	37	38	39	40	42	44	46	48
	0.79	*2.11*	*3.40*	*4.66*	*5.89*	*7.09*	*8.27*	*9.42*	*10.55*	*11.66*	*13.81*	*15.88*	*17.87*	*19.80*
sat)	(1.3336)	(1.3310)	(1.3285)	(1.3260)	(1.3236)	(1.3213)	(1.3190)	(1.3168)	(1.3146)	(1.3125)	(1.3084)	(1.3046)	(1.3009)	(1.2973)
0	1.3453	1.3411	1.3369	1.3328	1.3289	1.3250	1.3212	1.3175	—	—	—	—	—	—
0	1.3574	1.3532	1.3492	1.3452	1.3413	1.3375	1.3338	1.3301	1.3266	1.3231	1.3164	1.3099	1.3036	1.2976
0	1.3691	1.3649	1.3609	1.3570	1.3532	1.3494	1.3458	1.3422	1.3387	1.3353	1.3287	1.3224	1.3162	1.3103
0	1.3803	1.3762	1.3722	1.3684	1.3646	1.3609	1.3573	1.3538	1.3504	1.3470	1.3405	1.3343	1.3283	1.3225
0	1.3912	1.3871	1.3832	1.3793	1.3756	1.3720	1.3684	1.3650	1.3616	1.3583	1.3519	1.3457	1.3398	1.3341
0	1.4017	1.3977	1.3938	1.3900	1.3863	1.3827	1.3792	1.3758	1.3724	1.3692	1.3628	1.3567	1.3509	1.3453
0	1.4120	1.4080	1.4042	1.4004	1.3967	1.3932	1.3897	1.3863	1.3830	1.3797	1.3734	1.3674	1.3617	1.3561
0	1.4221	1.4181	1.4143	1.4105	1.4069	1.4033	1.3999	1.3965	1.3932	1.3900	1.3838	1.3778	1.3721	1.3666
0	1.4319	1.4280	1.4241	1.4204	1.4168	1.4133	1.4098	1.4065	1.4032	1.4000	1.3939	1.3880	1.3823	1.3768
0	1.4415	1.4376	1.4338	1.4301	1.4265	1.4230	1.4196	1.4163	1.4130	1.4098	1.4037	1.3978	1.3922	1.3868
0	1.4510	1.4470	1.4433	1.4396	1.4360	1.4325	1.4291	1.4258	1.4226	1.4194	1.4133	1.4075	1.4019	1.3965
0	1.4602	1.4563	1.4526	1.4489	1.4453	1.4419	1.4385	1.4352	1.4320	1.4288	1.4228	1.4170	1.4114	1.4061
0	1.4693	1.4655	1.4617	1.4581	1.4545	1.4510	1.4477	1.4444	1.4412	1.4381	1.4320	1.4263	1.4207	1.4154
0	1.4783	1.4744	1.4707	1.4671	1.4635	1.4601	1.4567	1.4534	1.4503	1.4471	1.4411	1.4354	1.4299	1.4246
0	1.4871	1.4833	1.4795	1.4759	1.4724	1.4689	1.4656	1.4623	1.4592	1.4561	1.4501	1.4444	1.4389	1.4336
0	1.4958	1.4920	1.4883	1.4846	1.4811	1.4777	1.4744	1.4711	1.4679	1.4648	1.4589	1.4532	1.4477	1.4425
0	1.5044	1.5006	1.4968	1.4932	1.4897	1.4863	1.4830	1.4797	1.4766	1.4735	1.4676	1.4619	1.4564	1.4512
0	1.5129	1.5090	1.5053	1.5017	1.4982	1.4948	1.4915	1.4883	1.4851	1.4820	1.4761	1.4704	1.4650	1.4598
0	1.5212	1.5174	1.5137	1.5101	1.5066	1.5032	1.4999	1.4966	1.4935	1.4904	1.4845	1.4789	1.4735	1.4683
0	1.5294	1.5256	1.5219	1.5183	1.5148	1.5115	1.5082	1.5049	1.5018	1.4987	1.4928	1.4872	1.4818	1.4766
0	1.5375	1.5337	1.5300	1.5265	1.5230	1.5196	1.5163	1.5131	1.5100	1.5069	1.5010	1.4954	1.4900	1.4848
0	1.5456	1.5418	1.5381	1.5346	1.5311	1.5277	1.5244	1.5212	1.5181	1.5150	1.5091	1.5035	1.4981	1.4930
0	1.5535	1.5497	1.5461	1.5425	1.5390	1.5357	1.5324	1.5292	1.5261	1.5230	1.5171	1.5115	1.5061	1.5010
0	1.5614	1.5576	1.5540	1.5504	1.5469	1.5436	1.5403	1.5371	1.5340	1.5309	1.5251	1.5195	1.5141	1.5090
0	1.5692	1.5654	1.5619	1.5582	1.5547	1.5514	1.5481	1.5449	1.5418	1.5387	1.5329	1.5273	1.5219	1.5168
0	1.5769	1.5731	1.5695	1.5659	1.5624	1.5591	1.5558	1.5526	1.5495	1.5465	1.5406	1.5350	1.5297	1.5246
0	1.5845	1.5757	1.5721	1.5735	1.5700	1.5667	1.5634	1.5602	1.5571	1.5541	1.5483	1.5427	1.5374	1.5323
0	1.5921	1.5883	1.5846	1.5811	1.5776	1.5743	1.5710	1.5678	1.5647	1.5617	1.5559	1.5503	1.5450	1.5399
0	—	—	—	—	—	—	—	—	—	1.5692	1.5634	1.5578	1.5525	1.5474
0	—	—	—	—	—	—	—	—	—	1.5766	1.5708	1.5652	1.5599	1.5548

np, F	Pressure in psia								Saturation temperature in italics (deg F)					
	48	50	52	54	56	58	60	62	64	66	68	70	75	80
	19.80	*21.67*	*23.48*	*25.23*	*26.94*	*28.59*	*30.21*	*31.78*	*33.31*	*34.81*	*36.27*	*37.70*	*41.13*	*44.40*
sat)	(1.2973)	(1.2939)	(1.2906)	(1.2875)	(1.2844)	(1.2815)	(1.2787)	(1.2759)	(1.2733)	(1.2707)	(1.2682)	(1.2658)	(1.2599)	(1.2545)
0	1.2976	—	—	—	—	—	—	—	—	—	—	—	—	—
0	1.3103	1.3046	1.2991	1.2937	1.2884	1.2834	—	—	—	—	—	—	—	—
0	1.3225	1.3169	1.3114	1.3062	1.3011	1.2961	1.2913	1.2866	1.2820	1.2775	1.2731	1.2688	—	—
0	1.3341	1.3268	1.3233	1.3181	1.3131	1.3082	1.3035	1.2989	1.2944	1.2900	1.2858	1.2816	1.2715	1.2619
0	1.3453	1.3399	1.3346	1.3295	1.3246	1.3199	1.3152	1.3107	1.3063	1.3020	1.2978	1.2937	1.2839	1.2745
0	1.3561	1.3508	1.3456	1.3406	1.3357	1.3310	1.3265	1.3220	1.3177	1.3135	1.3094	1.3054	1.2957	1.2866
0	1.3666	1.3613	1.3562	1.3513	1.3465	1.3418	1.3373	1.3330	1.3287	1.3245	1.3205	1.3166	1.3071	1.2981
0	1.3768	1.3716	1.3665	1.3616	1.3569	1.3523	1.3479	1.3435	1.3393	1.3352	1.3312	1.3274	1.3180	1.3092
0	1.3868	1.3816	1.3766	1.3717	1.3670	1.3625	1.3581	1.3538	1.3496	1.3456	1.3417	1.3378	1.3286	1.3199
0	1.3965	1.3914	1.3864	1.3816	1.3769	1.3724	1.3681	1.3638	1.3597	1.3557	1.3518	1.3480	1.3389	1.3303
0	1.4061	1.4009	1.3960	1.3912	1.3866	1.3821	1.3778	1.3736	1.3695	1.3655	1.3617	1.3579	1.3489	1.3404
0	1.4154	1.4103	1.4054	1.4006	1.3961	1.3916	1.3873	1.3831	1.3791	1.3751	1.3713	1.3676	1.3586	1.3502
0	1.4246	1.4195	1.4146	1.4099	1.4053	1.4009	1.3966	1.3925	1.3885	1.3846	1.3807	1.3770	1.3682	1.3598
0	1.4336	1.4286	1.4237	1.4190	1.4144	1.4100	1.4058	1.4017	1.3977	1.3938	1.3900	1.3863	1.3775	1.3692
0	1.4425	1.4374	1.4326	1.4279	1.4234	1.4190	1.4148	1.4107	1.4067	1.4028	1.3991	1.3954	1.3866	1.3784
0	1.4512	1.4462	1.4413	1.4367	1.4322	1.4278	1.4236	1.4195	1.4156	1.4117	1.4080	1.4043	1.3956	1.3874
0	1.4598	1.4548	1.4500	1.4453	1.4408	1.4365	1.4323	1.4282	1.4243	1.4205	1.4167	1.4131	1.4044	1.3963
0	1.4683	1.4633	1.4585	1.4538	1.4494	1.4450	1.4409	1.4368	1.4329	1.4291	1.4254	1.4217	1.4131	1.4050
0	1.4766	1.4716	1.4668	1.4622	1.4578	1.4535	1.4493	1.4453	1.4413	1.4375	1.4338	1.4302	1.4217	1.4136
0	1.4848	1.4799	1.4751	1.4705	1.4661	1.4618	1.4576	1.4536	1.4497	1.4459	1.4422	1.4386	1.4301	1.4220
0	1.4930	1.4880	1.4833	1.4787	1.4743	1.4700	1.4658	1.4618	1.4579	1.4541	1.4505	1.4469	1.4384	1.4304
0	1.5010	1.4960	1.4913	1.4867	1.4823	1.4780	1.4739	1.4699	1.4660	1.4623	1.4586	1.4550	1.4466	1.4386
0	1.5090	1.5040	1.4993	1.4947	1.4903	1.4860	1.4819	1.4779	1.4741	1.4703	1.4666	1.4631	1.4546	1.4467
0	1.5168	1.5119	1.5071	1.5026	1.4982	1.4939	1.4898	1.4858	1.4820	1.4782	1.4745	1.4710	1.4625	1.4547
0	1.5246	1.5197	1.5149	1.5104	1.5060	1.5017	1.4976	1.4937	1.4898	1.4861	1.4824	1.4789	1.4705	1.4626
0	1.5323	1.5274	1.5226	1.5181	1.5137	1.5094	1.5053	1.5014	1.4975	1.4938	1.4902	1.4866	1.4783	1.4704
0	1.5399	1.5350	1.5303	1.5257	1.5213	1.5171	1.5130	1.5091	1.5052	1.5015	1.4979	1.4943	1.4860	1.4781
0	1.5474	1.5425	1.5378	1.5332	1.5289	1.5246	1.5206	1.5116	1.5128	1.5091	1.5055	1.5019	1.4936	1.4857
0	1.5548	1.5500	1.5453	1.5407	1.5364	1.5321	1.5281	1.5241	1.5203	1.5166	1.5130	1.5095	1.5011	1.4933

Table 4. Ammonia—Entropy of Superheated Vapor (Btu/lb F from −40 F) (Continued)

Temp, deg F	Pressure in psia							Saturation temperature in italics (deg F)						
	80	85	90	95	100	105	110	115	120	125	130	135	140	14
	44.40	*47.50*	*50.47*	*53.32*	*56.05*	*58.67*	*61.21*	*63.65*	*66.02*	*68.31*	*70.53*	*72.69*	*74.79*	*76.*
(*at sat*)	(*1.2545*)	(*1.2494*)	(*1.2445*)	(*1.2399*)	(*1.2356*)	(*1.2314*)	(*1.2275*)	*1.2237*)	(*1.2201*)	(*1.2166*)	(*1.2132*)	(*1.2100*)	(*1.2068*)	(*1.20*
50	1.2619	1.2527	—	—	—	—	—	—	—	—	—	—	—	—
60	1.2745	1.2656	1.2571	1.2489	1.2409	1.2332	—	—	—	—	—	—	—	—
70	1.2866	1.2779	1.2695	1.2616	1.2539	1.2464	1.2392	1.2323	1.2255	1.2189	—	—	—	—
80	1.2981	1.2896	1.2814	1.2736	1.2661	1.2589	1.2519	1.2451	1.2386	1.2322	1.2660	1.2199	1.2140	1.2
90	1.3092	1.3008	1.2928	1.2852	1.2778	1.2708	1.2640	1.2574	1.2510	1.2448	1.2388	1.2329	1.2272	1.2
100	1.3199	1.3117	1.3038	1.2963	1.2891	1.2822	1.2755	1.2690	1.2628	1.2568	1.2509	1.2452	1.2396	1.2
110	1.3303	1.3221	1.3144	1.3070	1.2999	1.2931	1.2866	1.2802	1.2741	1.2682	1.2625	1.2569	1.2515	1.2
120	1.3404	1.3323	1.3247	1.3174	1.3104	1.3037	1.2972	1.2910	1.2850	1.2792	1.2736	1.2681	1.2628	1.2
130	1.3502	1.3422	1.3347	1.3275	1.3206	1.3139	1.3076	1.3015	1.2956	1.2899	1.2843	1.2790	1.2738	1.2
140	1.3598	1.3519	1.3444	1.3373	1.3305	1.3239	1.3176	1.3116	1.3058	1.3002	1.2947	1.2894	1.2843	1.2
150	1.3692	1.3614	1.3539	1.3469	1.3401	1.3336	1.3274	1.3215	1.3157	1.3102	1.3048	1.2996	1.2945	1.2
160	1.3784	1.3706	1.3633	1.3562	1.3495	1.3431	1.3370	1.3311	1.3254	1.3199	1.3146	1.3094	1.3045	1.2
170	1.3874	1.3797	1.3724	1.3654	1.3588	1.3521	1.3463	1.3405	1.3348	1.3294	1.3241	1.3191	1.3141	1.3
180	1.3963	1.3886	1.3813	1.3744	1.3678	1.3615	1.3555	1.3497	1.3441	1.3387	1.3335	1.3284	1.3236	1.3
190	1.4050	1.3974	1.3901	1.3833	1.3767	1.3704	1.3644	1.3587	1.3531	1.3478	1.3426	1.3376	1.3328	1.3
220	1.4136	1.4060	1.3988	1.3919	1.3854	1.3792	1.3732	1.3675	1.3620	1.3567	1.3516	1.3466	1.3418	1.3
210	1.4220	1.4145	1.4073	1.4005	1.3940	1.3878	1.3819	1.3762	1.3707	1.3654	1.3604	1.3554	1.3507	1.3
220	1.4304	1.4228	1.4157	1.4089	1.4024	1.3963	1.3904	1.3847	1.3793	1.3740	1.3690	1.3641	1.3594	1.3
230	1.4386	1.4311	1.4239	1.4172	1.4108	1.4046	1.3988	1.3931	1.3877	1.3825	1.3775	1.3726	1.3679	1.3
240	1.4467	1.4392	1.4321	1.4254	1.4190	1.4129	1.4070	1.4014	1.3960	1.3908	1.3858	1.3810	1.3763	1.3
250	1.4547	1.4472	1.4401	1.4334	1.4271	1.4210	1.4151	1.4096	1.4042	1.3990	1.3941	1.3893	1.3846	1.3
260	1.4626	1.4551	1.4481	1.4414	1.4350	1.4290	1.4232	1.4176	1.4123	1.4071	1.4022	1.3974	1.3928	1.3
270	1.4704	1.4629	1.4559	1.4492	1.4429	1.4369	1.4311	1.4256	1.4202	1.4151	1.4102	1.4054	1.4008	1.3
280	1.4781	1.4707	1.4637	1.4570	1.4507	1.4447	1.4389	1.4334	1.4281	1.4230	1.4181	1.4133	1.4088	1.4
290	1.4857	1.4783	1.4713	1.4647	1.4584	1.4524	1.4466	1.4411	1.4359	1.4308	1.4259	1.4212	1.4166	1.4
300	1.4933	1.4859	1.4789	1.4723	1.4660	1.4600	1.4543	1.4488	1.4435	1.4385	1.4336	1.4289	1.4243	1.4
320	—	—	—	—	—	—	—	—	—	1.4536	1.4487	1.4441	1.4395	1.4

Table 4. Ammonia—Entropy of Superheated Vapor (Btu/lb F from −40 F) (Concluded)

Temp, deg F	Pressure in psia						Saturation temperature in italics (deg F)										
	145	150	160	170	180	190	200	210	220	230	240	250	260	270	280	290	300
	76.83	*78.81*	*82.64*	*86.29*	*89.78*	*93.13*	*96.34*	*99.43*	*102.42*	*105.30*	*108.09*	*110.80*	*113.42*	*115.97*	*118.45*	*120.86*	*123.21*
(*at sat*)	(*1.2038*)	(*1.2009*)	(*1.1952*)	(*1.1900*)	(*1.1850*)	(*1.1802*)	(*1.1756*)	(*1.1713*)	(*1.1671*)	(*1.1631*)	(*1.1592*)	(*1.1555*)	(*1.1518*)	(*1.1483*)	(*1.1449*)	(*1.1415*)	(*1.1383*)
80	1.2082	1.2025	—	—	—	—	—	—	—	—	—	—	—	—	—	—	—
90	1.2216	1.2161	1.2055	1.1952	1.1853	—	—	—	—	—	—	—	—	—	—	—	—
100	1.2342	1.2289	1.2186	1.2087	1.1992	1.1899	1.1809	1.1721	—	—	—	—	—	—	—	—	—
110	1.2462	1.2410	1.2311	1.2215	1.2123	1.2034	1.1947	1.1863	1.1781	1.1700	1.1621	—	—	—	—	—	—
120	1.2577	1.2526	1.2429	1.2336	1.2247	1.2160	1.2077	1.1996	1.1917	1.1840	1.1764	1.1690	1.1617	1.1544	1.1473	—	—
130	1.2687	1.2638	1.2542	1.2452	1.2364	1.2281	1.2200	1.2121	1.2045	1.1971	1.1898	1.1827	1.1757	1.1689	1.1621	1.1554	1.1487
140	1.2793	1.2745	1.2652	1.2563	1.2477	1.2396	1.2317	1.2240	1.2167	1.2095	1.2025	1.1956	1.1889	1.1823	1.1759	1.1695	1.1632
150	1.2896	1.2849	1.2757	1.2669	1.2586	1.2506	1.2429	1.2354	1.2281	1.2213	1.2145	1.2078	1.2014	1.1950	1.1888	1.1827	1.1767
160	1.2996	1.2949	1.2859	1.2773	1.2691	1.2612	1.2537	1.2464	1.2394	1.2325	1.2259	1.2195	1.2132	1.2071	1.2011	1.1952	1.1894
170	1.3093	1.3047	1.2958	1.2873	1.2792	1.2715	1.2641	1.2569	1.2501	1.2434	1.2369	1.2306	1.2245	1.2185	1.2127	1.2070	1.2014
180	1.3188	1.3142	1.3054	1.2971	1.2891	1.2815	1.2742	1.2672	1.2604	1.2538	1.2475	1.2414	1.2354	1.2296	1.2239	1.2183	1.2129
190	1.3281	1.3236	1.3148	1.3066	1.2987	1.2912	1.2840	1.2771	1.2704	1.2640	1.2577	1.2517	1.2458	1.2401	1.2346	1.2292	1.2239
200	1.3372	1.3327	1.3240	1.3159	1.3081	1.3007	1.2935	1.2867	1.2801	1.2738	1.2677	1.2617	1.2560	1.2504	1.2449	1.2396	1.2344
210	1.3461	1.3416	1.3331	1.3249	1.3172	1.3099	1.3029	1.2961	1.2896	1.2834	1.2773	1.2715	1.2658	1.2603	1.2550	1.2497	1.2447
220	1.3548	1.3504	1.3419	1.3338	1.3262	1.3189	1.3120	1.3053	1.2989	1.2927	1.2867	1.2810	1.2754	1.2700	1.2647	1.2596	1.2546
230	1.3634	1.3590	1.3506	1.3426	1.3350	1.3278	1.3209	1.3143	1.3079	1.3018	1.2959	1.2902	1.2847	1.2794	1.2742	1.2691	1.2642
240	1.3718	1.3675	1.3591	1.3512	1.3436	1.3365	1.3296	1.3231	1.3168	1.3107	1.3049	1.2993	1.2938	1.2885	1.2834	1.2784	1.2736
250	1.3801	1.3758	1.3675	1.3596	1.3521	1.3450	1.3382	1.3317	1.3255	1.3195	1.3137	1.3081	1.3027	1.2975	1.2924	1.2875	1.2827
260	1.3883	1.3840	1.3757	1.3679	1.3605	1.3534	1.3467	1.3402	1.3340	1.3281	1.3224	1.3168	1.3115	1.3063	1.3013	1.2964	1.2917
270	1.3964	1.3921	1.3838	1.3761	1.3687	1.3617	1.3550	1.3486	1.3424	1.3365	1.3308	1.3253	1.3200	1.3149	1.3099	1.3051	1.3004
280	1.4043	1.4001	1.3919	1.3841	1.3786	1.3698	1.3631	1.3568	1.3507	1.3448	1.3392	1.3337	1.3285	1.3234	1.3184	1.3137	1.3090
290	1.4122	1.4079	1.3998	1.3921	1.3847	1.3778	1.3712	1.3649	1.3588	1.3530	1.3474	1.3420	1.3367	1.3317	1.3268	1.3221	1.3175
300	1.4199	1.4157	1.4076	1.3999	1.3926	1.3857	1.3791	1.3728	1.3668	1.3610	1.3554	1.3501	1.3449	1.3399	1.3350	1.3303	1.3257
320	1.4352	1.4310	1.4229	1.4153	1.4081	1.4012	1.3947	1.3884	1.3825	1.3767	1.3712	1.3659	1.3608	1.3559	1.3511	1.3464	1.3419
340	—	1.4459	1.4379	1.4303	1.4231	1.4163	1.4099	1.4037	1.3978	1.3921	1.3866	1.3814	1.3763	1.3714	1.3667	1.3621	1.3576
360	—	—	—	—	—	—	1.4247	1.4186	1.4127	1.4070	1.4016	1.3964	1.3914	1.3866	1.3819	1.3773	1.3729
380	—	—	—	—	—	—	1.4392	1.4331	1.4273	1.4217	1.4163	1.4111	1.4062	1.4014	1.3967	1.3922	1.3878
400	—	—	—	—	—	—	—	—	—	—	—	1.4255	1.4206	1.4158	1.4112	1.4067	1.4024

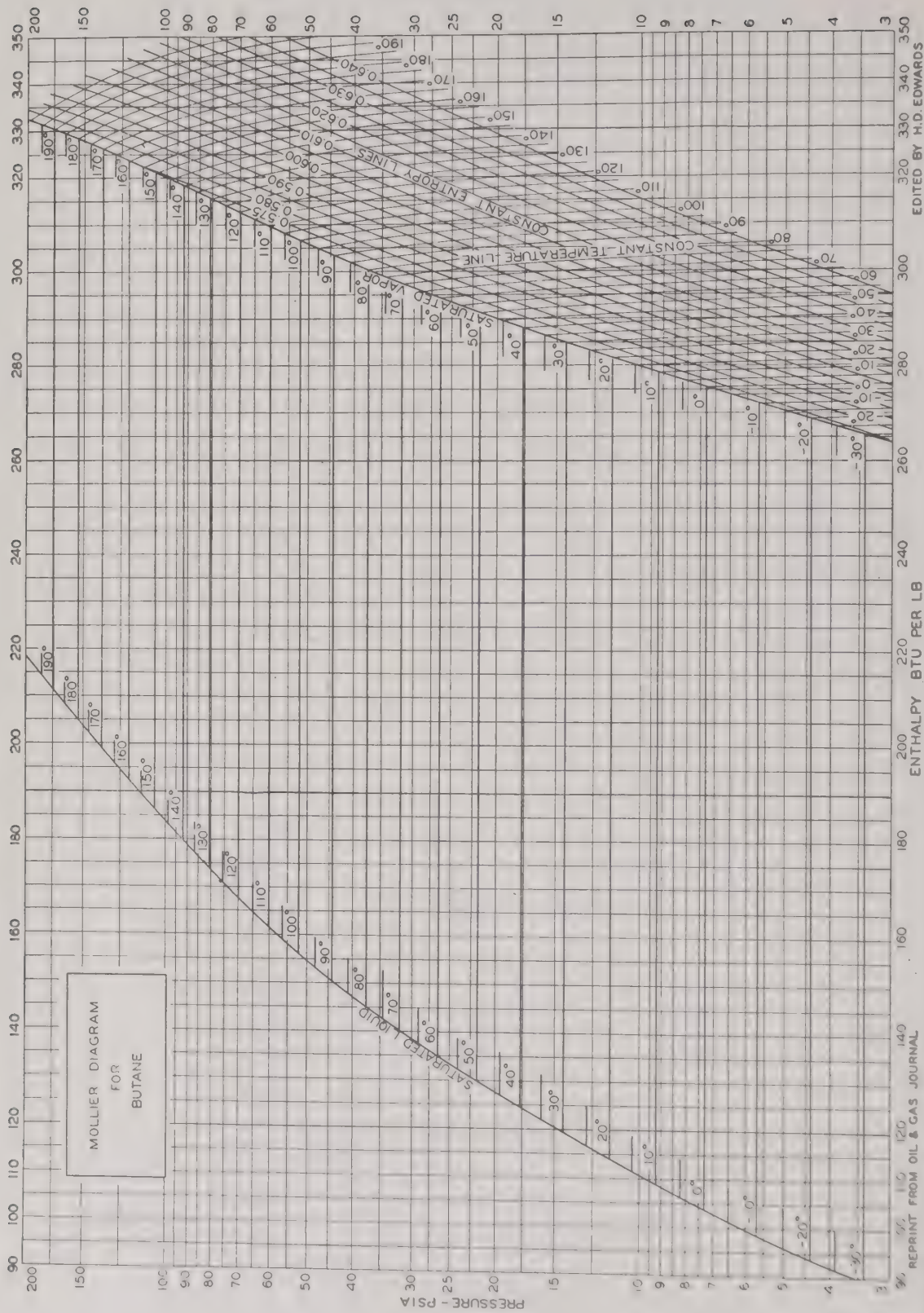
MOLLIER DIAGRAM
FOR
BUTANE
ENTHALPY BTU PER LB
PRESSURE - PSIA
CONSTANT ENTROPY LINES
CONSTANT TEMPERATURE LINE
SATURATED VAPOR
SATURATED LIQUID
EDITED BY H.D. EDWARDS
REPRINT FROM OIL & GAS JOURNAL

Table 5. Butane (Normal)—Properties of Liquid and Saturated Vapor

(Enthalpies and entropies are referred to saturated liquid at −200 F where the values are zero. Refer to Chart 2 for enthalpies and entropies of superheated butane vapor.)

Temp F t	Pressure†		Liquid density†	Vapor, sp vol	Enthalpy** Btu per lb		Entropy** Btu per lb F	
	psia	psig	lb per cu ft $1/v_f$	cu ft per lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
0	7.3	*15.0	38.59	11.1	103.8	275.3	0.226	0.572
5	8.2	*13.2	38.41	9.98	106.2	276.9	0.231	0.572
10	9.2	*11.1	38.24	8.95	108.8	278.5	0.236	0.574
15	10.4	*8.8	38.07	8.05	111.5	280.0	0.242	0.571
20	11.6	*6.3	37.89	7.23	114.0	281.6	0.248	0.571
25	13.0	*3.6	37.72	6.55	116.7	283.1	0.254	0.571
30	14.4	*0.6	37.54	5.90	119.2	284.9	0.260	0.571
35	16.0	1.3	37.37	5.37	121.8	286.4	0.264	0.571
40	17.7	3.0	37.19	4.77	124.2	288.0	0.270	0.571
45	19.6	4.9	37.00	4.47	126.8	289.5	0.276	0.571
50	21.6	6.9	36.82	4.07	129.6	291.2	0.282	0.571
55	23.8	9.1	36.63	3.73	132.1	292.7	0.287	0.570
60	26.3	11.6	36.45	3.40	134.8	294.2	0.293	0.570
65	28.9	14.2	36.24	3.12	137.3	295.9	0.298	0.570
70	31.6	16.9	36.06	2.98	140.1	297.5	0.304	0.570
75	34.5	19.8	35.86	2.65	142.6	298.9	0.310	0.570
80	37.6	22.9	35.65	2.46	145.0	300.3	0.315	0.570
85	40.9	26.2	35.45	2.28	147.8	302.0	0.322	0.570
90	44.5	29.8	35.24	2.10	150.5	303.5	0.326	0.571
95	48.2	33.5	35.04	1.96	153.1	305.0	0.332	0.571
100	52.2	37.5	34.84	1.81	156.2	306.7	0.340	0.571
105	56.4	41.7	34.62	1.70	159.1	308.1	0.346	0.572
110	60.8	46.1	34.41	1.58	161.9	309.5	0.352	0.572
115	65.6	50.9	34.19	1.48	165.0	311.1	0.359	0.572
120	70.8	56.1	33.96	1.37	167.8	312.7	0.365	0.572
125	76.0	61.3	33.77	1.28	171.0	314.0	0.372	0.573
130	81.4	66.7	33.56	1.19	174.0	315.5	0.378	0.573
135	87.0	72.3	33.34	1.11	177.1	317.0	0.385	0.573
140	92.6	77.9	33.14	1.04	179.9	318.2	0.391	0.574
145	100.0	85.3	32.92	0.966	183.1	319.5	0.398	0.574
150	108.0	93.7	32.70	0.897	186.5	321.0	0.405	0.574
155	115.0	100.3	32.43	0.840	189.3	322.3	0.411	0.574
160	122.0	107.3	32.15	0.785	192.7	323.8	0.418	0.575
165	130.0	115.3	31.90	0.733	195.8	325.0	0.425	0.575
170	140.0	125.3	31.62	0.687	199.2	326.1	0.433	0.575
175	150.0	135.3	31.36	0.643	202.1	327.2	0.439	0.575
180	160.0	145.3	31.10	0.602	205.0	328.4	0.445	0.575

† Based on material from Dana Jenkins, Burdick and Timm, published originally in *Refrigerating Engineering*, June, 1926, vol. 12, no. 12, page 402.

* Inches of mercury below one standard atmosphere (29.92 in.).

** From Mollier Diagrams for Butane, W. C. Edmister, Standard Oil Co. (Indiana).

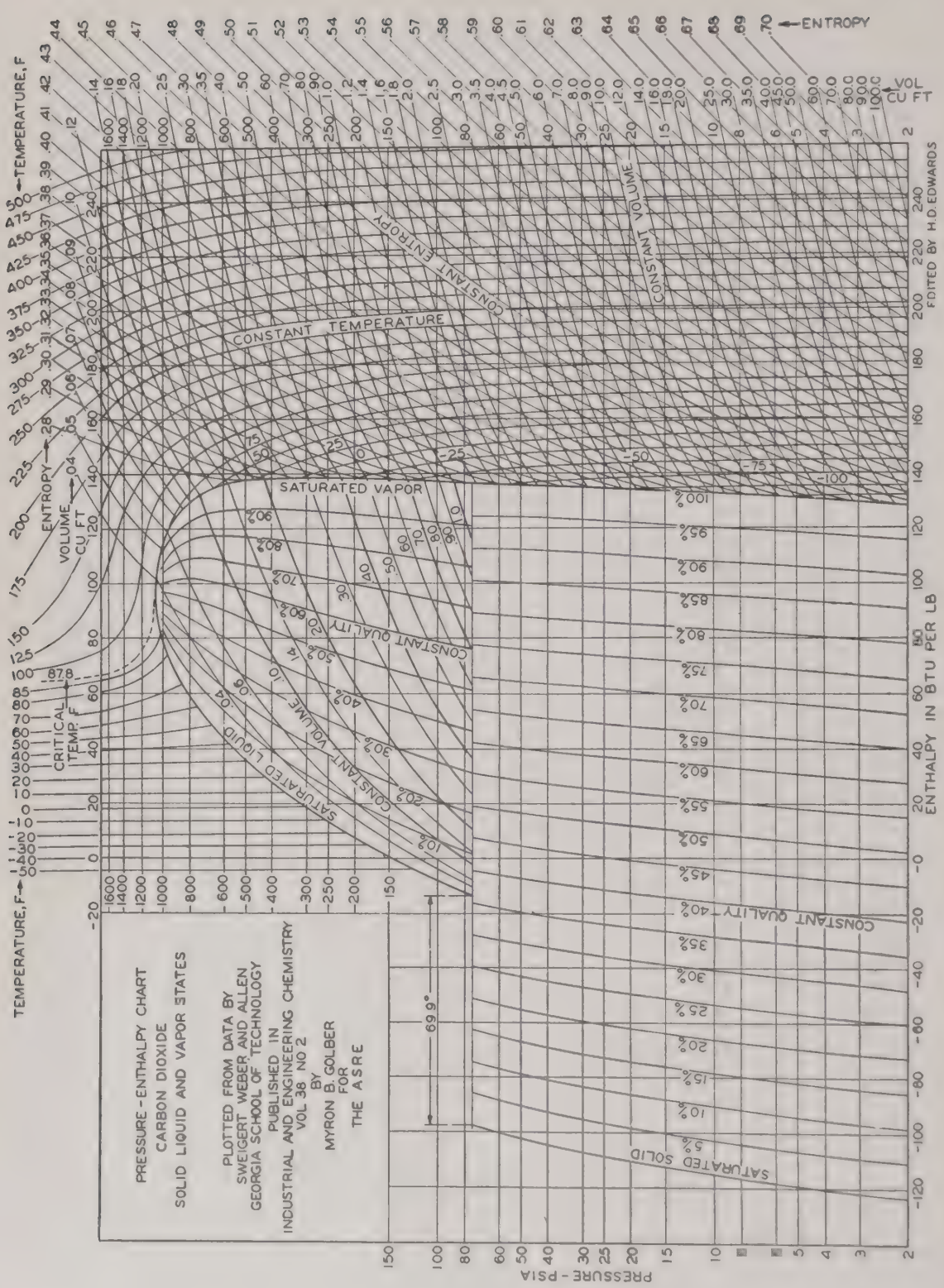
PRESSURE - ENTHALPY CHART
CARBON DIOXIDE
SOLID LIQUID AND VAPOR STATES
PLOTTED FROM DATA BY
SWEIGERT WEBER AND ALLEN
GEORGIA SCHOOL OF TECHNOLOGY
PUBLISHED IN
INDUSTRIAL AND ENGINEERING CHEMISTRY
VOL 38 NO 2
BY
MYRON B. GOLBER
FOR
THE ASRE
EDITED BY H.D.EDWARDS
TEMPERATURE, F
ENTROPY
VOLUME CU FT
CRITICAL TEMP. F
87.8
69.9°
CONSTANT TEMPERATURE
CONSTANT ENTROPY
CONSTANT VOLUME
CONSTANT QUALITY
SATURATED VAPOR
SATURATED LIQUID
SATURATED SOLID
PRESSURE - PSIA
ENTHALPY IN BTU PER LB

Chart 4

Table 6. Carbon Dioxide—Properties of Solid, Liquid and Saturated Vapor

Temp F t	Pressure psia	Pressure psig	Volume cu ft per lb: Solid or liquid v_i or v_f	Volume cu ft per lb: Vapor v_g	Enthalpy, datum −40 F Btu per lb: Solid or liquid h_i or h_f	Enthalpy, datum −40 F Btu per lb: Vapor h_g	Entropy, datum −40 F Btu per lb F: Solid or liquid s_f	Entropy, datum −40 F Btu per lb F: Vapor s_g
Solid and Vapor								
−147	2.14	25.56*	0.01004	35.80	−123.3	128.2	−.3214	+.4832
−145	2.43	24.97*	0.01005	32.40	−122.8	128.5	−.1396	.4792
−140	3.19	23.41*	0.01007	24.50	−121.4	129.3	−.3153	.4691
−135	4.16	21.43*	0.01009	18.70	−120.1	130.0	−.3110	.4593
−130	5.39	18.92*	0.01012	14.74	−118.7	130.7	−.3068	.4500
−125	6.98	15.68*	0.01015	11.56	−117.4	131.3	−.3027	.4409
−120	8.85	11.87*	0.01018	9.13	−116.0	132.1	−.2986	.4318
−115	11.20	7.08*	0.01022	7.27	−114.3	132.7	−.2944	.4230
−110	14.22	0.92*	0.01024	5.85	−113.1	133.3	−.2904	.4145
−109.4	14.67	0.00	0.01025	5.69	−112.9	133.4	−.2898	.4134
−105	17.80	3.13	0.01028	4.72	−111.5	133.9	−.2860	.4062
−100	22.34	7.67	0.01032	3.80	−110.0	134.4	−.2815	.3981
− 95	27.63	12.96	0.01036	3.09	−108.3	134.9	−.2768	.3902
− 90	34.05	19.38	0.01040	2.52	−106.5	135.3	−.2720	.3822
− 85	41.67	27.00	0.01044	2.07	−104.5	135.6	−.2667	.3742
− 80	50.70	36.03	0.01049	1.70	−102.3	135.8	−.2610	.3665
− 79	52.80	38.13	0.01050	1.63	−101.9	135.8	−.2599	.3649
− 78	55.00	40.33	0.01051	1.56	−101.4	135 9	−.2587	.3633
− 77	57.20	42.53	0.01052	1.51	−101.0	135.9	−.2575	.3617
− 76	59.44	44.77	0.01053	1.46	−100.5	135.9	−.2563	.3601
− 75	61.75	47.08	0.01054	1.40	−100.1	135.9	−.2551	.3585
− 74	64.25	49.58	0.01055	1.35	− 99.7	135.9	−.2539	.3570
− 73	66.75	52.08	0.01056	1.30	− 99.2	135.9	−.2528	.3554
− 72	69.40	54.73	0.01057	1.25	− 98.8	136.0	−.2516	.3539
− 71	72.10	57.43	0.01058	1.21	− 98.4	136.0	−.2505	.3523
− 70	74.90	60.23	0.01059	1.17	− 98.0	136.0	−.2494	.3508
− 69.9	75.1	60.4	0.01059	1.16	− 97.8	136.0	−.2493	.3506
			—Triple Point—					
Liquid and Vapor								
− 69.9	75.1	60.4	0.01360	1.1570	− 13.7	136.0	−.0333	.3506
− 68	78.59	63.92	0.01363	1.1095	− 12.8	136.2	−.0312	.3491
− 66	82.42	67.75	0.01369	1.0590	− 11.9	136.3	−.0290	.3475
− 64	86.39	71.72	0.01373	1.0100	− 10.9	136.4	−.0266	.3460
− 62	90.49	75.82	0.01378	0.9650	− 10.1	136.6	−.0243	.3444
− 60	94.75	80.08	0.01384	0.9520	− 9.1	136.7	−.0221	.3429
− 58	99.15	84.48	0.01389	0.8875	− 8.2	136.8	−.0198	.3413
− 56	103.69	89.02	0.01393	0.8520	− 7.3	137.0	−.0175	.3398
− 54	108.40	93.73	0.01398	0.8180	− 6.4	137.1	−.0153	.3383
− 52	113.25	98.58	0.01403	0.7840	− 5.5	137.2	−.0131	.3368
− 50	118.27	103.60	0.01409	0.7500	− 4.6	137.3	−.0109	.3354
− 48	123.45	108.78	0.01414	0.7200	− 3.6	137.5	−.0087	.3339
− 46	128.80	114.13	0.01419	0.6930	− 2.7	137.6	−.0065	.3325
− 44	134.31	119.64	0.01425	0.6660	− 1.8	137.7	−.0043	.3311
− 42	140.00	125.33	0.01430	0.6380	− 0.9	137.8	−.0021	.3297
− 40	145.87	131.20	0.01437	0.6113	0.0	137.9	.0000	.3285
− 38	151.92	137.25	0.01142	0.5880	+ 1.0	138.0	+.0022	.3271
− 36	158.15	143.48	0.01447	0.5650	1.9	138.1	.0043	.3258
− 34	164.56	149.89	0.01454	0.5420	2.8	138.2	.0065	.3245
− 32	171.17	156.50	0.01458	0.5210	3.8	138.3	.0085	.3232
− 30	177.97	163.30	0.01465	0.5025	4.7	138.3	.0106	.3219
− 28	184.97	170.30	0.01472	0.4845	5.6	138.4	.0126	.3205
− 26	192.17	177.50	0.01478	0.4670	6.5	138.5	.0147	.3193
− 24	199.57	184.90	0.01485	0.4500	7.4	138.6	.0168	.3180
− 22	207.19	192.58	0.01491	0.4325	8.3	138.6	.0190	.3167
− 20	215.02	200.35	0.01498	0.4165	9.2	138.7	.0210	.3155
− 18	223.06	208.39	0.01504	0.4015	10.3	138.7	.0231	.3142
− 16	231.32	216.65	0.01511	0.3865	11.2	138.8	.0252	.3130
− 14	239.81	225.14	0.01518	0.3725	12.1	138.8	.0272	.3117
− 12	248.52	233.85	0.01525	0.3590	12.9	138.8	.0293	.3104
− 10	257.46	242.79	0.01533	0.3465	13.9	138.9	.0314	.3091
− 8	266.63	251.96	0.01540	.3345	15.0	138.9	.0335	.3079
− 6	276.05	261.38	0.01547	.3228	15.9	138.9	.0356	.3067
− 4	285.70	271.03	0.01555	.3118	16.9	138.9	.0376	.3054
− 2	295.61	280.94	0.01563	.3012	17.9	138.9	.0397	.3042
0	305.76	291.09	0.01571	0.2905	18.8	138.9	.0419	.3030

* Inches of mercury below one atmosphere.

Table 6. Carbon Dioxide—Properties of Solid, Liquid and Saturated Vapor (Concluded)

Temp F t	Pressure		Volume cu ft per lb		Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	Solid or liquid v_i or v_f	Vapor v_g	Solid or liquid h_i or h_f	Vapor h_g	Solid or liquid s_f	Vapor s_g
+ 2	316.2	301.5	0.01579	0.2800	19.8	138.9	.0441	.3018
4	326.8	312.1	0.01587	.2708	20.8	138.9	.0462	.3006
6	337.7	323.0	0.01597	.2610	21.8	138.9	.0483	.2994
8	348.9	334.2	0.01605	.2520	22.9	138.8	.0503	.2982
10	360.4	345.7	0.01614	.2435	24.0	138.8	.0525	.2970
12	372.1	357.5	0.01623	0.2350	25.0	138.7	.0547	.2958
14	384.1	369.4	0.01632	.2274	26.2	138.7	.0569	.2945
16	396.4	381.7	0.01642	2195	27.3	138.6	.0591	.2933
18	409.0	394.3	0.01650	.2120	28.5	138.6	.0613	.2921
20	421.8	407.1	0.01662	.2048	29.6	138.5	.0636	2909
22	435.0	420.3	0.01672	0.1978	30.8	138.4	.0660	.2897
24	448.4	433.7	0.01684	.1910	31.8	138.3	.0684	.2885
26	462.2	447.5	0.01695	.1845	33.1	138.1	.0707	.2873
28	476.3	461.6	0.01707	.1782	34.3	138.0	.0730	.2861
30	490.6	475.9	0.01719	.1720	35.6	137.8	.0753	.2849
32	505.3	490.6	0.01731	0.1663	36.8	137.7	.0778	.2837
34	520.3	505.6	0.01743	.1605	38.0	137.5	.0800	.2883
36	535.7	521.0	0.01757	.1550	39.3	137.3	.0823	.2807
38	551.3	536.6	0.01771	.1495	40.5	137.0	.0848	.2791
40	567.3	552.6	0.01786	.1442	41.8	136.8	.0872	.2775
42	583.7	569.0	0.01801	.1390	43.1	136.5	.0898	.2759
44	600.4	585.7	0.01817	.1342	44.4	136.2	.0922	.2745
46	617.5	602.8	0.01834	.1298	45.7	135.8	.0947	.2730
48	634.9	620.2	0.01842	.1250	47.1	135.5	.0972	.2715
50	652.7	638.0	0.01867	.1204	48.5	135.1	.1000	.2699
52	670.8	656.1	0.01887	0.1163	49.9	134.6	.1026	.2681
54	689.4	674.7	0.01906	.1122	51.3	134.0	.1052	.2664
56	708.3	693.6	0.01926	.1080	52.8	133.4	.1080	.2645
58	727.6	712.9	0.01948	.1038	54.2	132.8	.1108	.2627
60	747.4	732.7	0.01970	.0995	55.7	132.2	.1136	.2606
62	767.5	752.8	0.01997	0.0960	57.2	131.5	.1164	.2585
64	788.1	773.4	0.02012	.0920	58.8	130.7	.1192	.2563
66	809.1	794.4	0.02049	.0880	60.4	129.8	.1221	.2540
68	830.6	815.9	0.02079	.0842	62.0	128.9	.1250	.2516
70	852.5	837.8	0.02109	.0800	63.7	127.8	.1282	.2485
72	874.9	860.2	0.02146	0.0760	65.6	126.5	.1316	.2455
74	897.7	883.0	0.02188	.0720	67.5	124.7	.1351	.2423
76	921.1	906.4	0.02240	.0680	69.6	122.9	.1390	.2386
78	945.0	930.3	0.02300	.0640	71.8	121.1	.1430	.2347
80	969.3	954.6	0.02370	.0600	74.0	119.0	.1469	.2305
82	994.2	979.5	0.02458	0.0560	76.6	116.6	.1514	.2258
84	1019.7	1005.0	0.02556	.0520	79.6	114.0	.1564	.2205
86	1045.7	1031.0	0.02686	.0479	83.4	110.5	.1636	.2133
87.8	1072.1	1057.4	0.03453	.0345	97.1	97.1	.1880	.1880

Liquid and Vapor

Table 7. Carbon Dioxide—Specific Volume of Superheated Vapor (cu ft per lb)

Pressure psia	Temperature											
	−75 F	−50 F	0 F	50 F	100 F	150 F	200 F	250 F	300 F	350 F	400 F	450 F
.20	469.4	499.9	560.9	621.9	682.8	743.8	804.8	865.8	926.7	987.7	—	—
.30	312.9	333.3	373.9	414.6	455.2	495.9	536.5	577.2	617.8	658.5	699.1	739.7
.40	234.7	249.9	280.4	310.9	341.4	371.9	402.4	432.9	463.3	493.8	524.3	554.8
.50	187.7	199.9	224.3	248.7	273.1	297.5	321.9	346.3	370.7	395.0	419.4	443.8
.60	156.4	166.6	186.9	207.3	227.6	247.9	268.2	288.6	308.9	329.2	349.5	369.8
.70	134.1	141.9	160.2	177.6	195.1	212.5	229.9	247.3	264.7	282.2	299.6	317.0
.80	117.4	125.0	140.2	155.5	170.7	186.0	201.2	216.5	231.7	246.9	262.2	277.4
.90	104.3	111.1	124.7	138.2	151.8	165.3	178.9	192.4	206.0	219.5	233.0	246.5
1.00	93.90	100.0	112.2	124.4	136.6	148.8	161.0	173.2	185.4	197.5	209.7	221.9
2.0	46.82	49.96	56.03	62.12	68.29	74.39	80.48	86.58	92.67	98.77	104.8	110.09
3.0	31.18	33.28	37.32	41.40	45.49	49.55	53.64	57.72	61.78	65.85	69.91	73.97
4.0	23.38	24.93	27.98	31.03	34.09	37.13	40.22	43.29	46.33	49.38	52.43	55.48
5.0	18.66	19.91	22.37	24.82	27.26	29.72	32.17	34.61	39.06	39.51	41.93	44.38
6.0	15.53	16.56	18.63	20.67	22.71	24.76	26.80	28.84	30.87	32.92	34.94	36.98
8.0	11.62	12.40	13.96	15.49	17.02	18.56	20.09	21.62	23.14	24.60	26.20	27.74
10.0	9.280	9.902	11.15	12.38	13.61	14.84	16.06	17.29	18.51	19.74	20.96	22.19
12.0	7.715	8.236	9.280	10.31	11.33	12.37	13.38	14.41	15.42	16.44	17.46	18.49
16.0	5.769	6.177	6.944	7.716	8.485	9.266	10.03	10.80	11.56	12.33	13.09	13.86
20.0	4.586	4.904	5.542	6.119	6.778	7.407	8.016	8.633	9.247	9.861	10.47	11.09
24.0	3.803	4.070	4.607	5.125	5.640	6.167	6.674	7.190	7.701	8.215	8.724	9.236
28.0	3.246	3.471	3.939	4.385	4.826	5.282	5.718	6.160	6.599	7.039	7.475	7.918
32.0	2.825	3.028	3.439	3.829	4.217	4.617	4.999	5.387	5.773	6.157	6.541	6.925
36.0	2.499	2.681	3.050	3.398	3.743	4.101	4.441	4.785	5.130	5.470	5.812	6.156
40.0	2.239	2.404	2.738	3.053	3.363	3.688	3.993	4.304	4.615	4.922	5.230	5.538
44.0	2.025	2.177	2.483	2.771	3.054	3.349	3.628	3.972	4.193	4.473	4.753	5.034
48.0	1.847	1.987	2.270	2.535	2.795	3.068	3.323	3.587	3.843	4.099	4.355	4.614
52.0	1.696	1.827	2.091	2.335	2.576	2.829	3.065	3.306	3.545	3.783	4.019	4.258
60.0	1.456	1.571	1.803	2.016	2.225	2.448	2.652	2.862	3.070	3.276	3.481	3.689
70.0	—	1.329	1.536	1.720	1.901	2.094	2.269	2.450	2.628	2.806	2.982	3.071
80.0	—	1.154	1.335	1.498	1.657	1.828	1.982	2.141	2.298	2.453	2.608	2.764
90.0	—	1.015	1.179	1.325	1.476	1.621	1.758	1.900	2.040	2.178	2.317	2.457
100.0	—	0.905	1.055	1.187	1.316	1.456	1.579	1.708	1.834	1.960	2.083	2.210
120.0	—	—	0.8655	0.9799	1.088	1 208	1.311	1.420	1.525	1.630	1.734	1.840
160.0	—	—	0.6305	0.7207	0.8033	0.8986	0.9760	1.059	1.139	1.219	1.297	1.378
200.0	—	—	0.4891	0.5652	0.6376	0.7125	0.7748	0.8424	0.9075	0.9720	1.035	1.100
240.0	—	—	0.3948	0.4614	0.5237	0.5886	0.6407	0.6981	0.7532	0.8074	0.8604	0.9152
280.0	—	—	0.3286	0.3865	0.4425	0.4994	0.5448	0.5950	0.6428	0.6896	0.7358	0.7833
320.0	—	—	—	0.3298	0.3815	0.4322	0.4729	0.5177	0.5601	0.6015	0.6422	0.6840
360.0	—	—	—	0.2858	0.3341	0.3780	0.4171	0.4576	0.4958	0.5329	0.5693	0.6071
400.0	—	—	—	0.2505	0.2961	0.3383	0.3724	0.4095	0.4443	0.4781	0.5110	0.5455
440.0	—	—	—	0.2216	0.2652	0.3040	0.3358	0.3701	0.4022	0.4332	0.4633	0.4950
520.0	—	—	—	0.1772	0.2174	0.2513	0.2795	0.3096	0.3374	0.3641	0.3901	0.4174
560.0	—	—	—	0.1598	0.1986	0.2304	0.2574	0.2858	0.3119	0.3369	0.3613	0.3869
600.0	—	—	—	0.1452	0.1823	0.2123	0.2383	0.2652	0.2898	0.3135	0.3363	0.3605
650.0	—	—	—	—	0.1648	0.1928	0.2176	0.2430	0.2661	0.2881	0.3094	0.3323
700.0	—	—	—	—	0.1453	0.1760	0.2000	0.2239	0.2457	0.2664	0.2864	0.3075
750.0	—	—	—	—	0.1320	0.1615	0.1846	0.2074	0.2281	0.2476	0.2664	0.2865
800.0	—	—	—	—	0.1196	0.1483	0.1712	0.1930	0.2126	0.2311	0.2489	0.2681
850.0	—	—	—	—	0.1089	0.1373	0.1594	0.1803	0.1990	0.2166	0.2335	0.2518
900.0	—	—	—	—	0.1000	0.1273	0.1489	0.1690	0.1869	0.2037	0.2198	0.2372
950.0	—	—	—	—	—	0.1183	0.1394	0.1588	0.1761	0.1921	0.2075	0.2243
1000.0	—	—	—	—	—	0.1101	0.1310	0.1497	0.1663	0.1819	0.1966	0.2126
1100.0	—	—	—	—	—	—	0.1164	0.1339	0.1495	0.1646	0.1778	0.1925
1200.0	—	—	—	—	—	—	0.1042	0.1208	0.1356	0.1493	0.1621	0.1756
1300.0	—	—	—	—	—	—	—	0.1100	0.1237	0.1368	0.1488	0.1614
1400.0	—	—	—	—	—	—	—	—	0.1136	0.1260	0.1375	0.1492
1500.0	—	—	—	—	—	—	—	—	0.1049	0.1167	0.1277	0.1386
1600.0	—	—	—	—	—	—	—	—	—	0.1086	0.1191	0.1294
1700.0	—	—	—	—	—	—	—	—	—	0.1014	0.1155	0.1213
1800.0	—	—	—	—	—	—	—	—	—	—	0.1047	0.1140
1900.0	—	—	—	—	—	—	—	—	—	—	—	0.1075
2000.0	—	—	—	—	—	—	—	—	—	—	—	0.1017

Table 8. Carbon Dioxide—Enthalpy of Superheated Vapor (Btu per lb from −40 F)

Pressure psia	Temperature											
	−75 F	−50 F	0 F	50 F	100 F	150 F	200 F	250 F	300 F	350 F	400 F	450 F
0.20	140.0	144.8	154.6	164.5	174.8	185.2	195.9	206.9	218.2	229.7	—	—
2.0	140.0	144.8	154.6	164.5	174.8	185.2	195.9	206.9	218.2	229.7	241.5	253.6
3.0	139.9	144.7	154.5	164.4	174.8	185.2	195.9	206.9	218.2	229.7	241.5	253.6
4.0	139.8	144.7	154.5	164.4	174.7	185.2	195.9	206.9	218.2	229.7	241.5	253.6
5.0	139.8	144.6	154.4	164.4	174.7	185.1	195.9	206.9	218.2	229.7	241.5	253.6
6.0	139.7	144.5	154.4	164.3	174.6	185.1	195.9	206.9	218.2	229.7	241.5	253.6
8.0	139.6	144.5	154.3	164.2	174.6	185.1	195.8	206.8	218.1	229.6	241.5	253.6
10.0	139.4	144.3	154.1	164.1	174.5	185.0	195.8	206.8	218.1	229.6	241.4	253.6
12.0	139.3	144.2	154.0	164.0	174.4	185.0	195.7	206.7	218.1	229.6	241.4	253.5
16.0	139.0	144.0	153.8	163.8	174.2	184.8	195.7	206.7	218.0	229.5	241.4	253.5
20.0	138.7	143.8	153.6	163.6	174.1	184.7	195.6	206.6	217.9	229.5	241.3	253.5
24.0	138.5	143.6	153.4	163.4	173.9	184.6	195.5	206.5	217.9	229.4	241.3	253.4
28.0	138.2	143.4	153.2	163.2	173.8	184.5	195.4	206.5	217.8	229.4	241.3	253.4
32.0	137.9	143.2	153.0	163.1	173.6	184.4	195.4	206.4	217.8	229.3	241.2	253.4
36.0	137.7	142.9	152.8	162.9	173.5	184.3	195.3	206.3	217.7	229.3	241.2	253.3
40.0	137.4	142.7	152.6	162.7	173.3	184.2	195.2	206.3	217.7	229.3	241.1	253.3
44.0	137.2	142.5	152.4	162.5	173.2	184.1	195.2	206.2	217.6	229 2	241.1	253.3
48.0	136.9	142.3	152.2	162.3	173.0	184.0	195.1	206.1	217.5	229.1	241.0	253.2
52.0	136.6	142.1	152.0	162.2	172.9	183.9	195.0	206.1	217.5	229.1	241.0	253.2
56.0	136.4	141.9	151.8	162.0	172.8	183.8	195.0	206.0	217.4	229.1	241.0	253.2
60.0	136.1	141.7	151.6	161.8	172.6	183.7	194.9	205.9	217.3	228.9	240.9	253.1
70.0	—	141.2	151.1	161.3	172.2	183.5	194.7	205.7	217.2	228.8	240.8	253.0
80.0	—	140.6	150.6	160.9	171.9	183.2	194.5	205.6	217.0	228.7	240.7	252.9
90.0	—	140.1	150.1	160.4	171.5	183.0	194.3	205.4	216.9	228.6	240.6	252.8
100.0	—	139.6	149.5	160.0	171.1	182.7	194.1	205.2	216.8	228.5	240.5	252.7
120.0	—	—	148.5	159.0	170.4	182.2	193.8	204.9	216.5	228.3	240.3	252.6
140.0	—	—	147.5	158.1	169.6	181.7	193.5	204.6	216.2	228.0	240.1	252.4
160.0	—	—	146.5	157.2	168.9	181.2	193.1	204.2	215.9	227.8	239.9	252.2
180.0	—	—	145.5	156.3	168.1	180.7	192.8	203.9	215.6	227.5	239.7	252.1
200.0	—	—	144.5	155.4	167.4	180.2	192.4	203.5	215.3	227.3	239.5	252.0
220.0	—	—	143.4	154.5	166.6	179.7	192.1	203.2	215.0	227.0	239.3	251.8
240.0	—	—	142.4	153.5	165.9	179.2	191.7	202.9	214.8	226.8	239.1	251.6
260.0	—	—	141.4	152.6	165.2	178.7	191.4	202.5	214.5	226.6	238.9	251.5
280.0	—	—	140.4	151.7	164.4	178.2	191.0	202.2	214.2	226.3	238.6	251.3
300.0	—	—	—	150.8	163.7	177.7	190.7	201.9	213.9	226.1	238.4	251.1
320.0	—	—	—	149.9	162.9	177.2	190.3	201.5	213.6	225.8	238.2	251.0
340.0	—	—	—	149.0	162.2	176.7	190.0	201.2	213.3	225.6	238.0	250.8
360.0	—	—	—	148.0	161.4	176.2	189.6	200.8	213.1	225.4	237.8	250.5
380.0	—	—	—	147.1	160.7	175.7	189.3	200.5	212.8	225.1	237.6	250.3
400.0	—	—	—	146.2	159.9	175.2	188.9	200.2	212.5	224.9	237.4	250.2
420.0	—	—	—	145.3	159.2	174.7	188.6	199.8	212.2	224.6	237.2	250.0
440.0	—	—	—	144.4	158.4	174.2	188.2	199.5	211.9	224.4	237.0	249.8
480.0	—	—	—	142.5	156.9	173.2	187.5	198.8	211.3	223.9	236.6	249.5
520.0	—	—	—	140.7	155.5	172.2	186.8	198.2	210.8	223.4	236.2	249.1
560.0	—	—	—	138.9	154.0	171.2	186.1	197.5	210.2	223.0	235.8	248.8
600.0	—	—	—	137.1	152.5	170.2	185.4	196.8	209.6	222.5	235.4	248.5
700.0	—	—	—	—	148.8	167.7	183.7	195.1	208.2	221.3	234.4	247.6
800.0	—	—	—	—	145.0	165.2	181.9	193.4	206.8	220.1	233.3	246.7
900.0	—	—	—	—	141.3	162.7	180.2	191.7	205.3	218.9	232.3	245.9
1000.0	—	—	—	—	—	160.2	178.4	190.1	203.9	217.7	231.3	245.0
1100.0	—	—	—	—	—	—	177.5	188.4	202.5	216.5	230.3	244.1
1200.0	—	—	—	—	—	—	175.2	186.7	201.0	215.3	229.3	243.3
1300.0	—	—	—	—	—	—	—	185.0	199.6	214.1	228.2	242.4
1400.0	—	—	—	—	—	—	—	—	198.2	212.9	227.2	241.6
1500.0	—	—	—	—	—	—	—	—	196.7	211.7	225.9	240.7
1600.0	—	—	—	—	—	—	—	—	—	210.5	224.4	239.8
1700.0	—	—	—	—	—	—	—	—	—	209.0	222.8	239.0
1800.0	—	—	—	—	—	—	—	—	—	—	220.8	238.1
1900.0	—	—	—	—	—	—	—	—	—	—	—	237.3
2000.0	—	—	—	—	—	—	—	—	—	—	—	236.2

Table 9. Carbon Dioxide—Entropy of Superheated Vapor (Btu per lb F from −40 F)

Pressure psia	Temperature											
	−75 F	−50 F	0 F	50 F	100 F	150 F	200 F	250 F	300 F	350 F	400 F	450 F
.2	.6276	.6396	.6616	.6820	.7010	.7188	.7356	.7516	.7669	.7815	—	—
.3	.6093	.6213	.6433	.6637	.6827	.7005	.7173	.7333	.7486	.7632	.7772	.7909
.4	.5963	.6083	.6303	.6507	.6697	.6875	.7043	.7203	.7356	.7502	.7642	.7779
.6	.5780	.5900	.6120	.6324	.6512	.6692	.6860	.7020	.7173	.7319	.7459	.7596
.8	.5650	.5770	.5990	.6194	.6384	.6562	.6730	.6890	.7043	.7189	.7329	.7466
1.0	.5550	.5670	.5890	.6094	.6284	.6462	.6630	.6790	.6943	.7089	.7229	.7366
1.2	.5468	.5588	.5808	.6012	.6202	.6380	.6548	.6708	.6861	.7007	.7147	.7284
1.4	.5398	.5518	.5738	.5942	.6132	.6310	.6478	.6638	.6791	.6937	.7077	.7214
1.6	.5338	.5458	.5778	.5882	.6072	.6250	.6418	.6578	.6731	.6877	.7017	.7154
1.8	.5285	.5405	.5625	.5829	.6019	.6197	.6365	.6525	.6678	.6824	.6964	.7101
2.0	.5237	.5357	.5577	.5781	.5971	.6149	.6317	.6477	.6630	.6776	.6916	.7053
3.0	.5054	.5174	.5394	.5598	.5788	.5966	.6134	.6294	.6447	.6593	.6733	.6870
4.0	.4925	.5045	.5265	.5469	.5659	.5837	.6005	.6165	.6318	.6464	.6604	.6741
6.0	.4742	.4862	.5082	.5286	.5476	.5654	.5822	.5982	.6145	.6281	.6421	.6558
8.0	.4612	.4732	.4952	.5156	.5346	.5524	.5692	.5852	.6005	.6151	.6291	.6428
10.0	.4511	.4631	.4851	.5055	.5245	.5423	.5591	.5751	.5904	.6050	.6190	.6327
12.0	.4429	.4549	.4769	.4973	.5163	.5341	.5509	.5669	.5822	.5968	.6108	.6245
14.0	.4358	.4479	.4699	.4903	.5093	.5271	.5439	.5599	.5752	.5898	.6038	.6175
16.0	.4297	.4418	.4639	.4843	.5033	.5211	.5379	.5539	.5692	.5838	.5978	.6115
20.0	.4195	.4316	.4537	.4742	.4932	.5110	.5278	.5438	.5591	.5737	.5877	.6014
28.0	.4038	.4159	.4381	.4587	.4778	.4957	.5126	.5287	.5440	.5586	.5726	.5863
36.0	.3918	.4040	.4263	.4469	.4661	.4841	.5010	.5171	.5325	.5472	.5612	.5749
44.0	.3818	.3941	.4166	.4374	.4567	.4747	.4917	.5079	.5233	.5380	.5521	.5659
52.0	.3722	.3850	.4079	.4289	.4484	.4666	.4837	.4999	.5154	.5302	.5443	.5581
60.0	.3579	.3739	.3991	.4210	.4409	.4593	.4766	.4930	.5086	.5234	.5376	.5515
70.0	—	.3638	.3898	.4132	.4335	.4521	.4695	.4860	.5016	.5164	.5306	.5445
80.0	—	.3556	.3822	.4062	.4268	.4457	.4633	.4798	.4955	.5104	.5246	.5385
90.0	—	.3487	.3759	.4002	.4211	.4401	.4578	.4744	.4901	.5050	.5193	.5332
100.0	—	.3427	.3702	.3949	.4160	.4351	.4529	.4695	.4853	.5002	.5145	.5284
140.0	—	—	.3525	.3783	.3998	.4191	.4370	.4537	.4696	.4846	.4990	.5130
180.0	—	—	.3368	.3643	.3871	.4069	.4250	.4419	.4580	.4730	.4875	.5014
220.0	—	—	.3232	.3526	.3765	.3969	.4152	.4324	.4485	.4637	.4782	.4922
260.0	—	—	.3122	.3421	.3674	.3883	.4075	.4244	.4407	.4559	.4706	.4846
300.0	—	—	—	.3340	.3591	.3807	.3997	.4173	.4338	.4493	.4640	.4780
340.0	—	—	—	.3255	.3514	.3736	.3932	.4112	.4279	.4435	.4582	.4722
380.0	—	—	—	.3173	.3439	.3668	.3870	.4054	.4224	.4381	.4529	.4670
400.0	—	—	—	.3135	.3405	.3637	.3842	.4028	.4199	.4357	.4506	.4647
420.0	—	—	—	.3098	.3371	.3606	.3813	.4001	.4173	.4332	.4482	.4624
440.0	—	—	—	.3060	.3337	.3575	.3784	.3974	.4158	.4308	.4459	.4601
480.0	—	—	—	.2993	.3276	.3519	.3732	.3925	.4102	.4264	.4415	.4559
520.0	—	—	—	.2926	.3216	.3465	.3683	.3880	.4059	.4224	.4377	.4522
560.0	—	—	—	.2864	.3159	.3413	.3635	.3835	.4017	.4184	.4339	.4486
600.0	—	—	—	.2798	.3101	.3361	.3587	.3791	.3976	.4146	.4303	.4455
650.0	—	—	—	—	.3047	.3312	.3541	.3747	.3934	.4106	.4264	.4410
700.0	—	—	—	—	.2994	.3264	.3496	.3705	.3895	.4069	.4228	.4378
750.0	—	—	—	—	.2939	.3214	.3450	.3662	.3854	.4030	.4190	.4341
800.0	—	—	—	—	.2889	.3169	.3409	.3624	.3819	.3997	.4158	.4310
850.0	—	—	—	—	.2839	.3124	.3368	.3586	.3783	.3963	.4126	.4279
900.0	—	—	—	—	.2788	.3079	.3327	.3548	.3748	.3930	.4095	.4249
950.0	—	—	—	—	—	.3038	.3289	.3512	.3715	.3899	.4066	.4221
1000.0	—	—	—	—	—	.2996	.3250	.3475	.3681	.3867	.4036	.4193
1100.0	—	—	—	—	—	—	.3183	.3413	.3622	.3811	.3983	.4142
1200.0	—	—	—	—	—	—	.3121	.3357	.3569	.3761	.3936	.4097
1300.0	—	—	—	—	—	—	—	.3308	.3524	.3719	.3895	.4057
1400.0	—	—	—	—	—	—	—	—	.3481	.3679	.3856	.4020
1500.0	—	—	—	—	—	—	—	—	.3437	.3639	.3817	.3983
1600.0	—	—	—	—	—	—	—	—	—	.3600	.3780	.3948
1700.0	—	—	—	—	—	—	—	—	—	.3561	.3744	.3913
1800.0	—	—	—	—	—	—	—	—	—	—	.3708	.3879
1900.0	—	—	—	—	—	—	—	—	—	—	—	.3841
2000.0	—	—	—	—	—	—	—	—	—	—	—	.3807

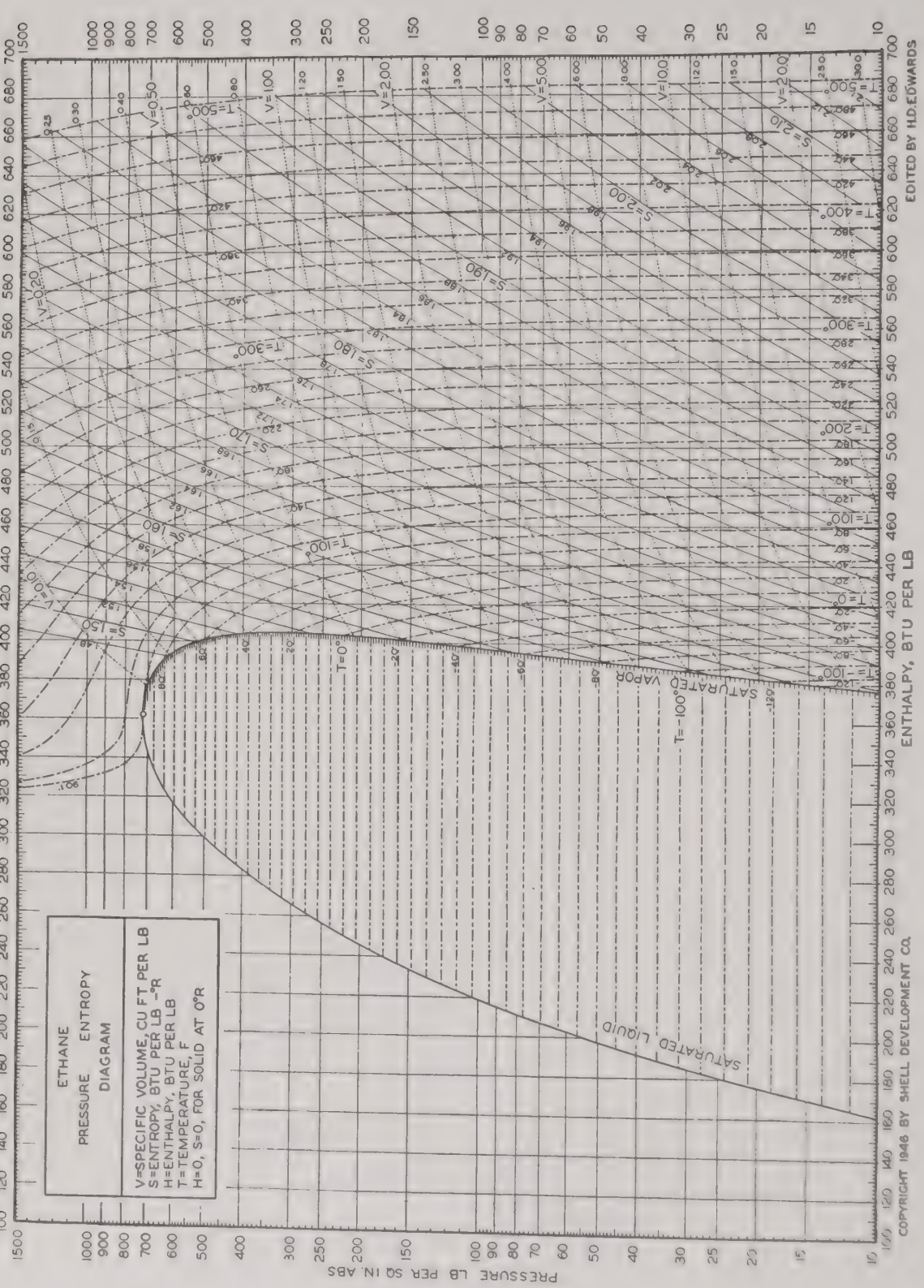
ETHANE
PRESSURE ENTROPY DIAGRAM
V=SPECIFIC VOLUME, CU FT PER LB
S=ENTROPY, BTU PER LB -°R
H=ENTHALPY, BTU PER LB
T=TEMPERATURE, F
H=0, S=0, FOR SOLID AT 0°R
SATURATED LIQUID
SATURATED VAPOR
T=-100°
T=0°
PRESSURE LB PER SQ IN. ABS
ENTHALPY, BTU PER LB
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Chart 5

Table 10. Ethane—Properties of Liquid and Saturated Vapor

Temp F t	Pressure psia	Pressure psig	Liquid, density lb/cu ft $1/v_f$	Vapor, sp vol cu ft/lb v_g	Enthalpy, datum −148 F Btu per lb: Liquid h_f	Enthalpy, datum −148 F Btu per lb: Vapor h_g	Entropy, datum −148 F Btu per lb F: Liquid s_f	Entropy, datum −148 F Btu per lb F: Vapor s_g
−148.0	7.62	14.40*	35.00	14.1	0	219.1	0.000	0.703
−144.4	8.60	12.42*	34.85	12.5	1.9	219.7	0.006	0.697
−140.8	9.67	10.24*	34.70	11.2	3.8	220.3	0.012	0.691
−137.2	10.85	7.84*	34.55	10.1	5.8	220.9	0.018	0.685
−133.6	12.17	5.15*	34.38	9.05	7.7	221.5	0.024	0.680
−130.0	13.60	2.24*	34.22	8.16	9.7	222.1	0.030	0.674
−126.4	15.16	+0.46	34.06	7.41	11.7	222.8	0.036	0.670
−122.8	16.86	2.16	33.90	6.73	13.7	223.6	0.042	0.665
−119.2	18.70	4.00	33.74	6.13	15.8	224.2	0.048	0.660
−115.6	20.69	5.99	33.58	5.62	17.9	224.9	0.054	0.656
−112.0	22.85	8.15	33.40	5.14	19.9	225.6	0.060	0.652
−108.4	25.17	10.47	33.24	4.72	22.1	226.4	0.066	0.648
−104.8	27.67	12.97	33.07	4.33	24.2	227.2	0.072	0.644
−101.2	30.36	15.66	32.90	3.98	26.4	227.9	0.078	0.641
− 97.6	33.23	18.53	32.72	3.66	28.5	228.7	0.084	0.637
− 94.0	36.33	21.63	32.55	3.39	30.7	229.6	0.090	0.634
− 90.4	39.63	24.93	32.38	3.13	33.0	230.5	0.096	0.631
− 86.8	43.14	28.44	32.20	2.90	35.2	231.2	0.102	0.628
− 83.2	46.88	32.18	32.03	2.68	37.5	231.9	0.108	0.624
− 79.6	50.87	36.18	31.86	2.48	39.8	232.8	0.114	0.622
− 76.0	55.09	40.39	31.68	2.30	42.2	233.7	0.120	0.619
− 72.4	59.58	44.88	31.50	2.13	44.5	234.6	0.126	0.617
− 68.8	64.33	49.63	31.32	1.98	46.9	235.5	0.132	0.615
− 65.2	69.35	54.65	31.14	1.84	49.3	236.3	0.139	0.614
− 61.6	74.66	59.96	30.96	1.71	51.8	237.2	0.146	0.612
− 58.0	80.28	65.58	30.76	1.60	54.2	238.2	0.151	0.609
− 54.4	86.15	71.45	30.58	1.48	56.7	239.0	0.157	0.607
− 50.8	92.36	77.66	30.38	1.38	59.2	239.9	0.163	0.605
− 47.2	98.89	84.19	30.18	1.29	61.8	240.9	0.169	0.603
− 43.6	105.7	91.0	29.98	1.21	64.4	241.9	0.175	0.602
− 40.0	112.9	98.2	29.78	1.136	67.0	242.8	0.181	0.600
− 36.4	120.5	105.8	29.58	1.064	69.6	243.7	0.187	0.598
− 32.8	128.4	113.7	29.37	0.996	72.3	244.6	0.194	0.597
− 29.2	136.7	122.0	29.16	0.936	75.0	245.5	0.200	0.596
− 25.8	145.3	130.6	28.94	0.882	77.7	246.4	0.206	0.595
− 22.0	154.4	139.7	28.72	0.831	80.5	247.4	0.212	0.594
− 18.4	163.8	149.1	28.50	0.782	83.4	248.3	0.219	0.593
− 14.8	173.7	159.0	28.27	0.737	86.2	249.1	0.225	0.591
− 11.2	184.0	169.3	28.04	0.695	89.1	249.9	0.231	0.590
− 7.6	194.7	180.0	27.80	0.654	92.0	250.8	0.238	0.589
− 4.0	205.9	191.2	27.57	0.617	95.0	251.6	0.244	0.588
− 0.4	217.6	202.9	27.33	0.582	98.0	252.4	0.250	0.587
+ 3.2	229.7	215.0	27.08	0.549	101.1	253.2	0.257	0.586
6.8	242.3	227.6	26.84	0.518	104.2	254.0	0.263	0.584
10.4	255.4	240.7	26.58	0.490	107.4	254.7	0.270	0.583
14.0	269.1	254.4	26.32	0.463	110.7	255.4	0.277	0.582
17.6	283.2	268.5	26.06	0.437	114.0	256.1	0.283	0.581
21.2	298.0	283.3	25.78	0.412	117.4	256.7	0.290	0.580
24.8	313.3	298.6	25.51	0.388	120.9	257.4	0.297	0.579
28.4	329.2	314.5	25.23	0.366	124.5	258.1	0.304	0.578
32.0	345.7	331.0	24.95	0.346	128.2	258.7	0.312	0.577
35.6	362.7	348.0	24.65	0.328	132.0	259.3	0.319	0.576
39.2	380.5	355.8	24.34	0.311	135.9	259.9	0.327	0.576
42.8	398.8	384.1	24.00	0.296	140.0	260.6	0.335	0.575
46.4	417.8	403.1	23.66	0.282	144.2	261.0	0.343	0.574
50.0	437.5	422.8	23.29	0.267	148.5	261.3	0.351	0.573
53.6	457.9	443.2	22.90	0.251	153.0	262	0.360	0.572
57.2	479.0	464.3	22.48	0.233	157	262	0.368	0.571
60.8	500.9	486.2	22.03	0.213	161	261	0.376	0.568
64.4	523.5	508.8	21.54	0.199	166	261	0.385	0.566
68.0	546.9	532.2	21.00	0.185	170	260	0.393	0.563
71.6	571.1	556.4	20.46	0.171	175	259	0.402	0.560
75.2	596.0	581.3	19.86	0.158	180	257	0.412	0.555
78.8	621.8	607.1	19.16	0.150	185	254	0.421	0.549
82.4	648.4	633.7	18.36	0.133	190	250	0.430	0.541
86.0	675.9	661.1	17.20	0.118	195	244	0.439	0.529
89.6	704.1	689.4	14.40	0.089	210	233	0.467	0.509
− 90.1†	707.8	693.1	12.63	0.079	220	220	0.485	0.485

* Inches of mercury below one standard atmosphere. † Critical.
The Linde Air Products Company, compilers.

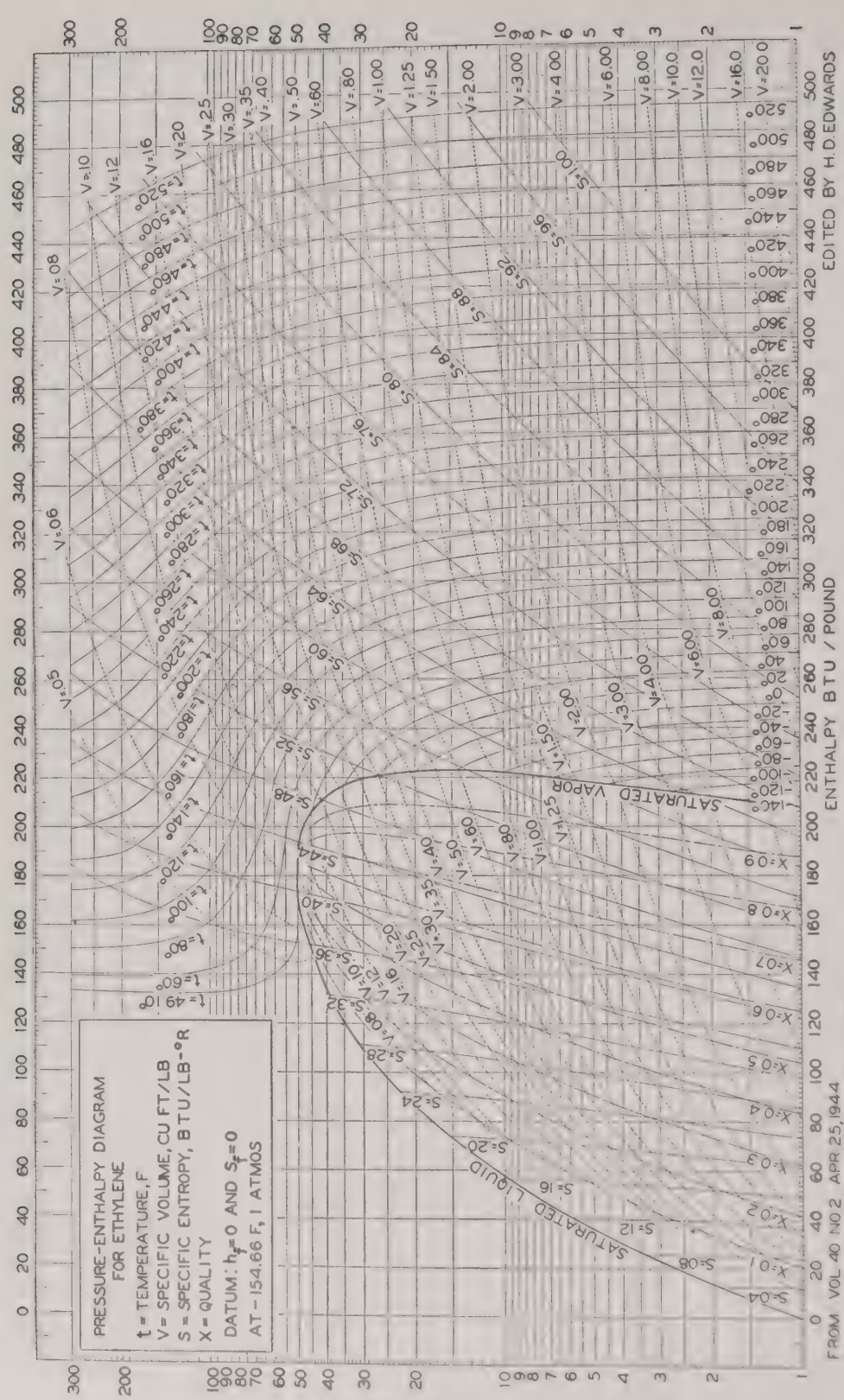

PRESSURE-ENTHALPY DIAGRAM
FOR ETHYLENE
t = TEMPERATURE, F
V = SPECIFIC VOLUME, CU FT/LB
S = SPECIFIC ENTROPY, BTU/LB-°R
X = QUALITY
DATUM: h_f=0 AND S_f=0
AT -154.66 F, 1 ATMOS
PRESSURE ATMOS
ENTHALPY BTU / POUND
SATURATED VAPOR
SATURATED LIQUID
EDITED BY H.D.EDWARDS
FROM VOL 40 NO 2 APR 25, 1944
TRANSACTIONS OF AMERICAN INSTITUTE
OF CHEMICAL ENGINEERS

Chart 6

Table 11. Ethylene—Properties of Liquid and Saturated Vapor

Temp F t	Pressure psia	Pressure psig	Liquid, density lb/cu ft $1/v_f$	Vapor, sp vol cu ft/lb v_g	Enthalpy, datum −176 F Btu per lb: Liquid h_f	Enthalpy, datum −176 F Btu per lb: Vapor h_g	Entropy, datum −176 F Btu per lb F: Liquid s_f	Entropy, datum −176 F Btu per lb F: Vapor s_g
−176.8	6.75	16.18*	36.60	16.4	0	213.9	0.000	0.756
−173.2	7.72	14.21*	36.42	14.5	1.8	214.5	0.006	0.749
−169.6	8.80	12.01*	36.23	12.7	3.5	215.1	0.012	0.742
−166.0	10.00	9.57*	36.05	11.1	5.3	215.8	0.018	0.735
−162.4	11.35	7.02*	35.87	9.71	7.1	216.3	0.024	0.728
−158.8	12.80	3.87*	35.69	8.62	9.0	216.9	0.030	0.722
−155.2	14.44	0.52*	35.51	7.52	10.8	217.5	0.036	0.716
−151.6	16.22	1.52	35.32	6.76	12.6	218.1	0.042	0.710
−148.0	18.17	3.47	35.14	6.06	14.5	218.7	0.048	0.704
−144.4	20.36	5.66	34.96	5.49	16.4	219.2	0.054	0.698
−140.8	22.65	7.95	34.78	4.98	18.3	219.9	0.060	0.693
−137.2	25.17	10.47	34.60	4.50	20.2	220.6	0.066	0.688
−133.6	27.91	13.21	34.42	4.08	22.2	221.2	0.072	0.683
−130.0	30.86	16.16	34.23	3.72	24.1	221.8	0.078	0.678
−126.4	34.06	19.36	34.05	3.39	26.0	222.3	0.084	0.673
−122.8	37.50	22.80	33.87	3.11	28.1	223.1	0.090	0.669
−119.2	41.19	26.49	33.69	2.85	30.1	223.8	0.096	0.665
−115.6	45.16	30.46	33.50	2.62	32.2	224.6	0.102	0.661
−112.0	49.40	34.70	33.30	2.40	34.2	225.3	0.108	0.658
−108.4	53.93	39.23	33.11	2.21	36.3	226.0	0.114	0.654
−104.8	58.76	44.06	32.91	2.04	38.4	226.7	0.120	0.650
−101.2	63.90	49.20	32.72	1.89	40.5	227.3	0.125	0.647
− 97.6	69.37	54.67	32.50	1.75	42.6	228.0	0.131	0.643
− 94.0	75.17	60.47	32.28	1.62	44.8	228.7	0.137	0.640
− 90.4	81.32	66.62	32.07	1.51	47.0	229.4	0.143	0.637
− 86.8	87.88	73.18	31.86	1.40	49.2	230.1	0.149	0.634
− 83.2	94.76	80.06	31.64	1.31	51.4	230.8	0.155	0.631
− 79.6	102.0	87.3	31.42	1.22	53.7	231.5	0.161	0.628
− 76.0	109.7	95.0	31.20	1.14	55.9	232.1	0.166	0.626
− 72.4	117.7	103.0	30.98	1.07	58.3	232.9	0.172	0.623
− 68.8	126.2	111.5	30.76	1.00	60.6	233.5	0.178	0.621
− 65.2	135.1	120.4	30.54	0.935	63.0	234.3	0.184	0.618
− 61.6	144.4	129.7	30.31	0.885	65.3	234.9	0.190	0.616
− 58.0	154.2	139.5	30.07	0.826	67.8	235.5	0.196	0.613
− 54.4	164.5	149.8	29.83	0.775	70.3	235.8	0.202	0.610
− 50.8	175.2	160.5	29.58	0.730	72.7	236.1	0.208	0.608
− 47.2	186.7	172.0	29.34	0.685	75.3	236.4	0.214	0.605
− 43.6	198.1	183.4	29.10	0.645	77.8	236.7	0.220	0.602
− 40.0	210.4	196.7	28.84	0.606	80.4	237.0	0.226	0.599
− 36.4	223.2	208.5	28.58	0.571	83.0	237.3	0.232	0.596
− 32.8	236.6	221.9	28.32	0.535	85.7	237.5	0.238	0.594
− 29.2	250.5	235.8	28.04	0.503	88.4	237.4	0.244	0.590
− 25.8	265.2	250.5	27.76	0.474	91.1	237.4	0.250	0.587
− 22.0	280.1	265.4	27.47	0.444	93.9	237.4	0.256	0.584
− 18.4	296.0	281.3	27.18	0.418	96.8	237.2	0.263	0.581
− 14.8	312.2	297.5	26.87	0.392	99.6	236.8	0.269	0.577
− 11.2	329.3	314.6	26.56	0.368	102.7	236.4	0.275	0.573
− 7.6	346.8	332.1	26.24	0.345	105.7	235.9	0.282	0.570
− 4.0	365.5	350.8	25.92	0.323	108.8	235.3	0.288	0.566
− 0.4	384.3	369.6	25.57	0.302	112.0	234.8	0.295	0.562
+ 3.2	404.4	389.7	25.22	0.283	115.2	234.2	0.301	0.558
6.8	424.7	410.0	24.84	0.266	118.7	233.6	0.308	0.555
10.4	446.5	431.8	24.45	0.249	122.3	233.0	0.316	0.551
14.0	468.7	454.0	24.03	0.234	126.0	232.4	0.323	0.548
17.6	491.8	477.1	23.60	0.218	130	232	0.331	0.545
21.2	515.0	500.3	23.13	0.205	134	230	0.339	0.541
24.8	540.4	525.7	22.63	0.190	138	229	0.348	0.537
28.4	556.7	551.0	22.09	0.176	142	228	0.357	0.533
32.0	593.3	578.6	21.48	0.163	147	226	0.367	0.529
35.6	619.8	606.1	20.83	0.149	152	224	0.377	0.523
39.2	649.5	635.1	20.04	0.135	158	222	0.389	0.520
42.8	678.0	664.3	19.12	0.121	166	218	0.404	0.508
46.4	710.3	695.6	17.92	0.103	177	211	0.420	0.494
48.85†	731.8	717.1	13.48	0.074	194	194	0.460	0.460

* Inches of mercury below one standard atmosphere. † Critical.

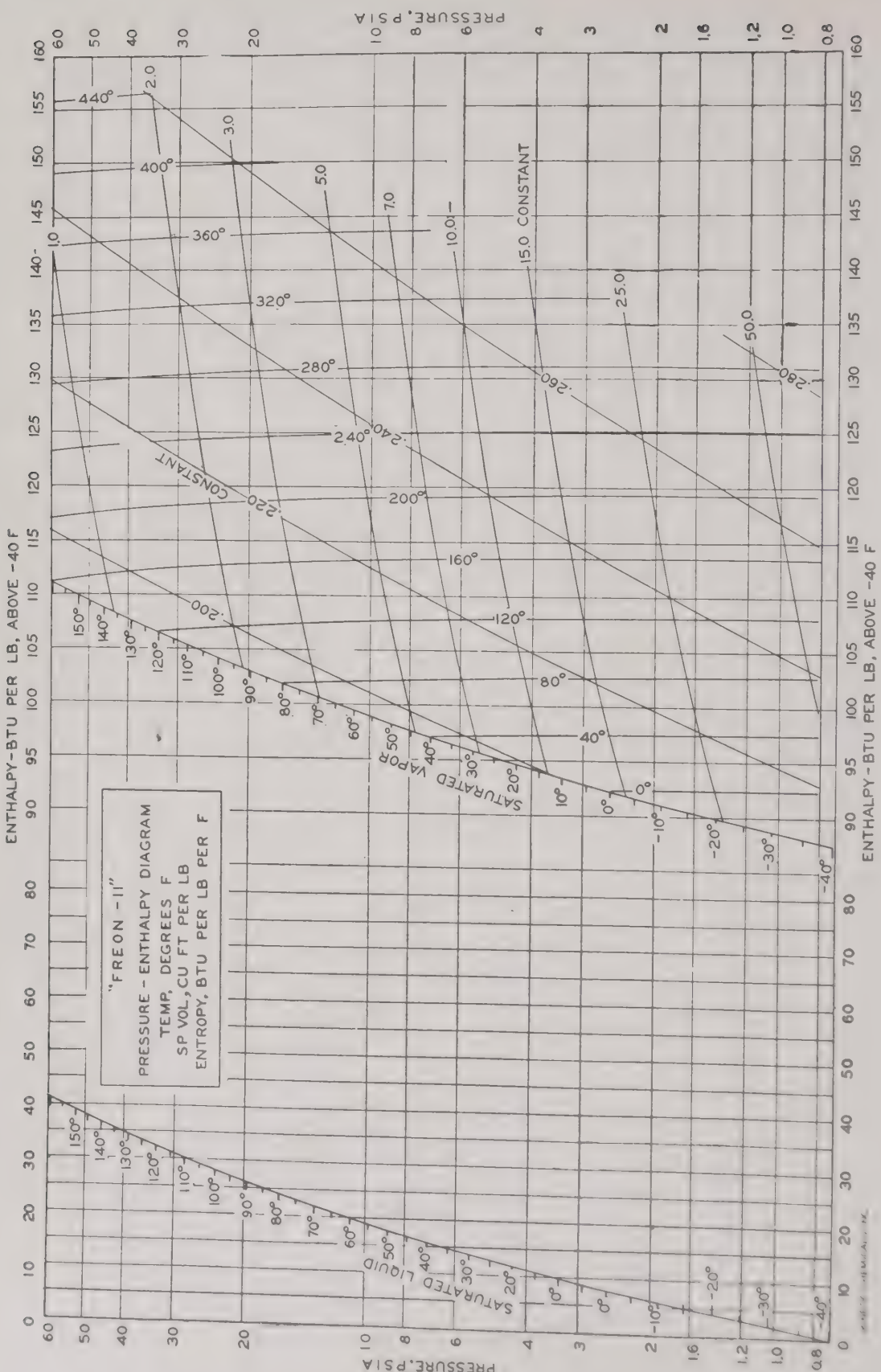

"FREON - 11"
PRESSURE - ENTHALPY DIAGRAM
TEMP, DEGREES F
SP VOL, CU FT PER LB
ENTROPY, BTU PER LB PER F
ENTHALPY - BTU PER LB, ABOVE -40 F
PRESSURE, P S I A
SATURATED LIQUID
SATURATED VAPOR
15.0 CONSTANT
.220 CONSTANT

Chart 7

Table 12. F-11 (CCl_3F) Trichloromonofluoromethane—Properties of Liquid and Saturated Vapor

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−90	0.100	29.72*	0.00964	289.0	−9.7	81.7	−0.0247	0.2226
−80	.157	29.60*	.00961	189.0	−7.7	82.9	− .0195	.2192
−70	.240	29.43*	.00967	127.0	−5.8	84.0	− .0144	.2161
−60	.358	29.19*	.00974	87.0	−3.8	85.2	− .0095	.2132
−50	.520	28.86*	.00981	61.3	−1.9	86.3	− .0047	.2106
−40	0.739	28.42*	0.00988	44.21	0.00	87.48	0.0000	0.2085
−36	0.847	28.20*	.00991	38.93	0.79	87.96	.0019	.2076
−32	0.968	27.95*	.00993	34.37	1.58	88.44	.0037	.2068
−28	1.103	27.67*	.00996	30.44	2.36	88.91	.0055	.2060
−24	1.253	27.37*	.00999	27.03	3.15	89.39	.0073	.2053
−20	1.420	27.03*	0.01002	24.06	3.94	89.87	0.0091	0.2046
−16	1.605	26.65*	.01005	21.47	4.73	90.35	.0109	.2040
−12	1.810	26.24*	.01008	19.20	5.52	90.83	.0127	.2033
− 8	2.035	25.78*	.01011	17.21	6.31	91.31	.0145	.2027
− 4	2.283	25.27*	.01015	15.47	7.10	91.79	.0162	.2021
0	2.555	24.72*	0.01018	13.94	7.89	92.27	0.0179	0.2015
4	2.852	24.11*	.01021	12.58	8.68	92.75	.0197	.2010
5†	2.931	23.95*	.01022	12.27	8.88	92.88	.0201	.2009
8	3.179	23.45*	.01024	11.38	9.48	93.24	.0213	.2005
12	3.534	22.73*	.01027	10.31	10.28	93.72	.0231	.2000
16	3.923	21.94*	.01031	9.359	11.07	94.21	.0248	.1996
20	4.342	21.08*	0.01034	8.519	11.87	94.69	0.0264	0.1991
24	4.801	20.15*	.01037	7.760	12.68	95.18	.0281	.1987
28	5.294	19.14*	.01041	7.087	13.48	95.66	.0297	.1983
32	5.830	18.05*	.01044	6.481	14.28	96.15	.0314	.1979
36	6.411	16.87*	.01048	5.934	15.08	96.63	.0330	.1976
40	7.032	15.61*	0.01051	5.447	15.89	97.11	0.0346	0.1972
44	7.702	14.24*	.01055	5.006	16.70	97.60	.0362	.1969
48	8.422	12.78*	.01058	4.607	17.52	98.08	.0378	.1966
52	9.199	11.20*	.01062	4.245	18.33	98.56	.0394	.1963
56	10.02	9.53*	.01066	3.921	19.15	99.05	.0410	.1960
60	10.90	7.73*	0.01069	3.636	19.96	99.53	0.0426	0.1958
64	11.85	5.80*	.01073	3.356	20.78	100.01	.0442	.1955
68	12.87	3.72*	.01077	3.107	21.61	100.49	.0457	.1953
72	13.95	1.53*	.01081	2.883	22.43	100.97	.0473	.1950
76	15.09	0.39	.01085	2.679	23.26	101.45	.0489	.1948
80	16.31	1.61	0.01088	2.492	24.09	101.93	0.0504	0.1947
84	17.60	2.90	.01092	2.322	24.93	102.41	.0519	.1945
86†	18.28	3.58	.01094	2.242	25.34	102.65	.0527	.1944
88	18.97	4.27	.01096	2.165	25.76	102.89	.0535	.1943
92	20.43	5.73	.01101	2.020	26.60	103.36	.0550	.1941
96	21.97	7.27	.01105	1.887	27.43	103.83	.0565	.1940
100	23.60	8.90	0.01109	1.765	28.27	104.30	0.0580	0.1938
104	25.33	10.63	.01113	1.652	29.12	104.77	.0595	.1937
108	27.15	12.45	.01117	1.548	29.97	105.24	.0610	.1936
112	29.05	14.35	.01122	1.452	30.82	105.71	.0625	.1935
116	31.07	16.37	.01126	1.363	31.67	106.17	.0639	.1934
120	33.20	18.50	0.01130	1.281	32.53	106.63	0.0654	0.1933
124	35.42	20.72	.01135	1.206	33.38	107.09	.0669	.1932
128	37.74	23.04	.01139	1.135	34.24	107.55	.0683	.1931
132	40.23	25.53	.01144	1.068	35.10	108.00	.0698	.1930
136	42.80	28.10	.01149	1.007	35.97	108.46	.0712	.1929
140	45.50	30.80	0.01154	0.9505	36.84	108.91	0.0727	0.1929
144	48.35	33.65	.01159	.8970	37.71	109.35	.0741	.1928
148	51.31	36.61	.01163	.8476	38.59	109.80	.0755	.1927
152	54.41	39.71	.01168	.8014	39.46	110.24	.0770	.1927
156	57.65	42.95	.01173	.7581	40.35	110.69	.0784	.1927
160	61.04	46.34	0.01179	0.7176	41.23	111.12	0.0798	0.1926

* Inches of mercury below one standard atmosphere.
† Standard cycle temperatures.

Table 13. F-11 (CCl_3F) Trichloromonofluoromethane—Properties of Superheated Vapor

Temp F	Pressure 0.2 psia Gage pressure 29.51 in. vac (Sat temp −74.4 F)			Pressure 0.4 psia Gage pressure 29.11 in. vac (Sat temp −57.2 F)			Pressure 0.6 psia Gage pressure 28.70 in. vac (Sat temp −46.1 F)			Pressure 0.8 psia Gage pressure 28.29 in. (Sat temp −37.7 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*150*)	(*83.5*)	(*0.2174*)	(*87.6*)	(*85.5*)	(*0.2125*)	(*53.8*)	(*86.8*)	(*0.2097*)	(*41.09*)	(*87.76*)	(*0.20*
0	180	92.5	0.2387	89.8	92.4	0.2285	59.8	92.4	0.2225	44.77	92.37	0.21
10	183	93.7	.2414	91.7	93.6	.2312	61.1	93.6	.2252	45.74	93.62	.22
20	187	94.9	.2440	93.7	93.8	.2339	62.4	94.8	.2278	46.73	94.88	.22
30	191	96.2	.2466	95.6	96.1	.2365	63.7	96.1	.2304	47.70	96.15	.22
40	195	97.5	.2492	97.6	97.4	.2391	65.0	97.4	.2330	48.69	97.43	.22
50	199	98.8	0.2518	99.5	98.7	0.2417	66.3	98.7	0.2356	49.66	98.72	0.23
60	203	100.1	.2543	102	100.0	.2442	67.7	100.0	.2382	50.65	100.03	.23
70	207	101.4	.2568	103	101.3	.2467	69.0	101.3	.2407	51.62	101.34	.23
80	211	102.8	.2593	105	102.7	.2492	70.3	102.7	.2432	52.60	102.67	.23
90	215	104.1	.2617	107	104.0	.2517	71.6	104.0	.2457	53.38	104.00	.24
100	219	105.5	0.2641	109	105.4	0.2541	72.9	105.4	0.2481	54.56	105.34	.24
110	222	106.8	.2665	111	106.7	.2565	74.2	106.7	.2505	55.54	106.69	.24
120	226	108.1	.2688	113	108.0	.2589	75.5	108.0	.2529	56.62	108.06	.24
130	230	109.5	.2711	115	109.4	.2612	76.8	109.4	.2552	57.50	109.43	.25
140	234	110.9	.2734	117	110.8	.2635	78.1	110.8	.2575	58.58	110.81	.25
150	238	112.3	0.2757	119	112.2	0.2658	79.4	112.2	0.2598	59.46	112.21	0.25
160	242	113.7	.2780	121	113.6	.2681	80.7	113.6	.2621	60.44	113.61	.25
170	246	115.1	.2803	123	115.0	.2704	82.0	115.0	.2644	61.42	115.03	.26
180	250	116.6	.2825	125	116.5	.2726	83.3	116.5	.2667	62.40	116.46	.26
190	254	118.0	.2847	127	117.9	.2748	84.6	117.9	.2689	63.37	117.89	.26

Temp F	Pressure 2.0 psia Gage pressure 25.85 in. vac (Sat temp −8.6 F)			Pressure 4.0 psia Gage pressure 21.78 in. vac (Sat temp 16.7 F)			Pressure 6.0 psia Gage pressure 17.71 in. vac (Sat temp 33.2 F)			Pressure 8.0 psia Gage pressure 13.64 in. v (Sat temp 45.7 F)		
(*at sat*)	(*17.50*)	(*91.24*)	(*0.2028*)	(*9.194*)	(*94.30*)	(*0.1995*)	(*6.313*)	(*96.29*)	(*0.1978*)	(*4.833*)	(*97.80*)	(*0.196*
50	19.81	98.66	0.2183	9.853	98.56	0.2081	6.534	98.46	0.2021	4.875	98.36	0.19
60	20.20	99.97	.2208	10.05	99.87	.2107	6.668	99.77	.2047	4.976	99.67	.20
70	20.59	101.28	.2233	10.25	101.18	.2132	6.802	101.09	.2072	5.077	100.99	.20
80	20.99	102.61	.2258	10.45	102.51	.2157	6.936	102.42	.2097	5.178	102.32	.20
90	21.38	103.94	.2282	10.65	103.45	.2181	7.069	103.76	.2121	5.279	103.66	.20
100	21.77	105.28	0.2307	10.84	105.20	0.2205	7.202	105.11	0.2146	5.380	105.02	0.21
110	22.17	106.64	.2331	11.04	106.56	.2229	7.335	106.47	.2170	5.480	106.38	.21
120	22.56	108.01	.2354	11.24	107.93	.2253	7.467	107.84	.2194	5.580	107.75	.21
130	22.95	109.38	.2378	11.44	109.30	.2277	7.600	109.22	.2217	5.680	109.13	.21
140	23.35	110.76	. 2401	11.64	110.68	.2300	7.733	110.61	.2241	5.780	110.52	.21
150	23.74	112.17	0.2424	11.83	112.08	0.2323	7.865	112.01	0.2264	5.880	111.92	0.22
160	24.13	113.57	.2447	12.03	113.49	.2346	7.998	113.41	.2287	5.980	113.34	.22
170	24.52	114.98	.2470	12.23	114.91	.2369	8.130	114.83	.2309	6.081	114.76	.22
180	24.92	116.41	.2492	12.43	116.34	.2391	8.263	116.26	.2332	6.181	116.19	.22
190	25.31	117.85	.2515	12.62	117.77	.2414	8.395	117.70	.2354	6.281	117.63	.23
200	25.70	119.30	0.2537	12.82	119.22	0.2436	8.527	119.15	0.2376	6.380	119.08	0.23
210	26.10	120.76	.2559	13.02	120.68	.2458	8.659	120.61	.2398	6.480	120.55	.23
220	26.49	122.22	.2581	13.22	122.15	.2480	8.791	122.08	.2420	6.579	122.02	.23
230	26.88	123.70	.2602	13.41	123.63	.2501	8.923	123.56	.2442	6.679	123.50	.24
240	27.27	125.19	.2624	13.61	125.13	.2523	9.056	125.06	.2463	6.779	124.09	.24

Temp F	Pressure 10 psia Gage pressure 9.57 in. vac (Sat temp 55.9 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp 90.8 F)			Pressure 30 psia Gage pressure 15.3 psig (Sat temp 113.9 F)			Pressure 40 psia Gage pressure 25.3 psi (Sat temp 131.6 F)		
(*at sat*)	(*3.928*)	(*99.04*)	(*0.1960*)	(*2.061*)	(*103.22*)	(*0.1942*)	(*1.409*)	(*105.98*)	(*0.1985*)	(*1.074*)	(*107.96*)	(*0.198*
100	4.285	104.93	0.2070	2.099	104.47	0.1964						
110	4.366	106.29	.2094	2.140	105.84	.1989						
120	4.447	107.67	.2118	2.182	107.23	.2013	1.426	106.79	0.1949			
130	4.528	109.05	.2141	2.224	108.63	.2036	1.455	108.19	.1973			
140	4.609	110.44	.2165	2.266	110.03	.2060	1.484	109.60	.1997	1.092	109.17	0.195
150	4.690	111.85	0.2188	2.307	111.44	0.2083	1.513	111.02	0.2020	1.115	110.60	0.197
160	4.771	113.26	.2211	2.349	112.87	.2107	1.541	112.45	.2044	1.137	112.04	.199
170	4.851	114.69	.2234	2.390	114.31	.2130	1.569	113.90	.2067	1.159	113.49	.202
180	4.932	116.12	.2257	2.431	115.75	.2152	1.598	115.35	.2090	1.181	114.95	.204
190	5.012	117.56	.2279	2.473	117.19	.2175	1.626	116.81	.2112	1.202	116.42	.206
200	5.092	119.01	0.2301	2.514	118.65	0.2197	1.654	118.27	0.2135	1.224	117.89	0.208
210	5.171	120.48	.2323	2.555	120.12	.2219	1.682	119.75	.2157	1.246	119.38	.211
220	5.251	121.95	.2345	2.596	121.60	.2241	1.711	121.24	.2179	1.268	120.88	.213
230	5.331	123.43	.2367	2.637	123.09	.2263	1.739	122.74	.2201	1.289	122.39	.215
240	5.411	124.92	.2388	2.678	124.59	.2285	1.767	125.25	.2223	1.311	124.91	.217
250	5.491	126.43	0.2409	2.719	126.11	0.2306	1.794	125.77	0.2244	1.332	125.44	0.219
260	5.571	127.95	.2431	2.760	127.63	.2328	1.822	127.30	.2266	1.353	126.97	.221
270	5.650	129.48	.2452	2.800	129.16	.2349	1.850	128.84	.2287	1.374	128.52	.224
280	5.730	131.01	.2473	2.841	130.70	.2370	1.878	130.39	.2308	1.396	130.08	.226
290	5.809	132.56	.2494	2.881	132.25	.2391	1.905	131.95	.2329	1.417	131.64	.228

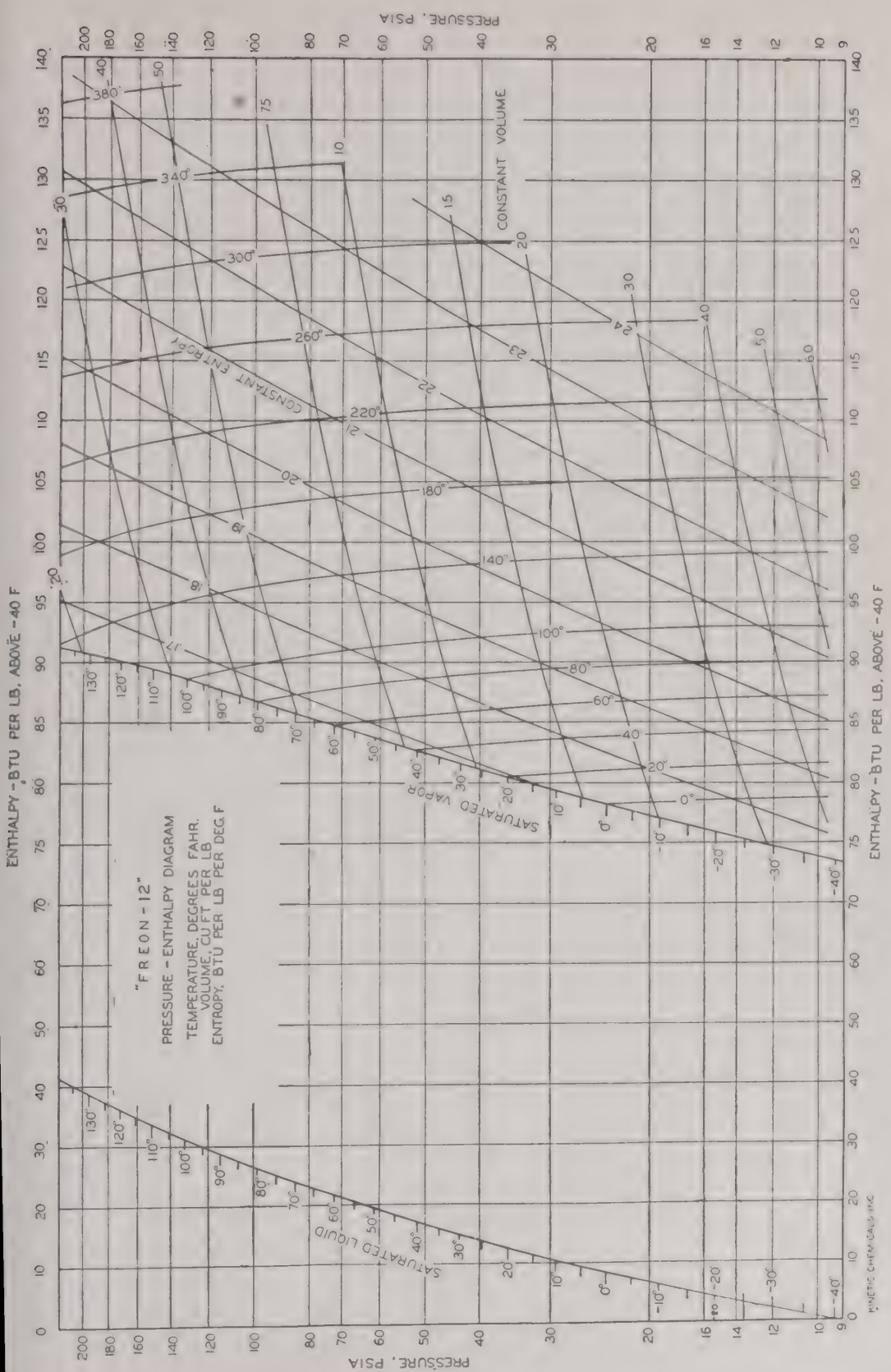

"FREON-12"
PRESSURE - ENTHALPY DIAGRAM
TEMPERATURE, DEGREES FAHR.
VOLUME, CU FT PER LB
ENTROPY, BTU PER LB PER DEG F
CONSTANT VOLUME
CONSTANT ENTROPY
SATURATED LIQUID
SATURATED VAPOR
PRESSURE, PSIA
ENTHALPY - BTU PER LB, ABOVE -40 F
KINETIC CHEMICALS, INC.

Chart 8

Table 14. F-12 (CCl_2F_2) Dichlorodifluoromethane—Properties of Liquid and Saturated Vapor

Temp F t	Pressure psia	Pressure psig	Liquid density lb/cu ft $1/v_f$	Vapor, sp vol cu ft/lb v_g	Enthalpy, datum −40 F Btu per lb Liquid h_f	Enthalpy, datum −40 F Btu per lb Vapor h_g	Entropy, datum −40 F Btu per lb F Liquid s_f	Entropy, datum −40 F Btu per lb F Vapor s_g
−155	0.1163	29.68*	0.00954	232.29	−24.61	60.00	−0.0686	0.2092
−150	0.1527	29.61*	0.00957	179.79	−23.50	60.57	−0.0650	0.2065
−145	.1985	29.52*	.00961	140.52	−22.39	61.14	− .0615	.2040
−140	.2554	29.40*	.00965	110.92	−21.29	61.72	− .0580	.2017
−135	.3256	29.26*	.00969	88.34	−20.19	62.30	− .0546	.1995
−130	.4116	29.08*	.00973	70.94	−19.10	62.88	− .0512	.1975
−125	0.5160	28.87*	0.00977	57.42	−18.02	63.46	−0.0480	0.1955
−120	.6417	28.61*	.00981	46.84	−16.94	64.04	− .0448	.1937
−115	.7921	28.31*	.00985	38.49	−15.85	64.63	− .0416	.1919
−110	.9709	27.94*	.00989	31.84	−14.78	65.22	− .0385	.1903
−105	1.182	27.51*	.00994	26.51	−13.71	65.81	− .0355	.1888
−100	1.430	27.01*	0.00998	22.20	−12.64	66.40	−0.0325	0.1873
− 95	1.719	26.42*	.01003	18.71	−11.58	66.99	− .0295	.1860
− 90	2.054	25.74*	.01007	15.86	−10.51	67.59	− .0266	.1847
− 85	2.441	24.95*	.01012	13.51	− 9.46	68.18	− .0238	.1835
− 80	2.885	24.05*	.01016	11.57	− 8.40	68.77	− .0210	.1823
− 75	3.393	23.01*	0.01021	9.958	− 7.35	69.36	−0.0182	0.1813
− 70	3.971	21.84*	.01026	8.608	− 6.30	69.95	− .0155	.1802
− 65	4.626	20.50*	.01031	7.474	− 5.25	70.54	− .0128	.1793
− 60	5.365	19.00*	.01036	6.516	− 4.20	71.13	− .0102	.1783
− 55	6.195	17.31*	.01041	5.704	− 3.15	71.72	− .0076	.1774
− 50	7.125	15.42*	0.01047	5.012	− 2.11	72.31	−0.0050	0.1767
− 45	8.163	13.31*	.01052	4.420	− 1.06	72.91	− .0025	.1759
− 40	9.317	10.96*	.01057	3.911	0.00	73.50	.0000	.1752
− 35	10.60	8.34*	.01063	3.471	1.01	74.10	+ .0024	.1745
− 30	12.02	5.45*	0.0107	3.088	2.03	74.70	0.00471	0.17387
− 28	12.62	4.23*	.0107	2.950	2.44	74.94	.00565	.17364
− 26	13.26	2.93*	.0107	2.820	2.85	75.18	.00659	.17340
− 24	13.90	1.63*	.0108	2.698	3.25	75.41	.00753	.17317
− 22	14.58	0.24*	.0108	2.583	3.66	75.64	.00846	.17296
− 20	15.28	0.58	0.0108	2.474	4.07	75.87	0.00940	0.17275
− 18	16.01	1.31	.0108	2.370	4.48	76.11	.01033	.17253
− 16	16.77	2.07	.0108	2.271	4.89	76.34	.01126	.17232
− 14	17.55	2.85	.0109	2.177	5.30	76.57	.01218	.17212
− 12	18.37	3.67	.0109	2.088	5.72	76.81	.01310	.17194
− 10	19.20	4.50	0.0109	2.003	6.14	77.05	0.01403	0.17175
− 8	20.08	5.38	.0109	1.922	6.57	77.29	.01496	.17158
− 6	20.98	6.28	.0110	1.845	6.99	77.52	.01589	.17140
− 4	21.91	7.21	.0110	1.772	7.41	77.75	.01682	.17123
− 2	22.87	8.17	.0110	1.703	7.83	77.98	.01775	.17107
0	23.87	9.17	0.0110	1.637	8.25	78.21	0.01869	0.17091
2	24.89	10.19	.0110	1.574	8.67	78.44	.01961	.17075
4	25.96	11.26	.0111	1.514	9.10	78.67	.02052	.17060
5†	26.51	11.81	.0111	1.485	9.32	78.79	.02097	.17052
6	27.05	12.35	.0111	1.457	9.53	78.90	.02143	.17045
8	28.18	13.48	.0111	1.403	9.96	79.13	.02235	.17030
10	29.35	14.65	0.0112	1.351	10.39	79.36	0.02328	0.17015
12	30.56	15.86	.0112	1.301	10.82	79.59	.02419	.17001
14	31.80	17.10	.0112	1.253	11.26	79.82	.02510	.16987
16	33.08	18.38	.0112	1.207	11.70	80.05	.02601	.16974
18	34.40	19.70	.0113	1.163	12.12	80.27	.02692	.16961
20	35.75	21.05	0.0113	1.121	12.55	80.49	0.02783	0.16949
22	37.15	22.45	.0113	1.081	13.00	80.72	.02873	.16938
24	38.58	23.88	.0113	1.043	13.44	80.95	.02963	.16926
26	40.07	25.37	.0114	1.007	13.88	81.17	.03053	.16913
28	41.59	26.89	.0114	0.973	14.32	81.39	.03143	.16900

* Inches of mercury below one atmosphere.
† Standard cycle temperature.

Table 14. F-12 (CCl_2F_2) Dichlorodifluoromethane—Properties of Liquid and Saturated Vapor (Concluded)

Temp, F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
30	43.16	28.46	0.0115	0.939	14.76	81.61	0.03233	0.16887
32	44.77	30.07	.0115	.908	15.21	81.83	.03323	.16876
34	46.42	31.72	.0115	.877	15.65	82.05	.03413	.16865
36	48.13	33.43	.0116	.848	16.10	82.27	.03502	.16854
38	49.88	35.18	.0116	.819	16.55	82.49	.03591	.16843
40	51.68	36.98	0.0116	0.792	17.00	82.71	0.03680	0.16833
42	53.51	38.81	.0116	.767	17.46	82.93	.03770	.16823
44	55.40	40.70	.0117	.742	17.91	83.15	.03859	.16813
46	57.35	42.65	.0117	.718	18.36	83.36	.03948	.16803
48	59.35	44.65	.0117	.695	18.82	83.57	.04037	.16794
50	61.39	46.69	0.0118	0.673	19.27	83.78	0.04126	0.16785
52	63.49	48.79	.0118	.652	19.72	83.99	.04215	.16776
54	65.63	50.93	.0118	.632	20.18	84.20	.04304	.16767
56	67.84	53.14	.0119	.612	20.64	84.41	.04392	.16758
58	70.10	55.40	.0119	.593	21.11	84.62	.04480	.16749
60	72.41	57.71	0.0119	0.575	21.57	84.82	0.04568	0.16741
62	74.77	60.07	.0120	.557	22.03	85.02	.04657	.16733
64	77.20	62.50	.0120	.540	22.49	85.22	.04745	.16725
66	79.67	64.97	.0120	.524	22.95	85.42	.04833	.16717
68	82.24	67.54	.0121	.508	23.42	85.62	.04921	.16709
70	84.82	70.12	0.0121	0.493	23.90	85.82	0.05009	0.16701
72	87.50	72.80	.0121	.479	24.37	86.02	.05097	.16693
74	90.20	75.50	.0122	.464	24.84	86.22	.05185	.16685
76	93.00	78.30	.0122	.451	25.32	86.42	.05272	.16677
78	95.85	81.15	.0123	.438	25.80	86.61	.05359	.16669
80	98.76	84.06	0.0123	0.425	26.28	86.80	0.05446	0.16662
82	101.7	87.00	.0123	.413	26.76	86.90	.05534	.16655
84	104.8	90.1	.0124	.401	27.24	87.18	.05621	.16648
86†	107.9	93.2	.0124	.389	27.72	87.37	.05708	.16640
88	111.1	96.4	.0124	.378	28.21	87.56	.05795	.16632
90	114.3	99.6	0.0125	0.368	28.70	87.74	0.05882	0.16624
92	117.7	103.0	.0125	.357	29.19	87.92	.05969	.16616
94	121.0	106.3	.0126	.347	29.68	88.10	.06056	.16608
96	124.5	109.8	.0126	.338	30.18	88.28	.06143	.16600
98	128.0	113.3	.0126	.328	30.67	88.45	.06230	.16592
100	131.6	116.9	0.0127	0.319	31.16	88.62	0.06316	0.16584
102	135.3	120.6	.0127	.310	31.65	88.79	.06403	.16576
104	139.0	124.3	.0128	.302	32.15	88.95	.06490	.16568
106	142.8	128.1	.0128	.293	32.65	89.11	.06577	.16560
108	146.8	132.1	.0129	.285	33.15	89.27	.06663	.16551
110	150.7	136.0	0.0129	0.277	33.65	89.43	0.06749	0.16542
112	154.8	140.1	.0130	.269	34.15	89.58	.06836	.16533
114	158.9	144.2	.0130	.262	34.65	89.73	.06922	.16524
116	163.1	148.4	.0131	.254	35.15	89.87	.07008	.16515
118	167.4	152.7	.0131	.247	35.65	90.01	.07094	.16505
120	171.8	157.1	0.0132	0.240	36.16	90.15	0.07180	0.16495
122	176.2	161.5	.0132	.233	36.66	90.28	.07266	.16484
124	180.8	166.1	.0133	.277	37.16	90.40	.07352	.16473
126	185.4	170.7	.0133	.220	37.67	90.52	.07437	.16462
128	190.1	175.4	.0134	.214	38.18	90.64	.07522	.16450
130	194.9	180.2	0.0134	0.208	38.69	90.76	0.07607	0.16438
132	199.8	185.1	.0135	.202	39.19	90.86	.07691	.16425
134	204.8	190.1	.0135	.196	39.70	90.96	.07775	.16411
135	209.9	195.2	.0136	.191	40.21	91.06	.07858	.16396
138	215.0	200.3	.0137	.185	40.72	91.51	.07941	.16380
140	220.2	205.5	0.0138	0.180	41.24	91.24	0.08024	0.16363

† Standard cycle temperature.

Table 15a. F-12 (CCl_2F_2) Dichlorodifluoromethane—Properties of Superheated Vapor

Temp F t	Pressure 0.20 psia Gage pressure 29.51 in. vac (Sat temp −144.9 F)			Pressure 0.40 psia Gage pressure 29.11 in. vac (Sat temp −130.7 F)			Pressure 0.60 psia Gage pressure 28.70 in. vac (Sat temp −121.7 F)			Pressure 0.80 psia Gage pressure 28.92 in. (Sat temp −114.8 F)		
	v	h	s	v	h	s	v	h	s	v	h	s
(at sat)	*(139.5)*	*(61.16)*	*(0.2040)*	*(72.84)*	*(62.80)*	*(0.1978)*	*(49.85)*	*(63.85)*	*(0.1943)*	*(38.13)*	*(64.66)*	*(0.19*
−140	141.7	61.72	0.2059									
−130	146.1	62.89	.2094	73.00	62.88	0.1980						
−120	150.6	64.07	.2129	75.22	64.06	.2015	50.11	64.04	0.1948			
−110	155.0	65.27	.2164	77.45	65.25	.2050	51.59	65.24	.1983	38.66	65.23	0.19
−100	159.5	66.47	0.2198	79.67	66.46	0.2084	53.08	66.45	0.2017	39.78	66.44	0.19
− 90	163.9	67.70	.2231	81.90	67.68	.2117	54.56	67.67	.2050	40.89	67.66	.20
− 80	168.3	68.93	.2264	84.12	68.92	.2150	56.05	68.91	.2083	41.02	68.90	.20
− 70	172.8	70.18	.2297	86.34	70.17	.2183	57.53	70.16	.2116	43.12	70.14	.20
− 60	177.2	71.44	.2329	88.57	71.43	.2215	59.01	71.42	.2148	44.24	71.41	.21
− 50	181.7	72.71	0.2360	90.79	72.70	0.2246	60.50	72.69	0.2179	45.35	72.68	0.21
− 40	186.1	74.00	.2391	93.01	73.99	.2277	61.98	73.98	.2210	46.47	73.97	.21
− 30	190.6	75.30	.2422	95.24	75.29	.2308	63.46	75.28	.2241	47.58	75.27	.21
− 20	195.0	76.62	.2452	97.46	76.61	.2338	64.95	76.60	.2271	48.69	76.59	.22
− 10	199.4	77.94	.2482	99.68	77.93	.2368	66.43	77.93	.2301	49.81	77.92	.22
0	203.9	79.29	0.2511	101.9	79.28	0.2397	67.91	79.27	0.2331	50.92	79.26	0.22
10	208.3	80.64	.2541	104.1	80.63	.2427	69.39	80.62	.2360	52.03	80.61	.23
20	212.8	82.01	.2569	106.4	82.00	.2455	70.88	81.99	.2380	53.14	81.98	.23
30	217.2	83.39	.2598	108.6	83.38	.2484	72.36	83.37	.2417	54.26	83.36	.23
40	221.6	84.78	.2626	110.8	84.77	.2512	73.84	84.77	.2445	55.37	84.76	.23
50	226.1	86.19	0.2654	113.0	86.18	0.2540	75.33	86.17	0.2473	56.48	86.17	0.24
60	230.5	87.61	.2682	115.2	87.60	.2568	76.81	87.59	.2501	57.59	87.59	.24
70	235.0	89.04	.2709	117.5	89.04	.2595	78.29	89.03	.2528	58.71	89.02	.24
80	239.4	90.49	.2736	119.7	90.48	.2622	79.77	90.48	.2555	59.82	90.47	.25
90	243.8	91.95	.2763	121.9	91.94	.2649	81.25	91.94	.2582	60.94	91.93	.25
100	248.3	93.42	0.2789	124.1	93.42	0.2675	82.73	93.41	0.2609	62.04	93.40	0.25
110	252.7	94.91	.2816	126.3	94.91	.2702	84.22	94.90	.2635	63.15	94.89	.25
120	257.2	96.41	.2842	128.6	96.41	.2728	85.70	96.40	.2661	64.27	96.39	.26
130	261.6	97.93	.2868	130.8	97.92	.2754	87.18	97.91	.2687	65.38	97.91	.26
140	266.0	99.45	.2893	133.0	99.45	.2779	88.66	99.44	.2713	66.49	99.44	.26
150	270.5	100.99	0.2919	135.2	100.99	0.2805	90.14	100.98	0.2738	67.60	100.98	0.26
160	274.9	102.55	.2944	137.5	102.54	.2830	91.62	102.54	.2764	68.71	102.53	.27
170				139.7	104.11	.2855	93.10	104.10	.2789	69.82	104.10	.27
180							94.59	105.68	.2814	70.93	105.68	.27

Temp F	Pressure 2.0 psia Gage pressure 25.85 in. vac (Sat temp −90.8 F)			Pressure 4.0 psia Gage pressure 21.78 in. vac (Sat temp −69.8 F)			Pressure 6.0 psia Gage pressure 17.71 in. vac (Sat temp −56.2 F)			Pressure 8.0 psia Gage pressure 13.64 in. (Sat temp −45.7 F)		
(at sat)	*(16.26)*	*(66.49)*	*(0.1849)*	*(8.549)*	*(69.79)*	*(0.1802)*	*(5.874)*	*(71.58)*	*(0.1776)*	*(4.502)*	*(72.80)*	*(0.17*
− 90	16.29	67.59	0.1851									
− 80	16.74	68.83	.1884									
− 70	17.19	70.08	.1917									
− 60	17.64	71.35	.1949	8.772	71.21	0.1833						
− 50	18.09	72.62	0.1980	8.999	72.49	0.1865	5.698	72.37	0.1796			
− 40	18.54	73.91	.2011	9.225	73.78	.1896	6.121	73.67	.1827	4.569	73.56	0.17
− 30	18.98	75.22	.2042	9.451	75.09	.1927	6.273	74.98	.1858	4.684	74.87	.18
− 20	19.43	76.53	.2072	9.677	76.41	.1957	6.425	76.30	.1889	4.799	76.20	.18
− 10	19.88	77.86	.2102	9.903	77.75	.1987	6.577	77.64	.1919	4.914	77.54	.18
0	20.33	79.21	0.2132	10.13	79.10	0.2017	6.729	78.98	0.1948	5.028	78.89	0.18
10	20.77	80.56	.2161	10.35	80.46	.2046	6.879	80.34	.1978	5.142	80.26	.19
20	21.22	81.93	.2190	10.58	81.83	.2075	7.031	81.73	.2007	5.257	81.64	.19
30	21.67	83.31	.2219	10.80	83.21	.2104	7.182	83.12	.2035	5.370	83.02	.19
40	22.11	84.71	.2247	11.03	84.61	.2132	7.334	84.52	.2064	5.484	84.43	.20
50	22.56	86.12	0.2275	11.25	86.03	0.2160	7.485	85.93	0.2092	5.598	85.85	0.20
60	23.01	87.54	.2302	11.48	87.45	.2187	7.635	87.37	.2119	5.711	87.27	.20
70	23.45	88.98	.2330	11.70	88.89	.2215	7.786	88.71	.2147	5.824	88.72	.20
80	23.90	90.43	.2357	11.93	90.35	.2242	7.936	90.26	.2174	5.938	90.18	.21
90	24.35	91.89	.2384	12.15	91.81	.2269	8.087	91.73	.2201	6.051	91.64	.21
100	24.79	93.37	0.2410	12.38	93.29	0.2296	8.237	93.21	0.2228	6.165	93.13	0.21
110	25.24	94.85	.2437	12.60	94.78	.2322	8.388	94.70	.2255	6.278	94.63	.22
120	25.68	96.36	.2463	12.82	96.28	.2348	8.538	96.20	.2281	6.391	96.13	.22
130	26.13	97.87	.2489	13.05	97.80	.2374	8.688	97.72	.2307	6.504	97.64	.22
140	26.58	99.40	.2514	13.27	99.33	.2400	8.838	99.25	.2333	6.617	99.18	.22
150	27.02	100.94	0.2540	13.50	100.88	0.2425	8.988	100.80	0.2358	6.730	100.73	0.23
160	24.47	102.50	.2565	13.72	102.43	.2451	9.138	102.36	.2383	6.843	102.29	.23
170	27.91	104.06	.2590	13.94	104.00	.2476	9.288	103.93	.2408	6.955	103.87	.23
180	28.36	105.65	.2615	14.17	105.58	.2501	9.437	105.51	.2433	7.068	105.44	.23
190	28.80	107.24	.2640	14.39	107.18	.2526	9.587	107.11	.2458	7.181	107.05	.24
200	29.25	108.85	0.2665	14.62	108.79	0.2550	9.737	108.72	0.2483	7.294	108.67	0.24
210	29.70	110.47	.2689	14.84	110.41	.2575	9.886	110.34	.2507	7.407	110.28	.24
220				15.06	112.05	.2599	10.04	111.98	.2532	7.520	111.93	.24
230				15.29	113.70	.2623	10.19	113.63	.2556	7.633	113.57	.25
240				15.51	115.36	.2647	10.34	115.29	.2580	7.746	115.23	.25

Table 15b. F-12 (CCl_2F_2) Dichlorodifluoromethane—Properties of Superheated Vapor

Temp F t	Pressure 10 psia Gage pressure 9.6 in. vac (Sat temp −37.3 F) v	h	s	Pressure 20 psia Gage pressure 5.3 in. vac (Sat temp −8.2 F) v	h	s	Pressure 30 psia Gage pressure 15.3 in. vac (Sat temp 11.1 F) v	h	s	Pressure 40 psia Gage pressure 25.3 in. vac (Sat temp 25.9 F) v	h	s
sat)	*(3.652)*	*(73.80)*	*(0.17480)*	*(1.925)*	*(77.27)*	*(0.17160)*	*(1.323)*	*(79.47)*	*(0.17008)*	*(1.009)*	*(81.16)*	*(0.16914)*
-30	3.728	74.77	0.17704									
-20	3.821	76.11	.18008									
-10	3.913	77.46	.18310									
0	4.006	78.81	0.18611	1.965	78.39	0.17407						
10	4.098	80.18	.18905	2.013	79.76	.17704						
20	4.189	81.56	.19194	2.060	81.14	.17996	1.350	80.73	0.17269			
30	4.280	82.94	.19482	2.107	82.55	.18286	1.383	82.15	.17562	1.019	81.76	0.17030
40	4.371	84.35	.19766	2.155	83.97	.18573	1.415	83.58	.17851	1.044	83.20	.17322
50	4.463	85.77	0.20047	2.203	85.40	0.18858	1.448	85.03	0.18138	1.070	84.65	0.17612
60	4.556	87.19	.20326	2.250	86.85	.19138	1.480	86.48	.18420	1.095	86.11	.17896
70	4.648	88.64	.20601	2.297	88.31	.19415	1.512	87.95	.18699	1.120	87.60	.18178
80	4.740	90.11	.20874	2.343	89.78	.19688	1.544	89.43	.18974	1.144	89.09	.18455
90	4.832	91.58	.21144	2.390	91.26	.19959	1.576	90.91	.19249	1.169	90.58	.18731
100	4.923	93.05	0.21411	2.437	92.75	0.20229	1.608	92.41	0.19519	1.194	92.09	0.19004
110	5.015	94.56	.21676	2.483	94.26	.20494	1.640	93.93	.19787	1.218	93.62	.19272
120	5.107	96.07	.21940	2.530	95.78	.20759	1.672	95.46	.20053	1.242	95.15	.19538
130	5.198	97.59	.22199	2.577	97.31	.21020	1.703	97.00	.20315	1.267	96.70	.19803
140	5.289	99.14	.22458	2.623	98.85	.21280	1.735	98.54	.20577	1.291	98.26	.20066
150	5.379	100.66	0.22713	2.699	100.40	0.21537	1.767	100.11	0.20836	1.315	99.83	0.20325
160	5.470	102.24	.22967	2.716	101.97	.21792	1.799	101.69	.21092	1.340	101.42	.20583
170	5.560	103.81	.23218	2.762	103.56	.22045	1.829	103.28	.21344	1.364	103.02	.20838
180	5.650	105.40	.23469	2.808	105.15	.22297	1.860	104.88	.21597	1.388	104.63	.21092
190	5.740	107.00	.21717	2.854	106.76	.22545	1.891	106.49	.21846	1.412	106.25	.21343
200	5.831	108.63	0.23963	2.901	108.38	0.22794	1.923	108.12	0.22096	1.435	107.88	0.21592
210	5.921	110.25	.24208	2.947	110.01	.23039	1.954	109.76	.22342	1.459	109.52	.21840
220	6.011	111.88	.24451	2.992	111.65	.23283	1.986	111.41	.22588	1.482	111.17	.22085
230	6.101	113.53	.24692	3.038	113.31	.23524	2.017	113.08	.22830	1.506	112.84	.22329
240				3.084	114.98	.23766	2.048	114.75	.23072	1.530	114.52	.22572
250				3.130	116.67	0.24005	2.079	116.44	0.23312	1.554	116.21	0.22813
260				3.117	118.36	.24242	2.110	118.15	.23550	1.577	117.92	.23052
270				3.223	120.07	.24477	2.141	119.87	.23787	1.601	119.65	.23289
280							2.172	121.60	.24023	1.625	121.40	.23526
290										1.649	123.15	.23760
300										1.673	124.92	0.23994

Temp F t	Pressure 50 psia Gage pressure 35.3 in. vac (Sat temp 38.3 F) v	h	s	Pressure 60 psia Gage pressure 45.3 in. vac (Sat temp 48.7 F) v	h	s	Pressure 70 psia Gage pressure 55.3 in. vac (Sat temp 57.9 F) v	h	s	Pressure 80 psia Gage pressure 65.3 in. vac (Sat temp 66.3 F) v	h	s
sat)	*(8.817)*	*(82.52)*	*(0.16841)*	*(0.688)*	*(83.65)*	*(0.16791)*	*(0.594)*	*(84.61)*	*(0.16749)*	*(0.521)*	*(85.45)*	*(0.16716)*
40	0.821	82.76	0.16895									
50	0.842	84.24	0.17187	0.690	83.83	0.16829						
60	0.863	85.72	.17475	0.708	85.33	.17120	0.597	84.94	0.16310			
70	0.884	87.22	.17760	0.726	86.84	.17407	0.612	86.44	.17097	0.526	86.01	0.16819
80	0.904	88.72	.18040	0.743	88.35	.17689	0.628	87.96	.17382	0.540	87.56	.17108
90	0.924	90.23	.18317	0.760	89.87	.17968	0.643	89.49	.17665	0.554	89.12	.17394
100	0.944	91.75	0.18591	0.778	91.41	0.18246	0.658	91.03	0.17943	0.568	90.68	0.17675
110	0.964	93.29	.18862	0.795	92.96	.18519	0.673	92.59	.18219	0.582	92.26	.17954
120	0.984	94.83	.19132	0.812	94.51	.18789	0.689	94.16	.18493	0.596	93.84	.18229
130	1.004	96.39	.19397	0.829	96.07	.19056	0.704	95.75	.18763	0.609	95.43	.18500
140	1.024	97.96	.19662	0.846	97.65	.19323	0.719	97.34	.19030	0.623	97.03	.18771
150	1.044	99.54	0.19923	0.863	99.24	0.19585	0.733	98.94	0.19293	0.636	98.64	0.19035
160	1.064	101.14	.20182	0.880	100.84	.19846	0.748	100.54	.19555	0.649	100.26	.19298
170	1.084	102.75	.20439	0.897	102.45	.20104	0.763	102.16	.19814	0.662	101.88	.19558
180	1.103	104.36	.20694	0.913	104.07	.20360	0.777	103.80	.20071	0.675	103.52	.19817
190	1.123	105.98	.20946	0.930	105.71	.20613	0.792	105.45	.20325	0.688	105.18	.20073
200	1.142	107.62	0.21196	0.946	107.36	0.20865	0.806	107.10	0.20579	0.701	106.84	0.20328
210	1.162	109.28	.21444	0.962	109.02	.21113	0.820	108.76	.20829	0.714	108.51	.20580
220	1.181	110.95	.21691	0.979	110.69	.21361	0.835	110.43	.21079	0.726	110.19	.20828
230	1.200	112.62	.21935	0.995	112.37	.21607	0.849	112.13	.21325	0.739	111.88	.21076
240	1.220	114.31	.22179	1.012	114.06	.21853	0.863	113.83	.21570	0.751	113.58	.21321
250	1.239	116.00	0.22419	1.028	115.77	0.22094	0.878	115.55	0.21815	0.764	115.30	0.21566
260	1.258	117.71	.22660	1.044	117.49	.22334	0.892	117.28	.22057	0.777	117.03	.21809
270	1.277	119.44	.22898	1.060	119.23	.22573	0.906	119.02	.22296	0.789	118.78	.22049
280	1.296	121.18	.23134	1.076	120.97	.22810	0.920	120.76	.22534	0.802	120.54	.22289
290	1.314	122.93	.23367	1.092	122.73	.23045	0.934	122.52	.22770	0.814	122.30	.22525
300	1.332	124.69	0.23600	1.108	124.50	0.23280	0.948	124.29	0.23006	0.826	124.08	0.22760
310	1.350	126.45	.23831	1.124	126.28	.23513	0.961	126.07	.23239	0.839	125.88	.22995
320				1.140	128.07	.23745	0.975	127.88	.23471	0.851	127.70	.23229
330							0.989	129.70	.23702	0.864	129.52	.23461

Table 15c. F-12 (CCl_2F_2) Dichlorodifluoromethane—Properties of Superheated Vapor

Temp F	Pressure 90 psia Gage pressure 75.3 psig (Sat temp 73.9 F)			Pressure 100 psia Gage pressure 85.3 psig (Sat temp 80.0 F)			Pressure 110 psia Gage pressure 95.3 psig (Sat temp 87.3 F)			Pressure 120 psia Gage pressure 105.3 ps (Sat temp 93.4 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*0.465*)	(*86.21*)	(*0.16685*)	(*0.419*)	(*86.89*)	(*0.16659*)	(*0.382*)	(*87.50*)	(*0.16635*)	(*0.350*)	(*88.05*)	(*0.166*
80	0.473	87.17	0.16862									
90	0.486	88.74	.17149	0.430	88.32	0.16926	0.385	87.91	0.16711			
100	0.499	90.31	0.17433	0.442	89.93	0.17210	0.396	89.51	0.17001	0.357	89.13	0.168
110	0.511	91.89	.17713	0.454	91.54	.17493	0.407	91.12	.17287	0.367	90.75	.170
120	0.523	93.48	.17990	0.465	93.15	.17733	0.417	92.74	.17568	0.377	92.38	.173
130	0.535	95.08	.18262	0.477	94.76	.18049	0.428	94.37	.17845	0.387	94.01	.176
140	0.547	96.69	.18533	0.488	96.37	.18231	0.438	96.01	.18122	0.397	95.65	.179
150	0.559	98.31	0.18799	0.499	97.99	0.18590	0.449	97.66	0.18394	0.407	97.30	0.182
160	0.571	100.04	.19065	0.510	99.63	.18856	0.459	99.31	.18660	0.417	98.96	.184
170	0.584	101.58	.19327	0.521	101.28	.19120	0.469	100.97	.18924	0.426	100.63	.187
180	0.596	103.23	.19588	0.531	102.84	.19381	0.479	102.64	.19187	0.436	102.31	.190
190	0.607	104.89	.19845	0.542	104.61	.19638	0.489	104.32	.19447	0.445	104.00	.192
200	0.619	106.56	0.20101	0.553	106.20	0.19894	0.499	106.01	0.19706	0.454	105.70	0.195
210	0.630	108.24	.20353	0.563	107.98	.20148	0.509	107.71	.19962	0.463	107.41	.197
220	0.642	109.93	.20603	0.574	109.68	.20401	0.519	109.42	.20216	0.472	109.13	.200
230	0.653	111.63	.20852	0.585	111.39	.20650	0.528	111.14	.20464	0.482	110.86	.202
240	0.665	113.35	.21100	0.595	113.11	.20899	0.538	112.87	.20712	0.491	112.60	.205
250	0.676	115.08	0.21345	0.606	114.84	0.21145	0.548	114.61	0.20959	0.500	114.35	0.207
260	0.688	116.82	.21589	0.616	116.58	.21389	0.557	116.36	.21205	0.508	116.11	.210
270	0.699	118.57	.21831	0.626	118.33	.21631	0.567	118.12	.21448	0.517	117.88	.212
280	0.710	120.33	.22070	0.636	120.10	.21870	0.576	119.89	.21690	0.526	119.66	.215
290	0.721	122.10	.22306	0.646	121.88	.22108	0.586	121.68	.21930	0.534	121.45	.217
300	0.732	123.88	0.22542	0.657	123.67	0.22347	0.595	123.48	0.22167	0.543	123.25	0.220
310	0.743	125.97	.22776	0.667	125.47	.22583	0.605	125.29	.22405	0.552	125.07	.222
320	0.754	127.48	.23008	0.677	127.28	.22817	0.614	127.11	.22639	0.560	126.90	.224
330	0.765	129.31	.23240	0.687	129.10	.23050	0.623	128.94	.22872	0.569	128.74	.227
340				0.697	130.94	.23281	0.632	130.78	.23103	0.578	130.59	.229
350				0.707	132.80	0.23150	0.641	132.63	0.23333	0.586	132.45	0.231
360				0.718	134.68	.23738	6.651	134.50	.23562	0.595	134.32	.234
370												

Temp F	Pressure 140 psia Gage pressure 125.3 psig (Sat temp 104.5 F)			Pressure 160 psia Gage pressure 145.3 psig (Sat temp 114.5 F)			Pressure 180 psia Gage pressure 165.3 psig (Sat temp 123.7 F)			Pressure 200 psia Gage pressure 185.3 ps (Sat temp 132.1 F)		
(*at sat*)	(*0.298*)	(*88.89*)	(*0.16566*)	(*0.259*)	(*89.77*)	(*0.16522*)	(*0.228*)	(*90.38*)	(*0.16476*)	(*0.202*)	(*90.86*)	(*0.164*
110	0.304	89.92	0.16725									
120	0.314	91.60	.17021	0.264	90.68	0.16682						
130	0.323	93.28	.17306	0.273	92.40	.16977	0.233	91.47	0.16665			
140	0.332	94.96	.17590	0.282	94.12	.17269	0.241	93.23	.16964	0.208	92.30	0.166
150	0.341	96.65	0.17868	0.290	95.84	0.17553	0.249	94.99	0.17254	0.216	94.10	0.169
160	0.350	98.34	.18142	0.298	97.57	.17832	0.257	96.75	.17541	0.224	95.90	.172
170	0.358	100.03	.18412	0.306	99.31	.18106	0.265	98.52	.17823	0.231	97.70	.175
180	0.366	101.72	.18678	0.313	101.05	.18377	0.272	100.29	.18102	0.238	99.51	.178
190	0.374	103.42	.18941	0.321	102.80	.18646	0.280	102.07	.18377	0.245	101.32	.181
200	0.383	105.14	0.19205	0.329	104.55	0.18913	0.287	103.85	0.18648	0.252	103.13	0.183
210	0.391	106.86	.19466	0.336	106.31	.19175	0.294	105.63	.18912	0.258	104.94	.186
220	0.399	108.59	.19724	0.344	108.07	.19435	0.301	107.42	.19174	0.265	106.76	.189
230	0.407	110.33	.19976	0.351	109.83	.19693	0.307	109.21	.19433	0.272	108.58	.191
240	0.415	112.09	.20229	0.358	111.60	.19949	0.314	111.01	.19693	0.278	110.40	.194
250	0.423	113.85	0.20479	0.366	113.38	0.20203	0.321	112.81	0.19947	0.284	112.23	0.197
260	0.431	115.63	.20728	0.373	115.17	.20453	0.327	114.62	.20199	0.290	114.06	.199
270	0.439	117.42	.20974	0.380	116.97	.20700	0.334	116.44	.20449	0.296	115.89	.202
280	0.447	119.32	.21219	0.387	118.78	.20946	0.340	118.26	.20698	0.302	117.73	.204
290	0.455	121.03	.21461	0.394	120.60	.21189	0.347	120.09	.20944	0.308	119.58	.207
300	0.462	122.85	0.21701	0.401	122.43	0.21432	0.353	121.92	0.21187	0.314	121.44	0.209
310	0.470	124.67	.21939	0.408	124.27	.21672	0.359	123.76	.21428	0.320	123.31	.212
320	0.477	126.50	.22174	0.414	126.12	.21909	0.365	125.61	.21665	0.326	125.19	.214
330	0.485	128.33	.22411	0.421	127.98	.22145	0.371	127.47	.21904	0.331	127.08	.216
340	0.492	130.17	.22646	0.428	129.85	.22381	0.377	129.34	.22140	0.337	128.98	.219
350	0.500	132.02	0.22880	0.435	131.73	0.22616	0.383	131.23	0.22374	0.343	130.89	0.221
360	0.507	133.89	.23109	0.442	133.63	.22849	0.390	133.13	.22608	0.348	132.81	.223
370	0.515	135.78	.23336	0.448	135.55	.23079	0.396	135.05	.22840	0.354	134.74	.226
380							0.402	136.98	.23072	0.360	136.68	.228
390							0.408	138.91	.23301	0.365	138.63	.230
400							0.414	140.85	0.23529	0.370	140.59	0.233

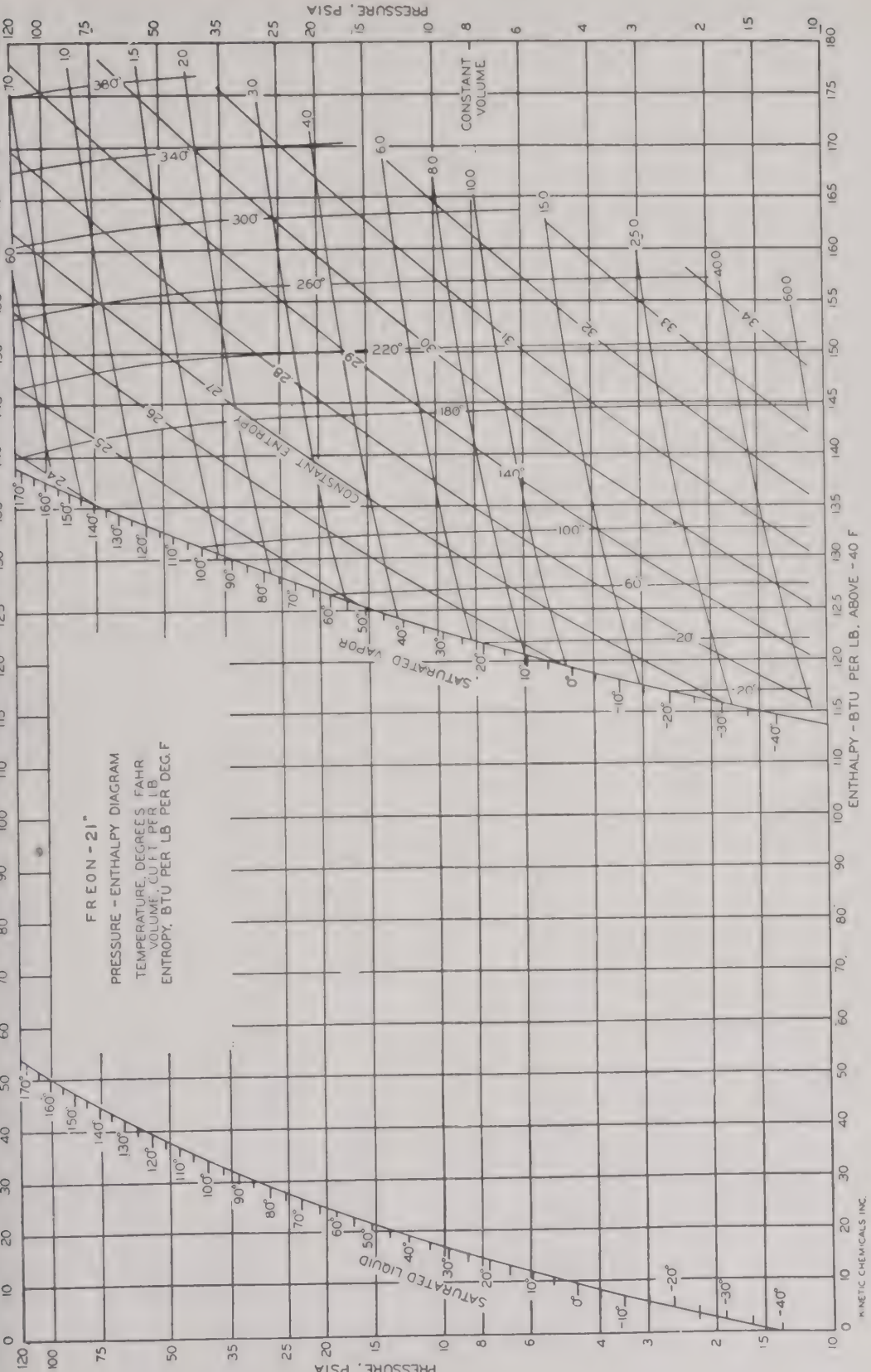

FREON-21
PRESSURE - ENTHALPY DIAGRAM
TEMPERATURE, DEGREES FAHR
VOLUME, CU FT PER LB
ENTROPY, BTU PER LB PER DEG F
PRESSURE, PSIA
ENTHALPY - BTU PER LB. ABOVE - 40 F
SATURATED LIQUID
SATURATED VAPOR
CONSTANT ENTROPY
CONSTANT VOLUME
KINETIC CHEMICALS INC.

Chart 9

Table 16. F-21 ($CHCl_2F$) Dichloromonofluoromethane—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−40	1.358	27.16*	0.01058	32.09	0.00	114.56	0.0000	0.2730
−30	1.888	26.08*	.01066	23.61	2.36	115.76	.0055	.2695
−20	2.578	24.67*	.01075	17.66	4.71	116.96	.0109	.2663
−10	3.463	22.87*	.01084	13.43	7.07	118.17	.0162	.2633
0	4.582	20.59*	.01093	10.35	9.44	119.37	.0214	.2606
2	4.838	20.07*	0.01095	9.840	9.92	119.61	0.0225	0.2601
4	5.105	19.53*	.01096	9.361	10.39	119.85	.0235	.2596
5†	5.243	19.25*	.01097	9.132	10.63	119.97	.0240	.2593
6	5.384	18.96*	.01098	8.910	10.86	120.09	.0245	.2591
8	5.674	18.37*	.01100	8.486	11.33	120.33	.0255	.2586
10	5.978	17.75*	0.01102	8.085	11.81	120.57	0.0265	0.2581
12	6.294	17.11*	.01104	7.707	12.29	120.81	.0275	.2577
14	6.625	16.43*	.01106	7.349	12.77	121.05	.0286	.2572
16	6.968	15.73*	.01108	7.012	13.25	121.30	.0296	.2567
18	7.325	15.01*	.01110	6.694	13.73	121.54	.0306	.2563
20	7.699	14.25*	0.01112	6.392	14.21	121.78	0.0316	0.2559
22	8.087	13.46*	.01114	6.107	14.68	122.02	.0326	.2555
24	8.488	12.64*	.01116	5.838	15.16	122.26	.0336	.2550
26	8.906	11.79*	.01118	5.584	15.64	122.50	.0346	.2546
28	9.341	10.90*	.01120	5.342	16.12	122.74	.0356	.2542
30	9.793	9.98*	0.01122	5.112	16.61	122.98	0.0365	0.2538
32	10.26	9.03*	.01124	4.894	17.09	123.22	.0375	.2534
34	10.75	8.03*	.01126	4.688	17.58	123.46	.0385	.2530
36	11.26	7.00*	.01128	4.492	18.07	123.71	.0395	.2526
38	11.78	5.94*	.01130	4.306	18.55	123.95	.0405	.2523
40	12.32	4.84*	0.01132	4.130	19.04	124.19	0.0414	0.2519
42	12.88	3.70*	.01134	3.963	19.53	124.43	.0424	.2516
44	13.46	2.52*	.01136	3.804	20.02	124.67	.0434	.2512
46	14.07	1.28*	.01138	3.651	20.51	124.91	.0444	.2509
48	14.69	0.01*	.01140	3.507	21.00	125.15	.0453	.2505
50	15.33	0.63	0.01142	3.370	21.49	125.39	0.0463	0.2502
52	16.00	1.30	.01145	3.239	21.99	125.63	.0473	.2499
54	16.69	1.99	.01147	3.114	22.48	125.87	.0482	.2495
56	17.40	2.70	.01149	2.995	22.98	126.11	.0492	.2492
58	18.14	3.44	.01151	2.881	23.48	126.36	.0502	.2489
60	18.90	4.20	0.01153	2.773	23.98	126.60	0.0511	0.2486
62	19.69	4.99	.01155	2.669	24.49	126.84	.0521	.2483
64	20.50	5.80	.01158	2.570	24.99	127.08	.0530	.2480
66	21.34	6.64	.01160	2.476	25.49	127.32	.0540	.2477
68	22.20	7.50	.01162	2.386	25.99	127.56	.0549	.2474
70	23.08	8.38	0.01164	2.300	26.49	127.79	0.0559	0.2471
72	24.00	9.30	.01167	2.218	27.00	128.03	.0568	.2469
74	24.95	10.25	.01169	2.139	27.51	128.27	.0577	.2466
76	25.92	11.22	.01171	2.064	28.02	128.51	.0587	.2463
78	26.93	12.23	.01174	1.992	28.52	128.74	.0596	.2460
80	27.96	13.26	0.01176	1.923	29.03	128.98	0.0606	0.2458
82	29.02	14.32	.01178	1.857	29.54	129.21	.0615	.2455
84	30.11	15.41	.01181	1.794	30.05	129.44	.0624	.2453
86†	31.23	16.53	.01183	1.733	30.56	129.68	.0634	.2450
88	32.39	17.69	.01186	1.675	31.07	129.91	.0643	.2448
90	33.58	18.88	0.01188	1.619	31.59	130.14	0.0652	0.2446
92	34.80	20.10	.01190	1.565	32.11	130.38	.0662	.2443
94	36.06	21.36	.01193	1.513	32.63	130.61	.0671	.2441
96	37.35	22.65	.01195	1.464	33.15	130.84	.0680	.2439
98	38.68	23.98	.01198	1.417	33.67	131.07	.0690	.2437
100	40.04	25.34	0.01200	1.371	34.18	131.29	0.0699	0.2434
110	47.40	32.70	.01213	1.169	36.79	132.42	.0745	.2424
120	55.75	41.05	.01226	1.001	39.46	133.53	.0791	.2414
130	65.15	50.45	.01240	.8623	42.13	134.61	.0837	.2405
140	75.72	61.02	.01254	.7457	44.86	135.66	.0882	.2396
150	87.51	72.81	0.01269	0.6476	47.62	136.68	0.0927	0.2388
160	100.6	85.91	.01284	.5646	50.43	137.69	.0972	.2381

* Inches of mercury below one atmosphere.
† Standard cycle temperatures.

Table 17. F-21 ($CHCl_2F$) Dichloromonofluoromethane—Properties of Superheated Vapor

mp F	Pressure 2 psia Gage pressure 25.85 in. vac (Sat temp −28.2 F)			Pressure 4 psia Gage pressure 21.78 in. vac (Sat temp 4.9 F)			Pressure 6 psia Gage pressure 17.71 in. vac (Sat temp 10.1 F)			Pressure 8 psia Gage pressure 13.64 in. vac (Sat temp 21.6 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
sat)	*(22.37)*	*(115.98)*	*(0.2689)*	*(11.74)*	*(118.78)*	*(0.2619)*	*(8.056)*	*(120.58)*	*(0.2581)*	*(6.169)*	*(121.96)*	*(0.2556)*
0	23.85	119.49	0.2768	11.87	119.40	0.2633	—	—	—	—	—	—
10	24.38	120.76	.2795	12.14	120.66	.2660	—	—	—	—	—	—
20	24.90	122.04	.2882	12.40	121.95	.2687	8.232	121.86	0.2608	—	—	—
30	25.43	123.34	.2849	12.66	123.25	.2714	8.408	123.16	.2635	6.281	123.06	0.2578
40	25.95	124.65	.2876	12.93	124.57	.2741	8.585	124.48	.2661	6.414	124.39	.2605
50	26.47	125.98	0.2902	13.19	125.90	0.2767	8.762	125.81	0.2688	6.547	125.73	0.2631
60	27.00	127.33	.2928	13.45	127.25	.2793	8.939	127.17	.2714	6.681	127.08	.2657
70	27.52	128.69	.2954	13.72	128.61	.2819	9.115	128.53	.2740	6.814	128.44	.2683
80	28.05	130.07	.2979	13.98	129.99	.2845	9.291	129.91	.2766	6.947	129.82	.2709
90	28.57	131.45	.3005	14.24	131.37	.2870	9.467	131.29	.2791	7.079	131.21	.2735
00	29.09	132.85	0.3030	14.51	132.77	0.2895	9.643	132.69	0.2816	7.212	132.61	0.2760
10	29.62	134.26	.3055	14.77	134.18	.2920	9.819	134.11	.2841	7.345	134.03	.2785
20	30.14	135.68	.3080	15.03	135.61	.2945	9.995	135.54	.2866	7.477	135.46	.2810
30	30.66	137.12	.3104	15.29	137.04	.2970	10.17	136.97	.2891	7.609	136.90	.2835
40	31.19	138.57	.3129	15.56	138.49	.2994	10.35	138.42	.2915	7.742	138.35	.2859
50	31.71	140.03	0.3153	15.82	139.96	0.3019	10.52	139.89	0.2940	7.874	139.82	0.2883
60	32.23	141.52	.3177	16.08	141.45	.3043	10.70	141.39	.2964	8.006	141.31	.2908
70	32.76	143.02	.3201	16.35	142.95	.3067	10.87	142.89	.2988	8.138	142.82	.2932
80	33.28	144.53	.3225	16.61	144.46	.3091	11.05	144.40	.3012	8.270	144.33	.2956
90	33.80	146.05	.3249	16.87	145.98	.3114	11.22	145.92	.3035	8.402	145.86	.2979

mp F	Pressure 10 psia Gage pressure 9.57 in. vac (Sat temp 30.9 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp 62.8 F)			Pressure 30 psia Gage pressure 15.3 psig (Sat temp 83.8 F)			Pressure 40 psia Gage pressure 25.3 psig (Sat temp 99.9 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
sat)	*(5.014)*	*(123.10)*	*(0.2536)*	*(2.630)*	*(126.94)*	*(0.2482)*	*(1.800)*	*(129.42)*	*(0.2453)*	*(1.378)*	*(131.28)*	*(0.2435)*
00	5.753	132.54	0.2176	2.834	132.14	0.2578	1.860	131.72	0.2495	1.373	131.29	0.2435
10	5.860	133.95	.2741	2.889	133.56	.2603	1.898	133.16	.2520	1.402	132.74	.2460
20	5.966	135.38	.2766	2.943	135.00	.2628	1.935	134.61	.2546	1.430	134.20	.2486
30	6.072	136.82	.2791	2.998	136.45	.2653	1.972	136.06	.2571	1.459	135.67	.2511
40	6.178	138.28	.2815	3.052	137.91	.2678	2.009	137.53	.2596	1.487	137.15	.2536
50	6.285	139.75	0.2840	3.106	139.39	0.2702	2.046	139.02	0.2620	1.515	138.65	0.2561
60	6.391	141.24	.2864	3.160	140.90	.2727	2.083	140.53	.2645	1.543	140.17	.2585
70	6.497	142.75	.2888	3.214	142.41	.2751	2.119	142.06	.2669	1.572	141.70	.2610
80	6.603	144.27	.2912	3.268	143.93	.2775	2.156	143.59	.2693	1.600	143.24	.2634
90	6.709	145.79	.2936	3.322	146.46	.2799	2.193	145.13	.2717	1.628	144.79	.2658
00	6.815	147.33	0.2959	3.376	147.01	0.2822	2.229	146.68	0.2741	1.656	146.35	0.2682
10	6.921	148.89	.2983	3.430	148.58	.2846	2.266	148.25	.2764	1.683	147.93	.2706
20	7.027	150.46	.3006	3.484	150.16	.2869	2.302	149.84	.2788	1.711	149.52	.2729
30	7.133	152.05	.3029	3.537	151.75	.2892	2.338	151.44	.2811	1.739	151.13	.2753
40	7.329	153.65	.3052	3.591	153.35	.2915	2.375	153.05	.2834	1.767	152.75	.2776
50	7.344	155.26	0.3075	3.645	154.97	0.2938	2.411	154.68	0.2857	1.794	154.38	0.2799
60	7.450	156.88	.3098	3.698	156.60	.2961	2.447	156.31	.2880	1.822	156.01	.2822
70	7.555	158.52	.3120	3.752	158.24	.2984	2.483	157.95	.2903	1.849	157.66	.2845
80	7.661	160.17	.3143	3.805	159.90	.3007	2.520	159.61	.2926	1.877	159.33	.2868
90	7.767	161.84	.3165	3.859	161.57	.3029	2.556	161.30	.2948	1.904	161.02	.2890

mp F	Pressure 50 psia Gage pressure 35.3 psig (Sat temp 113.3 F)			Pressure 60 psia Gage pressure 45.3 psig (Sat temp 124.7 F)			Pressure 70 psia Gage pressure 55.3 psig (Sat temp 134.7 F)			Pressure 80 psia Gage pressure 65.3 psig (Sat temp 143.8 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
sat)	*(1.111)*	*(132.79)*	*(0.2421)*	*(0.9333)*	*(134.03)*	*(0.2410)*	*(0.8045)*	*(135.10)*	*(0.2401)*	*(0.7069)*	*(136.06)*	*(0.2393)*
50	1.197	138.26	0.2513	0.9836	137.86	0.2474	0.8312	137.45	0.2440	0.7167	137.01	0.2409
60	1.220	139.79	.2538	1.003	139.49	.2499	.8486	139.01	.2465	.7321	138.58	.2435
70	1.243	141.33	.2563	1.023	140.95	.2524	.8658	140.58	.2490	.7476	140.16	.2460
80	1.265	142.88	.2587	1.042	142.51	.2548	.8828	142.13	.2515	.7629	141.74	.2485
90	1.288	144.43	.2612	1.062	144.07	.2573	.8998	143.71	.2539	.7781	143.33	.2509
00	1.311	146.00	0.2636	1.081	145.65	0.2597	0.9168	145.30	0.2564	0.7933	144.93	0.2534
10	1.334	147.59	.2659	1.100	147.25	.2621	.9336	146.90	.2588	.8083	146.54	.2558
20	1.356	149.19	.2683	1.120	148.86	.2645	.9504	148.52	.2612	.8233	148.17	.2582
30	1.379	150.80	.2707	1.139	150.48	.2668	.9671	150.15	.2635	.8382	149.81	.2606
40	1.401	152.42	.2730	1.158	152.11	.2692	.9837	151.79	.2659	.8530	151.46	.2630
50	1.424	154.06	0.2753	1.177	153.76	0.2715	1.000	153.44	0.2682	0.8678	153.12	0.2654
60	1.446	155.71	.2776	1.196	155.41	.2738	1.017	155.10	.2706	.8825	154.78	.2677
70	1.469	157.37	.2799	1.215	157.08	.2761	1.033	156.77	.2729	.8971	156.46	.2700
80	1.491	159.05	.2822	1.234	158.76	.2784	1.050	158.45	.2752	.9117	158.16	.2723
90	1.513	160.74	.2845	1.252	160.45	.2807	1.066	160.16	.2775	.9262	159.87	.2746
00	1.535	162.44	0.2867	1.271	162.16	0.2830	1.082	161.88	0.2797	0.9406	161.59	0.2769
10	1.558	164.16	.2890	1.290	163.88	.2852	1.099	163.61	.2820	.9550	163.32	.2792
20	1.580	165.89	.2912	1.309	165.62	.2875	1.115	165.35	.2842	.9694	165.07	.2814
30	1.602	167.63	.2934	1.327	167.36	.2897	1.131	167.10	.2865	.9838	166.83	.2837
40	1.624	169.38	.2956	1.346	169.12	.2919	1.147	168.86	.2887	.9981	168.60	.2859

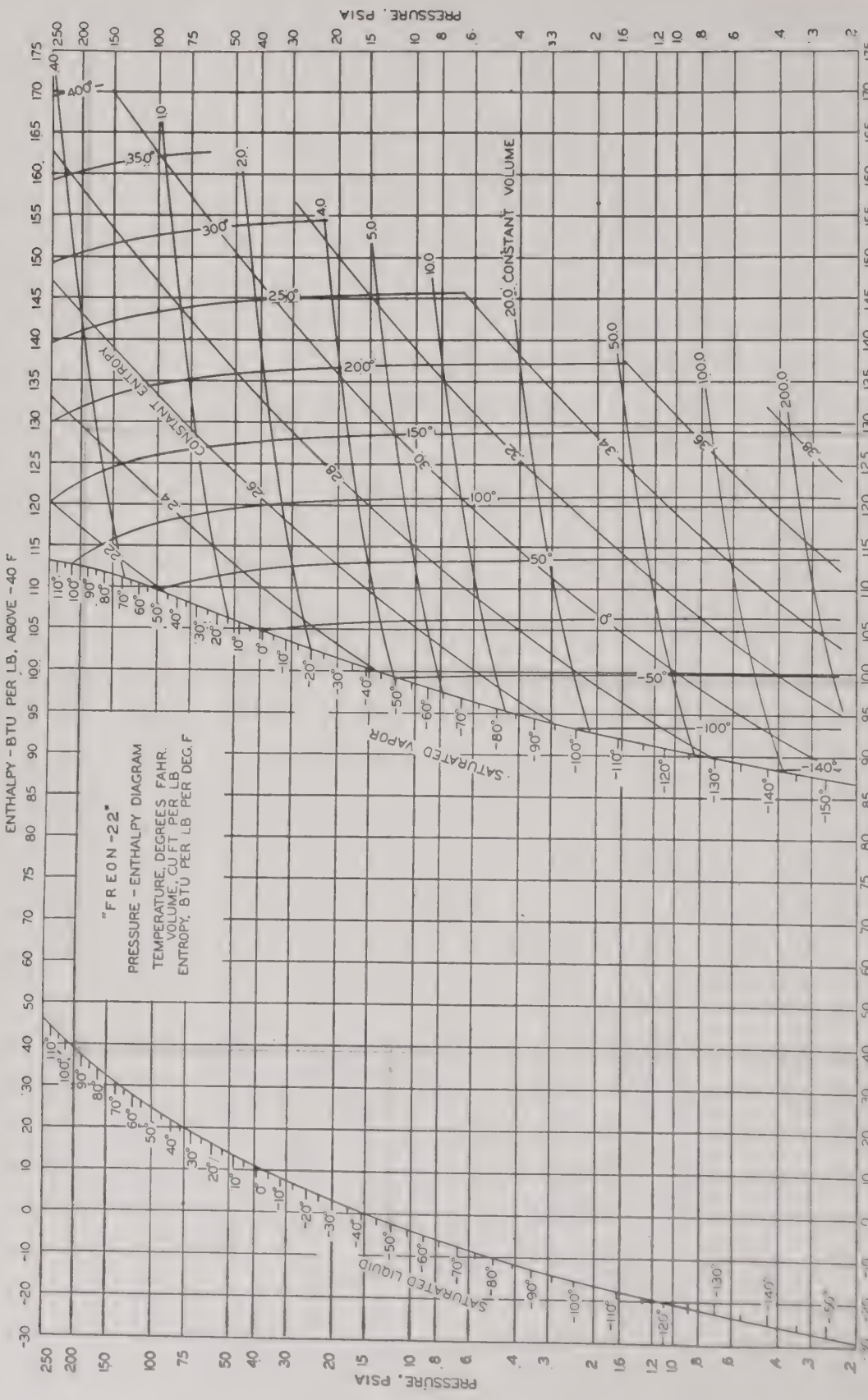

"FREON-22"
PRESSURE - ENTHALPY DIAGRAM
TEMPERATURE, DEGREES FAHR.
VOLUME, CU FT PER LB
ENTROPY, BTU PER LB PER DEG. F
ENTHALPY - BTU PER LB, ABOVE -40 F
PRESSURE, PSIA
SATURATED LIQUID
SATURATED VAPOR
CONSTANT ENTROPY
CONSTANT VOLUME

Chart 10

Table 18. F-22 ($CHClF_2$) Monochlorodifluoromethane—Properties of Liquid and Saturated Vapor

Temp F t	Pressure psia	Pressure psig	Liquid, density lb/cu ft $1/v_f$	Vapor, sp vol cu ft/lb v_g	Enthalpy, datum −40 F Btu per lb Liquid h_f	Enthalpy, datum −40 F Btu per lb Vapor h_g	Entropy, datum −40 F Btu per lb F Liquid s_f	Entropy, datum −40 F Btu per lb F Vapor s_g
−155	0.19901	29.51*	97.67	188.1	−29.07	86.78	−0.0808	0.2996
−150	0.2605	29.39*	97.33	146.1	−27.79	87.36	−0.0767	0.2952
−145	0.3375	29.23*	96.99	114.5	−26.52	87.94	− .0727	.2912
−140	0.4332	29.04*	96.63	90.61	−25.25	88.53	−0.0687	0.2874
−135	0.5511	28.80*	96.27	72.33	−23.99	89.11	− .0647	.2837
−130	0.6949	28.51*	95.91	58.21	−22.73	89.70	−0.0609	0.2803
−125	0.8692	28.15*	95.53	47.23	−21.47	90.29	− .0571	.2770
−120	1.079	27.72*	95.15	38.60	−20.22	90.88	−0.0534	0.2738
−115	1.329	27.21*	94.76	31.77	−18.98	91.47	− .0497	.2708
−110	1.626	26.61*	94.37	26.33	−17.73	92.07	−0.0461	0.2680
−105	1.976	25.90*	93.97	21.96	−16.48	92.67	− .0425	.2653
−100	2.386	25.06*	93.56	18.43	−15.23	93.27	−0.0390	0.2627
− 95	2.865	24.09*	93.14	15.54	−13.98	93.87	− .0356	.2602
− 90	3.417	22.96*	92.72	13.20	−12.73	94.47	−0.0322	0.2579
− 85	4.055	21.67*	92.29	11.26	−11.47	95.08	− .0288	.2556
− 80	4.787	20.18*	91.85	9.650	−10.22	95.68	−0.0255	0.2535
− 78	5.100	19.55*	91.67	9.086	− 9.72	95.92	− .0242	.2526
− 76	5.430	18.87*	91.49	8.561	− 9.21	96.16	− .0229	.2518
− 74	5.79	18.14*	91.31	8.072	− 8.70	96.40	− .0216	.2510
− 72	6.17	17.37*	91.13	7.616	− 8.20	96.64	− .0203	.2502
− 70	6.57	16.55*	90.95	7.192	− 7.69	96.88	−0.0253	0.2494
− 68	6.99	15.70*	90.77	6.795	− 7.19	97.12	− .0177	.2487
− 66	7.40	14.86*	90.58	6.426	− 6.68	97.36	− .0164	.2479
− 64	7.86	13.93*	90.39	6.079	− 6.17	97.60	− .0151	.2472
− 62	8.35	12.93*	90.21	5.755	− 5.67	97.84	− .0138	.2465
− 60	8.86	11.89*	90.03	5.452	− 5.16	98.08	−0.0126	0.2458
− 58	9.39	10.81*	89.84	5.166	− 4.65	98.32	− .0113	.2451
− 56	9.94	9.69*	89.65	4.900	− 4.13	98.56	− .0100	.2444
− 54	10.51	8.53*	89.46	4.650	− 3.61	98.80	− .0087	.2438
− 52	11.11	7.31*	89.27	4.415	− 3.09	99.04	− .0075	.2431
− 50	11.74	6.03*	89.08	4.192	− 2.58	99.28	−0.0062	0.2425
− 48	12.40	4.68*	88.88	3.986	− 2.06	99.52	− .0050	.2418
− 46	13.09	3.28*	88.68	3.793	− 1.54	99.76	− .0037	.2412
− 44	13.80	1.83*	88.49	3.611	− 1.02	100.00	− .0025	.2406
− 42	14.54	0.326*	88.30	3.440	− 0.51	100.23	− .0012	.2400
− 40	15.31	0.610	88.10	3.279	0.00	100.46	0.0000	0.2394
− 38	16.12	1.42	87.90	3.126	0.53	100.70	.0013	.2389
− 36	16.97	2.27	87.70	2.981	1.05	100.93	.0025	.2383
− 34	17.85	3.15	87.50	2.844	1.58	101.17	.0037	.2377
− 32	18.77	4.07	87.29	2.713	2.10	101.40	.0050	.2372
− 30	19.72	5.02	87.09	2.590	2.62	101.63	0.0062	0.2367
− 28	20.71	6.01	86.89	2.474	3.15	101.86	.0074	.2361
− 26	21.73	7.03	86.69	2.365	3.69	102.10	.0086	.2356
− 24	22.79	8.09	86.48	2.262	4.22	102.33	.0099	.2351
− 22	23.88	9.18	86.27	2.165	4.75	102.56	.0111	.2346
− 20	25.01	10.31	86.06	2.074	5.28	102.79	0.0123	0.2341
− 18	26.18	11.48	85.85	1.987	5.82	103.02	.0135	.2336
− 16	27.39	12.69	85.64	1.905	6.40	103.25	.0147	.2331
− 14	28.64	13.94	85.43	1.827	6.90	103.48	.0159	.2326
− 12	29.94	15.24	85.21	1.752	7.43	103.70	.0170	.2321
− 10	31.29	16.59	84.99	1.681	7.96	103.92	0.0182	0.2316
− 8	32.69	17.99	84.78	1.613	8.49	104.14	.0194	.2312
− 6	34.14	19.44	84.56	1.549	9.02	104.36	.0205	.2307
− 4	35.64	20.94	84.34	1.488	9.55	104.58	.0217	.2302
− 2	37.19	22.49	84.12	1.429	10.09	104.80	.0228	.2298
0	38.79	24.09	83.90	1.373	10.63	105.02	0.0240	0.2293
2	40.43	25.73	83.68	1.320	11.17	105.24	0.0251	0.2289
4	42.14	27.44	83.45	1.270	11.70	105.45	.0262	.2285
5	43.02	28.33	83.34	1.246	11.97	105.56	.0268	.2283
6	43.91	29.21	83.23	1.221	12.23	105.66	.0274	.2280
8	45.74	31.04	83.01	1.175	12.76	105.87	.0285	.2276
10	47.63	32.93	82.78	1.130	13.29	106.08	0.0296	0.2272
12	49.58	34.88	82.55	1.088	13.82	106.29	.0307	.2268
14	51.59	36.89	82.32	1.048	14.36	106.50	.0319	.2264
16	53.66	38.96	82.09	1.009	14.90	106.71	.0330	.2260
18	55.79	41.09	81.86	0.9721	15.44	106.92	.0341	.2257

* Inches mercury below one atmosphere

Table 18. F-22 ($CHClF_2$) Monochlorodifluoromethane—Properties of Liquid and Saturated Vapor (Concluded)

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
20	57.98	43.28	81.63	0.9369	15.98	107.13	0.0352	0.2253
22	60.23	45.53	81.39	.9032	16.52	107.33	.0364	.2249
24	62.55	47.85	81.16	.8707	17.06	107.53	.0375	.2246
26	64.94	50.24	80.92	.8398	17.61	107.73	.0379	.2242
28	67.40	52.70	80.69	.8100	18.17	107.93	.0398	.2239
30	69.93	55.23	80.45	0.7816	18.74	108.13	0.0409	0.2235
32	72.53	57.83	80.21	.7543	19.32	108.33	.0421	.2232
34	75.21	60.51	79.97	.7283	19.90	108.52	.0433	.2228
36	77.97	63.27	79.73	.7032	20.49	108.71	.0445	.2225
38	80.81	66.11	79.49	.6791	21.09	108.90	.0457	.2222
40	83.72	69.02	79.25	0.6559	21.70	109.09	0.0469	0.2218
42	86.69	71.99	79.00	.6339	22.29	109.27	.0481	.2215
44	89.74	75.04	78.76	.6126	22.90	109.45	.0493	.2211
46	92.88	78.18	78.51	.5922	23.50	109.63	.0505	.2208
48	96.10	81.40	78.26	.5726	24.11	109.80	.0516	.2205
50	99.40	84.70	78.02	0.5537	24.73	109.98	0.0528	0.2201
52	102.8	88.10	77.77	.5355	25.34	110.14	.0540	.2198
54	106.2	91.5	77.51	.5184	25.95	110.30	.0552	.2194
56	109.8	95.1	77.26	.5014	26.58	110.47	.0564	.2191
58	113.5	98.8	77.01	.4849	27.22	110.63	.0576	.2188
60	117.2	102.5	76.75	0.4695	27.83	110.78	0.0588	0.2185
62	121.0	106.3	76.50	.4546	28.46	110.93	.0600	.2181
64	124.9	110.2	76.24	.4403	29.09	111.08	.0612	.2178
66	128.9	114.2	75.98	.4264	29.72	111.22	.0624	.2175
68	133.0	118.3	75.72	.4129	30.35	111.35	.0636	.2172
70	137.2	122.5	75.46	0.4000	30.99	111.49	0.0648	0.2168
72	141.5	126.8	75.20	.3875	31.65	111.63	.0661	.2165
74	145.9	131.2	74.94	.3754	32.29	111.75	.0673	.2162
76	150.4	135.7	74.68	.3638	32.94	111.88	.0684	.2158
78	155.0	140.3	74.41	.3526	33.61	112.01	.0696	.2155
80	159.7	145.0	74.15	0.3417	34.27	112.13	0.0708	0.2151
82	164.5	149.8	73.89	.3313	34.92	112.24	.0720	.2148
84	169.4	154.7	73.63	.3212	35.60	112.36	.0732	.2144
86	174.5	159.8	73.36	.3113	36.28	112.47	.0744	.2140
88	179.6	164.9	73.09	.3019	36.94	112.57	.0756	.2137
90	184.8	170.1	72.81	0.2928	37.61	112.67	0.0768	0.2133
92	190.1	175.4	72.53	.2841	38.28	112.76	.0780	.2130
94	195.6	180.9	72.24	.2755	38.97	112.85	.0792	.2126
96	201.2	186.5	71.95	.2672	39.65	112.93	.0803	.2122
98	206.8	192.1	71.65	.2594	40.32	113.00	.0815	.2119
100	212.6	197.9	71.35	0.2517	40.98	113.06	0.0827	0.2115
102	218.5	203.8	71.05	.2443	41.65	113.12	.0839	.2111
104	224.6	209.9	70.74	.2370	42.32	113.16	.0851	.2107
106	230.7	216.0	70.42	.2301	42.98	113.20	.0862	.2104
108	237.0	222.3	70.11	.2233	43.66	113.24	.0874	.2100
110	243.4	228.7	69.78	0.2167	44.35	113.29	0.0886	0.2096
112	249.9	235.2	69.45	.2104	45.04	113.34	.0898	.2093
114	256.6	241.9	69.12	.2043	45.74	113.38	.0909	.2089
116	263.4	248.7	68.78	.1983	46.44	113.42	.0921	.2085
118	270.3	255.6	68.44	.1926	47.14	113.46	.0933	.2081
120	277.3	262.6	68.10	0.1871	47.85	113.52	0.0945	0.2078
122	384.4	269.7	67.75	.1825	48.6	113.57		
124	391.6	276.9	67.40	.1772	49.4	113.61		
126	398.8	284.1	67.05	.1724	50.2	113.65		
128	306.1	291.4	66.70	.1675	50.8	113.69		
130	313.5	298.8	66.35	0.1629	51.5	113.71		
132	321.0	306.3	66.00	.1585	52.3	113.74		
134	328.7	314.0	65.65	.1538	53.1	113.77		
136	336.6	321.9	65.25	.1492	53.8	113.79		
138	344.6	329.9	64.85	.1449	54.6	113.80		
140	352.7	338.0	64.45	0.1408	55.3	113.81		
142	361.0	346.3	64.05	.1368	56.1	113.80		
144	369.7	355.0	63.65	.1330	56.9	113.79		
146	379.0	364.3	63.25	.1292	57.7	113.78		
148	388.8	374.1	62.85	.1253	58.4	113.76		
150	399.0	384.3	62.45	0.1216	59.2	113.74		
152	407.0	392.3	62.02	.1179	60.0	113.71		
154	416.0	401.3	61.58	.1141	60.8	113.67		
156	426.0	411.3	61.13	.1105	61.6	113.62		
158	436.5	421.8	60.67	.1070	62.5	113.56		
160	448.0	433.3	60.20	0.1035	63.5	113.50		

Table 19. F-22 ($CHClF_2$) Monochlorodifluoromethane—Properties of Superheated Vapor

mp	Pressure .25 psia Gage pressure 29.41 in. vac (Sat temp −150.9 F)			Pressure .30 psia Gage pressure 29.31 in. vac (Sat temp −147.4 F)			Pressure .35 psia Gage pressure 29.21 in. vac (Sat temp −144.3 F)			Pressure .40 psia Gage pressure 29.11 in. vac (Sat temp −141.7 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
sat)	*(151.8)*	*(87.26)*	*(0.2960)*	*(127.9)*	*(87.66)*	*(0.2932)*	*(110.7)*	*(88.02)*	*(0.2907)*	*(97.62)*	*(88.33)*	*(0.2887)*
50	152.19	87.36	.2962									
40	157.12	88.53	.2999	130.91	88.53	.2958	112.19	88.53	.2923	98.15	88.53	.2892
30	162.04	89.72	.3036	135.01	89.72	.2995	115.71	89.72	.2959	101.23	89.71	.2929
20	166.95	90.93	.3072	139.12	90.92	.3031	119.23	90.92	.2995	104.31	90.92	.2965
10	171.89	92.14	.3107	143.22	92.14	.3066	122.75	92.14	.3031	107.39	92.14	.3000
00	176.81	93.38	.3142	147.33	93.37	.3101	126.26	93.37	.3065	110.47	93.37	.3035
90	181.74	94.62	.3176	151.43	94.62	.3135	129.78	94.62	.3100	113.55	94.62	.3069
80	186.66	95.89	.3210	155.53	95.88	.3168	133.30	95.88	.3133	116.62	95.88	.3103
70	191.59	97.16	.3243	159.64	97.16	.3201	136.82	97.16	.3166	119.70	97.16	.3136
60	196.51	98.46	.3276	163.74	98.45	.3234	140.33	98.45	.3199	122.78	98.45	.3168
50	201.43	99.76	.3308	167.84	99.76	.3266	143.85	99.76	.3231	125.86	99.75	.3201
40	206.35	101.08	.3340	171.95	101.08	.3298	147.37	101.08	.3263	128.94	101.08	.3233
30	211.28	102.42	.3371	176.05	102.42	.3330	150.89	102.41	.3295	132.01	102.41	.3264
20	216.20	103.77	.3403	180.15	103.77	.3361	154.40	103.77	.3326	135.09	103.76	.3295
10	221.12	105.14	.3433	184.25	105.13	.3392	157.92	105.13	.3357	138.17	105.13	.3326
0	226.04	106.52	.3464	188.36	106.51	.3422	161.44	106.51	.3387	141.25	106.51	.3357
10	230.97	107.91	.3494	192.46	107.91	.3452	164.96	107.91	.3417	144.33	107.91	.3387
20	235.89	109.32	.3524	196.56	109.32	.3482	168.47	109.32	.3447	147.40	109.32	.3417
30	240.81	110.75	.3553	200.66	110.75	.3512	171.99	110.74	.3476	150.48	110.74	.3446
40	245.73	112.19	.3582	204.76	112.18	.3541	175.50	112.18	.3506	153.56	112.18	.3475
50	250.66	113.64	.3611	208.87	113.64	.3569	179.02	113.64	.3534	156.63	113.64	.3504
60	255.58	115.11	.3639	212.97	115.11	.3598	182.54	115.11	.3563	159.71	115.10	.3532
70	260.50	116.59	.3668	217.07	116.59	.3626	186.05	116.59	.3591	162.79	116.59	.3561
80	265.42	118.09	.3696	221.17	118.09	.3654	189.57	118.09	.3619	165.87	118.09	.3589
90	270.34	119.61	.3724	225.28	119.60	.3682	193.09	119.60	.3647	168.94	119.60	.3617
00	275.27	121.13	.3751	229.38	121.13	.3710	196.60	121.13	.3675	172.02	121.13	.3644
10	280.19	122.68	.3779	233.48	122.68	.3737	200.12	122.68	.3702	175.10	122.67	.3672
20	285.11	124.24	.3806	237.58	124.23	.3764	203.63	124.23	.3729	178.17	124.23	.3699
30	290.03	125.81	.3833	241.68	125.81	.3791	207.15	125.81	.3756	181.25	125.80	.3726
40	294.95	127.40	.3859	245.78	127.40	.3818	210.66	127.39	.3783	184.33	127.39	.3752
50	299.87	129.00	.3886	249.89	129.00	.3844	214.18	129.00	.3809	187.40	129.00	.3779
60				253.99	130.62	.3871	217.70	130.61	.3835	190.48	130.61	.3805

mp	Pressure .45 psia Gage pressure 29.00 in. vac (Sat temp −139.3 F)			Pressure .50 psia Gage pressure 28.90 in. vac (Sat temp −137.2 F)			Pressure .55 psia Gage pressure 28.80 in. vac (Sat temp −135.0 F)			Pressure .60 psia Gage pressure 28.70 in. vac (Sat temp −133.3 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
sat)	*(87.42)*	*(88.61)*	*(0.2868)*	*(79.18)*	*(88.80)*	*(0.2853)*	*(72.47)*	*(89.10)*	*(0.2838)*	*(66.77)*	*(89.31)*	*(0.2825)*
30	89.97	89.71	.2902	80.96	89.71	.2878	73.59	89.71	.2856	67.44	89.70	.2837
20	92.70	90.92	.2938	83.42	90.91	.2914	75.83	90.91	.2892	69.50	90.91	.2873
10	95.44	92.13	.2974	85.89	92.13	.2949	78.07	92.13	.2928	71.55	92.13	.2908
00	98.18	93.37	.3008	88.35	93.36	.2984	80.31	93.36	.2962	73.61	93.36	.2943
90	100.92	94.61	.3042	90.81	94.61	.3018	82.55	94.91	.2997	75.66	94.61	.2977
80	103.65	95.88	.3076	93.28	95.87	.3052	84.79	95.87	.3031	77.71	95.87	.3010
70	106.39	97.15	.3109	95.74	97.15	.3085	87.03	97.15	.3063	79.77	97.15	.3043
60	109.13	98.45	.3142	98.21	98.44	.3118	89.27	98.44	.3096	81.32	98.44	.3076
50	111.86	99.75	.3174	100.67	99.75	.3150	91.51	99.75	.3128	83.88	99.75	.3108
40	114.60	101.07	.3206	103.13	101.07	.3182	93.75	101.07	.3160	85.93	101.07	.3140
30	117.34	102.41	.3237	105.60	102.41	.3213	95.99	102.41	.3192	87.98	102.40	.3172
20	120.07	103.76	.3269	108.06	103.76	.3245	98.23	103.76	.3223	90.03	103.76	.3203
10	122.81	105.13	.3299	110.52	105.13	.3275	100.46	105.12	.3254	92.09	105.12	.3234
0	125.54	106.51	.3330	112.98	106.51	.3306	102.70	106.50	.3284	94.14	106.50	.3264
10	128.28	107.90	.3360	115.45	107.90	.3336	104.94	107.90	.3314	96.19	107.90	.3294
20	131.02	109.31	.3390	117.91	109.31	.3366	107.18	109.31	.3344	98.24	109.31	.3324
30	133.75	110.74	.3419	120.37	110.74	.3395	109.42	110.74	.3373	100.30	110.73	.3354
40	136.49	112.18	.3448	122.83	112.18	.3424	111.66	112.18	.3403	102.35	112.17	.3383
50	139.22	113.63	.3477	125.29	113.63	.3453	113.90	113.63	.3431	104.40	113.63	.3412
60	141.96	115.10	.3506	127.76	115.10	.3482	116.14	115.10	.3460	106.45	115.10	.3440
70	144.69	116.59	.3534	130.22	116.59	.3510	118.37	116.58	.3488	108.50	116.58	.3468
80	147.43	118.09	.3562	132.68	118.09	.3538	120.61	118.08	.3516	110.55	118.08	.3496
90	150.16	119.60	.3590	135.14	119.60	.3566	122.85	119.60	.3544	112.61	119.60	.3524
00	152.90	121.13	.3617	137.60	121.13	.3593	125.09	121.13	.3572	114.66	121.12	.3552
10	155.63	122.67	.3645	140.07	122.67	.3621	127.33	122.67	.3599	116.71	122.67	.3579
20	158.37	124.23	.3672	142.53	124.23	.3648	129.56	124.23	.3626	118.76	124.23	.3606
30	161.10	125.80	.3699	144.99	125.80	.3675	131.80	125.80	.3653	120.81	125.80	.3633
40	163.84	127.39	.3725	147.45	127.39	.3701	134.04	127.39	.3680	122.87	127.39	.3660
50	166.57	128.99	.3752	149.91	128.99	.3728	136.28	128.99	.3706	124.92	128.99	.3686
60	169.31	130.61	.3778	152.37	130.61	.3754	138.52	130.61	.3732	126.97	130.61	.3713
70	172.04	132.24	.3804	154.83	132.24	.3780	140.75	132.24	.3759	129.02	132.24	.3739

Table 19. F-22 ($CHClF_2$) Monochlorodifluoromethane— Properties of Superheated Vapor (Continued)

Temp F	Pressure .65 psia Gage pressure 28.60 in. vac (Sat temp −131.6 F)			Pressure .70 psia Gage pressure 28.50 in. vac (Sat temp −129.9 F)			Pressure .75 psia Gage pressure 28.39 in. vac (Sat temp −128.4 F)			Pressure .80 psia Gage pressure 28.29 in. v (Sat temp −127.0 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
(*at sat*)	(61.94)	(89.52)	(0.2813)	(57.81)	(89.72)	(0.2802)	(54.19)	(89.89)	(0.2792)	(51.01)	(90.06)	(0.27
−130	62.25	89.70	.2818									
−120	64.14	90.90	.2854	59.55	90.90	.2837	55.58	90.90	.2822	52.09	90.90	.280
−110	66.04	92.12	.2890	61.31	92.12	.2873	57.22	92.12	.2857	53.64	92.11	.284
−100	67.94	93.36	.2924	63.08	93.35	.2907	58.86	93.35	.2892	55.18	93.35	.287
− 90	69.83	94.60	.2958	64.84	94.60	.2942	60.51	94.60	.2926	56.72	94.60	.291
− 80	71.73	95.87	.2992	66.60	95.87	.2975	62.15	95.86	.2959	58.26	95.86	.294
− 70	73.62	97.14	.3025	68.36	97.14	.3005	63.79	97.14	.2992	59.80	97.14	.297
− 60	75.52	98.44	.3058	70.12	98.44	.3041	65.44	98.43	.3025	61.34	98.43	.301
− 50	77.42	99.74	.3090	71.88	99.74	.3073	67.08	99.74	.3057	62.78	99.74	.304
− 40	79.31	101.07	.3122	73.64	101.06	.3105	68.73	101.06	.3089	64.42	101.06	.307
− 30	81.21	102.40	.3154	75.40	102.40	.3137	70.37	102.40	.3121	65.96	102.40	.310
− 20	83.10	103.75	.3185	77.16	103.75	.3168	72.01	103.75	.3152	67.50	103.75	.313
− 10	85.00	105.12	.3216	78.92	105.12	.3199	73.65	105.12	.3183	69.04	105.11	.316
0	86.89	106.50	.3246	80.68	106.50	.3229	75.30	106.50	.3213	70.58	106.50	.319
10	88.79	107.90	.3276	82.44	107.89	.3259	76.94	107.89	.3244	72.12	107.89	.322
20	90.68	109.31	.3306	84.20	109.31	.3289	78.58	109.30	.3273	73.66	109.30	.325
30	92.58	110.73	.3335	85.96	110.73	.3318	80.22	110.73	.3303	75.20	110.73	.328
40	94.47	112.17	.3364	87.72	112.17	.3348	81.86	112.17	.3332	76.74	112.17	.331
50	96.36	113.63	.3393	89.48	113.63	.3376	83.51	113.62	.3361	78.28	113.62	.334
60	98.26	115.10	.3422	91.23	115.09	.3405	85.15	115.09	.3389	79.82	115.09	.337
70	100.15	116.58	.3450	92.99	116.58	.3433	86.79	116.58	.3417	81.36	116.58	.340
80	102.05	118.08	.3478	94.75	118.08	.3461	88.43	118.08	.3446	82.90	118.08	.343
90	103.94	119.59	.3506	96.51	119.59	.3489	90.07	119.59	.3473	84.44	119.59	.345
100	105.84	121.12	.3534	98.27	121.12	.3517	91.72	121.12	.3501	85.98	121.12	.348
110	107.73	122.67	.3561	100.03	122.67	.3544	93.36	122.66	.3528	87.52	122.66	.351
120	109.62	124.22	.3588	101.79	124.22	.3571	95.00	124.22	.3555	89.06	124.22	.354
130	111.52	125.80	.3615	103.55	125.80	.3598	96.64	125.80	.3582	90.60	125.79	.356
140	113.41	127.39	.3642	105.31	127.38	.3625	98.28	127.38	.3609	92.14	127.38	.359
150	115.30	128.99	.3668	107.06	128.99	.3651	99.92	128.99	.3636	93.68	128.98	.361
160	117.20	130.61	.3694	108.82	130.60	.3678	101.56	130.60	.3662	95.21	130.60	.364
170	119.09	132.24	.3721	110.58	132.24	.3704	103.21	132.24	.3688	96.75	132.23	.367
180				112.34	133.88	.3730	104.85	133.88	.3714	98.29	133.88	.369

Temp F	Pressure .90 psia Gage pressure 28.09 in. vac (Sat temp −124.3 F)			Pressure 1.00 psia Gage pressure 27.88 in. vac (Sat temp −121.9 F)			Pressure 1.25 psia Gage pressure 27.38 in. vac (Sat temp −116.6 F)			Pressure 1.50 psia Gage pressure 26.87 in. v (Sat temp −112.1 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
(*at sat*)	(45.70)	(90.38)	(0.2765)	(41.42)	(90.66)	(0.2750)	(33.63)	(91.28)	(0.2718)	(28.38)	(91.82)	(0.26
−120	46.29	90.89	.2780	41.65	90.88	.2756						
−110	47.66	92.11	.2815	42.89	92.10	.2791	34.29	92.09	.2740	28.55	92.08	.269
−100	49.03	93.34	.2850	44.12	93.34	.2826	35.27	93.33	.2775	29.38	93.31	.273
− 90	50.41	94.59	.2884	45.35	94.59	.2860	36.26	94.58	.2809	30.20	94.56	.276
− 80	51.78	95.86	.2918	46.59	95.85	.2894	37.25	95.84	.2843	31.02	95.83	.280
− 70	53.15	97.13	.2951	47.82	97.13	.2927	38.24	97.12	.2876	31.85	97.11	.283
− 60	54.52	98.43	.2984	49.05	98.42	.2960	39.22	98.41	.2909	32.67	98.40	.286
− 50	55.89	99.73	.3016	50.29	99.73	.2992	40.21	99.72	.2941	33.49	99.71	.289
− 40	57.26	101.06	.3048	51.52	101.05	.3024	41.20	101.04	.2973	34.32	101.03	.293
− 30	58.63	102.39	.3079	52.75	102.39	.3055	42.19	102.38	.3004	35.14	102.37	.296
− 20	59.99	103.74	.3111	53.99	103.74	.3087	43.17	103.73	.3036	35.96	103.72	.299
− 10	61.36	105.11	.3141	55.22	105.11	.3117	44.16	105.10	.3066	36.79	105.09	.302
0	62.73	106.49	.3172	56.45	106.49	.3148	45.15	106.48	.3097	37.61	106.47	.305
10	64.10	107.89	.3202	57.68	107.88	.3178	46.13	107.87	.3127	38.43	107.86	.308
20	65.47	109.30	.3232	58.92	109.29	.3208	47.12	109.29	.3157	39.25	109.28	.311
30	66.84	110.72	.3261	60.15	110.72	.3237	48.10	110.71	.3186	40.08	110.70	.314
40	68.21	112.16	.3290	61.38	112.16	.3266	49.09	112.15	.3215	40.90	112.14	.317
50	69.58	113.62	.3319	62.61	113.62	.3295	50.08	113.61	.3244	41.72	113.60	.320
60	70.95	115.09	.3348	63.84	115.08	.3324	51.06	115.07	.3273	42.54	115.06	.323
70	72.31	116.57	.3376	65.08	116.57	.3352	52.05	116.56	.3301	43.36	116.55	.325
80	73.68	118.07	.3404	66.31	118.07	.3380	53.03	118.06	.3329	44.18	118.05	.328
90	75.05	119.59	.3432	67.54	119.58	.3408	54.02	119.58	.3357	45.01	119.57	.331
100	76.42	121.12	.3459	68.77	121.11	.3435	55.01	121.10	.3384	45.83	121.10	.334
110	77.79	122.66	.3487	70.00	122.66	.3463	55.99	122.65	.3412	46.65	122.64	.337
120	79.16	124.22	.3514	71.24	124.21	.3490	56.98	124.21	.3439	47.47	124.20	.339
130	80.53	125.79	.3541	72.47	125.79	.3517	57.96	125.78	.3466	48.29	125.77	.342
140	81.89	127.38	.3567	73.70	127.38	.3543	58.95	127.37	.3493	49.12	127.36	.345
150	83.26	128.99	.3594	74.93	128.99	.3570	59.93	128.97	.3519	49.94	128.97	.347
160	84.63	130.60	.3620	76.16	130.60	.3596	60.92	130.59	.3545	50.76	130.58	.350
170	86.00	132.23	.3646	77.39	132.23	.3622	61.90	132.22	.3571	51.58	132.22	.353
180	87.37	133.88	.3762	78.62	133.88	.3648	62.89	133.87	.3597	52.40	133.86	.355
190							63.88	135.53	.3623	53.22	135.53	.358

Table 19. F-22 ($CHClF_2$) Monochlorodifluoromethane— Properties of Superheated Vapor (Continued)

emp F	Pressure 1.75 psia Gage pressure 26.36 in. vac (Sat temp −108.2 F)			Pressure 2.0 psia Gage pressure 25.85 in. vac (Sat temp −104.7 F)			Pressure 2.5 psia Gage pressure 24.83 in. vac (Sat temp −98.8 F)			Pressure 3.0 psia Gage pressure 23.81 in. vac (Sat temp −93.8 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
t sat)	*(24.58)*	*(92.28)*	*(0.2670)*	*(27.71)*	*(92.72)*	*(0.2651)*	*(17.64)*	*(93.41)*	*(0.2621)*	*(14.89)*	*(94.02)*	*(0.2597)*
100	25.16	93.30	.2698	22.00	93.29	.2667						
90	25.87	94.55	.2732	22.62	94.54	.2702	18.08	94.51	.2650	15.05	94.49	.2609
80	26.58	95.82	.2766	23.24	95.81	.2735	18.57	95.78	.2684	15.46	95.76	.2642
70	27.28	97.10	.2799	23.86	97.08	.2768	19.07	97.06	.2717	15.87	97.04	.2676
60	27.99	98.39	.2832	24.48	98.38	.2801	19.56	98.36	.2750	16.29	98.34	.2708
50	28.70	99.70	.2864	25.10	99.69	.2833	20.06	99.67	.2782	16.70	99.65	.2741
40	29.40	101.02	.2896	25.72	101.01	.2865	20.55	100.99	.2814	17.11	100.97	.2773
30	30.11	102.36	.2927	26.33	102.35	.2897	21.05	102.33	.2846	17.53	102.31	.2804
20	30.81	103.71	.2959	26.95	103.70	.2928	21.54	103.68	.2877	17.94	103.66	.2835
10	31.52	105.08	.2990	27.57	105.07	.2959	22.04	105.05	.2908	18.35	105.03	.2866
0	32.22	106.46	.3020	28.19	106.45	.2990	22.53	106.43	.2939	18.76	106.41	.2897
10	32.93	107.85	.3050	28.80	107.85	.3020	23.03	107.83	.2969	19.18	107.81	.2927
20	33.63	109.27	.3080	29.42	109.26	.3049	23.52	109.24	.2998	19.59	109.22	.2957
30	34.34	110.69	.3109	30.04	110.68	.3079	24.02	110.67	.3028	20.00	110.65	.2986
40	35.04	112.13	.3139	30.66	112.13	.3108	24.51	112.11	.3057	20.41	112.09	.3015
50	35.75	113.59	.3167	31.27	113.58	.3137	25.00	113.57	.3086	20.83	113.55	.3044
60	36.45	115.06	.3196	31.89	115.05	.3165	25.50	115.04	.3114	21.24	115.02	.3073
70	37.16	116.55	.3224	32.51	116.54	.3194	25.99	116.52	.3143	21.65	116.51	.3101
80	37.86	118.05	.3252	33.12	118.04	.3222	26.49	118.02	.3171	22.06	118.01	.3129
90	38.57	119.56	.3280	33.74	119.55	.3250	26.98	119.54	.3199	22.47	119.52	.3157
100	39.27	121.09	.3308	34.36	121.08	.3277	27.47	121.07	.3226	22.88	121.05	.3185
110	39.98	122.63	.3335	34.97	122.63	.3305	27.97	122.61	.3254	23.30	122.60	.3212
120	40.68	124.19	.3362	35.59	124.19	.3332	28.46	124.17	.3281	23.71	124.16	.3239
130	41.39	125.77	.3389	36.21	125.76	.3359	28.95	125.75	.3308	24.12	125.73	.3266
140	42.09	127.36	.3416	36.82	127.35	.3385	29.45	127.34	.3334	24.53	127.32	.3293
150	42.80	128.96	.3442	37.44	128.95	.3412	29.94	128.94	.3361	24.94	128.93	.3319
160	43.50	130.58	.3469	38.06	130.57	.3438	30.44	130.56	.3387	25.35	130.54	.3346
170	44.20	132.21	.3495	38.67	132.20	.3464	30.93	132.19	.3413	25.77	132.18	.3372
180	44.91	133.86	.3521	39.29	133.85	.3490	31.42	133.84	.3439	26.18	133.83	.3398
190	45.61	135.52	.3547	39.91	135.51	.3516	31.91	135.50	.3465	26.59	135.49	.3424
200	46.32	137.20	.3572	40.52	137.19	.3542	32.41	137.18	.3491	27.10	137.17	.3449
210							32.90	138.87	.3516	27.41	138.86	.3475

emp F	Pressure 3.5 psia Gage pressure 22.80 in. vac (Sat temp −89.4 F)			Pressure 4.0 psia Gage pressure 21.78 in. vac (Sat temp −85.4 F)			Pressure 4.5 psia Gage pressure 20.76 in. vac (Sat temp −82.0 F)			Pressure 5.0 psia Gage pressure 19.74 in. vac (Sat temp −78.7 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
t sat)	*(12.90)*	*(94.55)*	*(0.2576)*	*(11.40)*	*(95.03)*	*(0.2558)*	*(10.22)*	*(95.46)*	*(0.2543)*	*(9.266)*	*(95.83)*	*(0.2529)*
80	13.14	95.74	.2607	11.57	95.72	.2576	10.27	95.69	.2549			
70	13.59	97.02	.2640	11.88	97.00	.2609	10.55	96.97	.2582	9.483	96.95	.2558
60	13.95	98.31	.2673	12.19	98.29	.2642	10.83	98.27	.2615	9.733	98.25	.2591
50	14.30	99.62	.2705	12.50	99.60	.2674	11.10	99.58	.2647	9.983	99.56	.2623
40	14.66	100.95	.2737	12.81	100.93	.2706	11.38	100.91	.2679	10.232	100.89	.2655
30	15.01	102.29	.2769	13.12	102.27	.2738	11.66	102.25	.2711	10.481	102.23	.2687
20	15.37	103.64	.2800	13.43	103.62	.2769	11.93	103.60	.2742	10.729	103.58	.2718
10	15.72	105.01	.2831	13.74	104.99	.2800	12.21	104.98	.2773	10.978	104.96	.2749
0	16.07	106.40	.2861	14.05	106.38	.2831	12.48	106.36	.2804	11.226	106.34	.2780
10	16.43	107.80	.2892	14.36	107.78	.2861	12.76	107.76	.2834	11.475	107.74	.2810
20	16.78	109.21	.2921	14.67	109.19	.2891	13.04	109.17	.2864	11.723	109.16	.2840
30	17.13	110.63	.2951	14.98	110.62	.2920	13.31	110.60	.2893	11.972	110.59	.2869
40	17.49	112.07	.2980	15.29	112.06	.2950	13.59	112.05	.2922	12.220	112.03	.2898
50	17.84	113.53	.3009	15.60	113.52	.2978	13.86	113.50	.2951	12.468	113.49	.2927
60	18.19	115.00	.3037	15.91	114.99	.3007	14.14	114.98	.2980	12.715	114.96	.2956
70	18.55	116.49	.3066	16.22	116.48	.3035	14.41	116.46	.3008	12.963	116.45	.2984
80	18.90	117.99	.3094	16.53	117.98	.3063	14.69	117.97	.3036	13.211	117.95	.3012
90	19.25	119.51	.3122	16.84	119.50	.3091	14.96	119.48	.3064	13.460	119.47	.3040
100	19.61	121.04	.3149	17.15	121.03	.3119	15.24	121.01	.3092	13.707	121.00	.3068
110	19.96	122.58	.3177	17.46	122.57	.3146	15.51	122.56	.3119	13.955	122.55	.3095
120	20.31	124.14	.3204	17.77	124.13	.3173	15.79	124.12	.3146	14.202	124.11	.3122
130	20.67	125.72	.3231	18.08	125.70	.3200	16.06	125.69	.3173	14.450	125.68	.3149
140	21.02	127.31	.3258	18.39	127.29	.3227	16.34	127.28	.3200	14.697	127.27	.3176
150	21.37	128.91	.3284	18.69	128.90	.3254	16.61	128.89	.3227	14.945	128.88	.3202
160	21.73	130.53	.3310	19.00	130.52	.3280	16.89	130.51	.3253	15.192	130.49	.3229
170	22.08	132.17	.3337	19.31	132.15	.3306	17.16	132.14	.3279	15.440	132.13	.3255
180	22.43	133.81	.3363	19.62	133.80	.3332	17.44	133.79	.3305	15.687	133.78	.3281
190	22.78	135.48	.3388	19.93	135.46	.3358	17.71	135.45	.3331	15.934	135.44	.3307
200	23.14	137.15	.3414	20.24	137.14	.3383	17.98	137.13	.3357	16.181	137.12	.3332
210	23.49	138.85	.3439	20.55	138.84	.3409	18.26	138.82	.3382	16.429	138.81	.3358
220	23.84	140.56	.3465	20.86	140.54	.3434	18.53	140.53	.3407	16.676	140.52	.3383
230										16.923	142.24	.3408

Table 19. F-22 ($CHClF_2$) Monochlorodifluoromethane— Properties of Superheated Vapor (Continued)

Temp F	Pressure 6 psia Gage pressure 17.71 in. vac (Sat temp −73.0 F)			Pressure 7 psia Gage pressure 15.67 in. vac (Sat temp −68.0 F)			Pressure 8 psia Gage pressure 13.64 in. vac (Sat temp −63.5 F)			Pressure 9 psia Gage pressure 11.60 in. vac (Sat temp −59.5 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
(*at sat*)	(*7.823*)	(*96.52*)	(*0.2506*)	(*6.781*)	(*97.12*)	(*0.2487*)	(*5.991*)	(*97.66*)	(*0.2471*)	(*5.370*)	(*98.14*)	(*0.2456*)
− 70	7.886	96.91	.2516									
− 60	8.095	98.20	.2549	6.924	98.16	.2513	6.046	98.12	.2482			
− 50	8.303	99.52	.2581	7.103	99.48	.2545	6.204	99.44	.2514	5.503	99.39	.2487
− 40	8.511	100.85	.2613	7.282	100.80	.2577	6.361	100.76	.2546	5.643	100.72	.2519
− 30	8.719	102.19	.2645	7.461	102.15	.2609	6.517	102.10	.2578	5.783	102.06	.2551
− 20	8.927	103.54	.2676	7.640	103.51	.2640	6.673	103.47	.2610	5.923	103.43	.2582
− 10	9.135	104.92	.2707	7.818	104.88	.2671	6.830	104.84	.2641	6.062	104.81	.2613
0	9.342	106.31	.2738	7.996	106.27	.2702	6.986	106.23	.2671	6.201	106.19	.2644
10	9.549	107.71	.2768	8.174	107.67	.2732	7.143	107.63	.2701	6.341	107.60	.2674
20	9.757	109.12	.2798	8.352	109.09	.2762	7.299	109.05	.2731	6.479	109.02	.2704
30	9.964	110.55	.2827	8.530	110.52	.2792	7.455	110.48	.2761	6.618	110.45	.2734
40	10.171	112.00	.2856	8.708	111.96	.2821	7.611	111.93	.2790	6.757	111.90	.2763
50	10.378	113.46	.2885	8.886	113.42	.2850	7.767	113.39	.2819	6.896	113.36	.2792
60	10.585	114.93	.2914	9.064	114.90	.2878	7.922	114.87	.2848	7.035	114.83	.2820
70	10.792	116.42	.2942	9.241	116.39	.2907	8.078	116.36	.2876	7.173	116.32	.2849
80	10.999	117.92	.2970	9.419	117.89	.2935	8.234	117.86	.2904	7.312	117.83	.2877
90	11.206	119.44	.2998	9.597	119.41	.2963	8.389	119.38	.2932	7.450	119.35	.2905
100	11.413	120.97	.3026	9.774	120.94	.2990	8.545	120.91	.2960	7.589	120.88	.2933
110	11.620	122.52	.3053	9.952	122.49	.3018	8.700	122.46	.2987	7.727	122.43	.2960
120	11.826	124.08	.3080	10.129	124.05	.3045	8.856	124.02	.3014	7.865	124.00	.2987
130	12.033	125.66	.3107	10.306	125.63	.3072	9.011	125.60	.3041	8.004	125.57	.3014
140	12.239	127.25	.3134	10.483	127.22	.3099	9.166	127.19	.3068	8.142	127.17	.3041
150	12.446	128.85	.3161	10.660	128.82	.3125	9.321	128.80	.3095	8.280	128.77	.3068
160	12.652	130.47	.3187	10.837	130.45	.3152	9.477	130.42	.3121	8.418	130.39	.3094
170	12.859	132.11	.3213	11.014	132.08	.3178	9.632	132.06	.3147	8.556	132.03	.3120
180	13.065	133.75	.3239	11.192	133.73	.3204	9.787	133.71	.3173	8.694	133.68	.3146
190	13.271	135.41	.3265	11.369	135.40	.3230	9.942	135.37	.3199	8.832	135.35	.3172
200	13.477	137.09	.3291	11.545	137.08	.3255	10.097	137.05	.3225	8.970	137.03	.3198
210	13.683	138.79	.3316	11.722	138.77	.3281	10.252	138.75	.3250	9.108	138.72	.3223
220	13.889	140.50	.3341	11.899	140.47	.3306	10.407	140.45	.3276	9.245	140.43	.3249
230	14.096	142.22	.3366	12.076	142.20	.3331	10.562	142.18	.3301	9.383	142.15	.3274
240				12.253	143.94	.3356	10.717	143.91	.3326	9.521	143.89	.3299
250										9.659	145.65	.3324

Temp F	Pressure 10 psia Gage pressure 9.57 in. vac (Sat temp −55.8 F)			Pressure 11 psia Gage pressure 7.53 in. vac (Sat temp −52.4 F)			Pressure 12 psia Gage pressure 5.50 in. vac (Sat temp −49.2 F)			Pressure 14 psia Gage pressure 1.42 in. vac (Sat temp −43.5 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
(*at sat*)	(*4.870*)	(*98.58*)	(*0.2444*)	(*4.458*)	(*98.99*)	(*0.2432*)	(*4.112*)	(*99.37*)	(*0.2422*)	(*3.562*)	(*100.05*)	(*0.2404*)
− 50	4.943	99.35	.2462	4.485	99.31	.2440						
− 40	5.069	100.68	.2494	4.600	100.64	.2472	4.209	100.60	.2452	3.595	100.51	.2415
− 30	5.195	102.02	.2526	4.715	101.98	.2504	4.315	101.94	.2484	3.685	101.86	.2447
− 20	5.321	103.29	.2558	4.830	103.35	.2535	4.420	103.31	.2515	3.776	103.23	.2479
− 10	5.447	104.77	.2589	4.944	104.73	.2567	4.525	104.69	.2546	3.867	104.62	.2510
0	5.573	106.16	.2619	5.059	106.12	.2597	4.630	106.08	.2577	3.957	106.01	.2541
10	5.699	107.56	.2650	5.173	107.53	.2628	4.735	107.49	.2607	4.048	107.42	.2571
20	5.824	108.98	.2680	5.288	108.95	.2658	4.840	108.91	.2637	4.138	108.84	.2601
30	5.950	110.40	.2709	5.402	110.38	.2687	4.945	110.35	.2667	4.228	110.28	.2631
40	6.075	111.86	.2739	5.516	111.83	.2716	5.050	111.79	.2696	4.318	111.73	.2660
50	6.200	113.32	.2767	5.630	113.99	.2745	5.155	113.26	.2725	4.408	113.19	.2689
60	6.325	114.80	.2796	5.743	114.77	.2774	5.260	114.74	.2754	4.499	114.67	.2718
70	6.450	116.29	.2824	5.857	116.26	.2802	5.364	116.23	.2782	4.588	116.17	.2746
80	6.575	117.80	.2853	5.971	117.77	.2831	5.468	117.74	.2810	4.678	117.68	.2775
90	6.700	119.32	.2881	6.085	119.29	.2859	5.573	119.26	.2838	4.768	119.20	.2803
100	6.834	120.85	.2908	6.199	120.82	.2886	5.677	120.80	.2866	4.857	120.74	.2830
110	6.949	122.40	.2936	6.312	122.37	.2914	5.781	122.35	.2894	4.947	122.29	.2858
120	7.073	123.97	.2963	6.425	123.94	.2941	5.885	123.91	.2921	5.036	123.86	.2885
130	7.198	125.55	.2990	6.539	125.59	.2968	5.989	125.49	.2948	5.126	125.44	.2912
140	7.322	127.14	.3017	6.652	127.11	.2995	6.093	127.08	.2975	5.215	127.03	.2939
150	7.447	128.74	.3043	6.766	128.72	.3021	6.197	128.69	.3001	5.304	128.64	.2966
160	7.571	130.37	.3070	6.879	130.34	.3048	6.301	130.32	.3028	5.394	130.27	.2992
170	7.696	132.01	.3096	6.992	131.98	.3074	6.405	131.96	.3054	5.483	131.91	.3018
180	7.820	133.66	.3122	7.105	133.63	.3100	6.509	133.61	.3080	5.573	133.56	.3044
190	7.945	135.32	.3148	7.218	135.30	.3126	6.613	135.27	.3106	5.661	135.23	.3070
200	8.069	137.00	.3173	7.331	136.98	.3152	6.717	136.96	.3132	5.750	136.91	.3096
210	8.193	138.70	.3199	7.444	138.68	.3177	6.820	138.65	.3157	5.840	138.61	.3121
220	8.317	140.41	.3224	7.557	140.39	.3202	6.924	140.36	.3182	5.929	140.32	.3147
230	8.441	142.13	.3250	7.670	142.11	.3228	7.028	142.09	.3208	6.018	142.05	.3173
240	7.565	143.87	.3275	7.783	143.85	.3253	7.131	143.83	.3233	6.107	143.79	.3197
250	8.689	145.63	.3300	7.896	145.60	.3278	7.234	145.58	.3258	6.196	145.54	.3222
260							7.338	147.35	.3282	6.285	147.31	.3247

Table 19. F-22 ($CHClF_2$) Monochlorodifluoromethane— Properties of Superheated Vapor (Continued)

ap	Pressure 16 psia Gage pressure 1.3 psig (Sat temp −38.3 F)			Pressure 18 psia Gage pressure 3.3 psig (Sat temp −33.7 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp −29.4 F)			Pressure 25 psia Gage pressure 10.3 psig (Sat temp −20.0 F)		
	v	h	s	v	h	s	v	h	s	v	h	s
at)	(3.147)	(100.63)	(0.2390)	(2.820)	(101.19)	(0.2376)	(2.556)	(101.70)	(0.2365)	(2.074)	(102.79)	(0.2341)
0	3.213	101.78	.2416	2.846	101.70	.2388						
0	3.293	103.16	.2447	2.917	103.08	.2420	2.616	103.00	.2394			
0	3.373	104.54	.2479	2.988	104.46	.2451	2.681	104.38	.2426	2.127	104.19	.2372
0	3.452	105.93	.2509	3.059	105.86	.2482	2.745	105.78	.2457	2.179	105.59	.2403
0	3.532	107.34	.2540	3.130	107.27	.2512	2.809	107.20	.2487	2.231	107.01	.2434
0	3.611	108.77	.2570	3.201	108.70	.2542	2.873	108.63	.2517	2.283	108.45	.2464
0	3.690	110.21	.2600	3.272	110.14	.2572	2.937	110.07	.2547	2.335	109.89	.2494
0	3.769	111.66	.2629	3.343	111.59	.2601	3.001	111.52	.2577	2.386	111.35	.2524
0	3.848	113.13	.2658	3.413	113.06	.2630	3.065	113.00	.2606	2.437	112.83	.2553
0	3.928	114.61	.2687	3.484	114.55	.2659	3.128	114.48	.2634	2.489	114.32	.2582
0	4.006	116.11	.2715	3.554	116.04	.2688	3.192	115.98	.2663	2.540	115.82	.2610
0	4.085	117.62	.2744	3.624	117.56	.2716	3.255	117.49	.2691	2.591	117.34	.2639
0	4.164	119.14	.2772	3.694	119.08	.2744	3.319	119.02	.2719	2.643	118.87	.2667
0	4.243	120.68	.2799	3.764	120.62	.2772	3.382	120.56	.2747	2.693	120.42	.2695
0	4.321	122.23	.2827	3.834	122.18	.2799	3.445	122.12	.2775	2.744	121.98	.2722
0	4.399	123.80	.2854	3.904	123.74	.2827	3.508	123.69	.2802	2.795	123.55	.2750
0	4.478	125.38	.2881	3.975	125.33	.2854	3.571	125.27	.2829	2.846	125.14	.2777
0	4.557	126.98	.2908	4.044	126.92	.2881	3.635	126.87	.2856	2.897	126.74	.2804
0	4.635	128.59	.2935	4.114	128.54	.2907	3.698	128.49	.2883	2.947	128.36	.2831
0	4.713	130.21	.2961	4.184	130.16	.2934	3.761	130.11	.2909	2.998	129.99	.2857
0	4.792	131.86	.2987	4.254	131.81	.2960	3.824	131.76	.2936	3.049	131.63	.2883
0	4.870	133.51	.3014	4.324	133.46	.2986	3.886	133.41	.2962	3.099	133.29	.2910
0	4.948	135.18	.3039	4.393	135.13	.3012	3.949	135.08	.2988	3.150	134.96	.2936
0	5.026	136.86	.3065	4.463	136.82	.3038	4.012	136.77	.3013	3.200	136.65	.2961
0	5.104	138.56	.3091	4.532	138.52	.3063	4.074	138.47	.3039	3.251	138.36	.2987
0	5.182	140.27	.3116	4.602	140.32	.3089	4.137	140.19	.3064	3.301	140.07	.3012
0	5.260	142.00	.3141	4.671	141.96	.3114	4.200	141.91	.3090	3.352	141.80	.3038
0	5.338	143.74	.3166	4.740	143.70	.3139	4.263	143.66	.3115	3.402	143.55	.3063
0	5.416	145.50	.3191	4.810	145.46	.3164	4.325	145.42	.3140	3.452	145.31	.3088
0	5.494	147.27	.3216	4.880	147.23	.3189	4.388	147.19	.3165	3.503	147.08	.3113
0	5.572	149.05	.3241	4.949	149.02	.3214	4.450	148.98	.3189	3.553	148.88	.3138
0							4.512	150.78	.3214	3.603	150.68	.3162
0										3.653	152.50	.3187

np	Pressure 30 psia Gage pressure 15.3 psig (Sat temp −11.9 F)			Pressure 35 psia Gage pressure 20.3 psig (Sat temp −4.9 F)			Pressure 40 psia Gage pressure 25.3 psig (Sat temp −1.5 F)			Pressure 45 psia Gage pressure 30.3 psig (Sat temp −7.2 F)		
	v	h	s	v	h	s	v	h	s	v	h	s
sat)	(1.749)	(103.71)	(0.2322)	(1.513)	(104.49)	(0.2306)	(1.334)	(105.17)	(0.2293)	(1.193)	(105.81)	(0.2281)
10	1.758	103.98	.2328									
0	1.802	105.39	.2359	1.532	105.19	.2321						
10	1.845	106.82	.2390	1.570	106.62	.2352	1.363	106.41	.2319	1.202	106.22	.2290
20	1.889	108.26	.2420	1.608	108.07	.2383	1.396	107.87	.2350	1.232	107.68	.2321
30	1.933	109.71	.2450	1.645	109.53	.2413	1.430	109.34	.2380	1.262	109.16	.2351
40	1.976	111.18	.2480	1.683	111.00	.2443	1.463	110.82	.2410	1.292	110.65	.2381
50	2.019	112.66	.2509	1.720	112.49	.2472	1.496	112.31	.2439	1.322	112.15	.2411
60	2.063	114.15	.2538	1.758	113.99	.2501	1.529	113.81	.2469	1.351	113.64	.2440
70	2.106	115.66	.2567	1.795	115.50	.2530	1.562	115.33	.2498	1.381	115.16	.2469
80	2.149	117.18	.2595	1.832	117.02	.2559	1.595	116.86	.2526	1.410	116.71	.2498
90	2.192	118.72	.2624	1.869	118.56	.2587	1.628	118.40	.2555	1.440	118.26	.2526
00	2.235	120.27	.2652	1.906	120.12	.2615	1.660	119.96	.2583	1.469	119.81	.2554
10	2.277	121.83	.2679	1.943	121.68	.2643	1.693	121.53	.2610	1.498	121.38	.2582
20	2.320	123.41	.2707	1.980	123.26	.2670	1.726	123.12	.2638	1.527	122.97	.2610
30	2.362	125.00	.2734	2.017	124.86	.2697	1.758	124.72	.2665	1.556	124.57	.2637
40	2.405	126.61	.2761	2.054	126.47	.2724	1.790	126.33	.2693	1.585	126.19	.2664
50	2.448	128.22	.2788	2.091	128.09	.2751	1.823	127.96	.2719	1.614	127.82	.2691
60	2.490	129.86	.2814	2.127	129.73	.2778	1.855	129.60	.2746	1.643	129.47	.2718
70	2.533	131.51	.2841	2.164	131.38	.2804	1.887	131.25	.2773	1.672	131.12	.2745
80	2.575	133.17	.2867	2.200	133.04	.2830	1.919	132.92	.2799	1.701	132.79	.2771
90	2.617	134.84	.2893	2.237	134.72	.2857	1.951	134.60	.2825	1.730	134.48	.2797
00	2.659	136.54	.2919	2.273	136.42	.2882	1.983	136.30	.2851	1.758	136.18	.2823
10	2.702	138.24	.2944	2.310	138.12	.2908	2.016	138.01	.2877	1.787	137.89	.2849
20	2.744	139.96	.2970	2.346	139.83	.2934	2.047	139.73	.2902	1.815	139.62	.2875
30	2.786	141.69	.2995	2.382	141.58	.2959	2.079	141.47	.2928	1.844	141.35	.2900
40	2.829	143.44	.3020	2.419	143.33	.2984	2.111	143.22	.2953	1.872	143.11	.2925
50	2.871	145.20	.3045	2.455	145.10	.3009	2.143	144.99	.2978	1.902	144.87	.2950
60	2.913	146.98	.3070	2.491	146.88	.3034	2.175	146.77	.3003	1.931	146.66	.2975
70	2.955	148.77	.3095	2.527	148.67	.3059	2.207	148.57	.3028	1.959	148.46	.3000
80	2.997	150.58	.3120	2.563	150.48	.3084	2.239	150.38	.3052	1.987	150.28	.3025
90	3.039	152.40	.3144	2.600	152.30	.3108	2.270	152.20	.3077	2.016	152.11	.3049
00				2.636	154.14	.3132	2.302	154.04	.3101	2.044	153.94	.3074
10							2.334	155.89	.3125	2.072	155.80	.3098

Table 19. F-22 ($CHClF_2$) Monochlorodifluoromethane— Properties of Superheated Vapor (Continued)

Temp F	Pressure 50 psia Gage pressure 35.3 psig (Sat temp 12.4 F)			Pressure 55 psia Gage pressure 40.3 psig (Sat temp 17.2 F)			Pressure 60 psia Gage pressure 45.3 psig (Sat temp 21.8 F)			Pressure 65 psia Gage pressure 50.3 psig (Sat temp 26.0 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*1.080*)	(*106.36*)	(*0.2271*)	(*.9851*)	(*106.85*)	(*0.2261*)	(*.9061*)	(*107.32*)	(*0.2253*)	(*.8398*)	(*107.73*)	(*0.224*
20	1.100	107.48	.2294	.9922	107.27	.2270						
30	1.128	108.96	.2325	1.0176	108.76	.2300	.9255	108.56	.2278	.8480	108.36	.225
40	1.155	110.46	.2355	1.0425	110.27	.2331	.9491	110.07	.2308	.8700	109.88	.228
50	1.182	111.96	.2384	1.0675	111.78	.2360	.9721	111.59	.2338	.8917	111.39	.231
60	1.209	113.47	.2414	1.0921	113.28	.2390	.9950	113.10	.2368	.9130	112.91	.234
70	1.236	114.99	.2443	1.1168	114.82	.2419	1.0180	114.64	.2397	.9343	114.46	.237
80	1.262	116.54	.2472	1.1414	116.37	.2448	1.0406	116.20	.2426	.9557	116.02	.240
90	1.289	118.10	.2500	1.1659	117.93	.2477	1.0634	117.77	.2455	.9763	117.60	.243
100	1.316	119.66	.2528	1.1902	119.50	.2505	1.0858	119.35	.2483	.9973	119.18	.246
110	1.342	121.23	.2556	1.2146	121.08	.2533	1.1082	120.92	.2511	1.0183	120.77	.249
120	1.369	122.83	.2584	1.2386	122.68	.2561	1.1305	122.53	.2539	1.0390	122.38	.251
130	1.395	124.43	.2612	1.2627	124.29	.2588	1.1529	124.14	.2567	1.0598	124.00	.254
140	1.421	126.05	.2639	1.2870	125.91	.2616	1.1751	125.77	.2594	1.0804	125.63	.257
150	1.448	127.68	.2666	1.3110	127.54	.2643	1.1974	127.41	.2621	1.1010	127.27	.260
160	1.474	129.33	.2693	1.3349	129.20	.2670	1.2193	129.06	.2648	1.1217	128.93	.262
170	1.500	130.99	.2719	1.3590	130.86	.2696	1.2415	130.73	.2675	1.1421	130.60	.265
180	1.526	132.67	.2764	1.3827	132.54	.2723	1.2634	132.41	.2702	1.1627	132.28	.268
190	1.552	134.35	.2772	1.4065	134.23	.2749	1.2853	134.10	.2728	1.1829	133.98	.270
200	1.578	136.05	.2798	1.4301	135.93	.2775	1.3071	135.81	.2754	1.2031	135.68	.273
210	1.604	137.77	.2824	1.4539	137.65	.2801	1.3290	137.53	.2780	1.2234	137.41	.276
220	1.630	139.50	.2849	1.4774	139.38	.2827	1.3509	139.26	.2805	1.2437	139.15	.278
230	1.655	141.24	.2875	1.5011	141.12	.2852	1.3726	141.01	.2831	1.2639	140.89	.281
240	1.681	143.00	.2900	1.5246	142.88	.2877	1.3943	142.77	.2856	1.2841	142.66	.283
250	1.707	144.76	.2925	1.5481	144.66	.2902	1.4159	144.55	.2882	1.3041	144.43	.286
260	1.732	146.55	.2950	1.5716	146.44	.2927	1.4375	146.33	.2907	1.3240	146.23	.288
270	1.758	148.35	.2975	1.5950	148.25	.2952	1.4591	148.14	.2932	1.3441	148.04	.291
280	1.784	150.18	.3000	1.6184	150.07	.2977	1.4806	149.97	.2956	1.3641	149.87	.293
290	1.810	152.01	.3024	1.6417	151.91	.3002	1.5021	151.81	.2981	1.3840	151.71	.296
300	1.835	153.85	.3049	1.6651	153.75	.3026	1.5235	153.65	.3005	1.4038	153.55	.298
310	1.861	155.70	.3073	1.6883	155.61	.3051	1.5451	155.51	.3030	1.4237	155.41	.301
320	1.886	157.58	.3097	1.7116	157.48	.3075	1.5665	157.39	.3054	1.4436	157.29	.303
330							1.5879	159.27	.3078	1.4634	159.18	.305

Temp F	Pressure 70 psia Gage pressure 55.3 psig (Sat temp 30.0 F)			Pressure 75 psia Gage pressure 60.3 psig (Sat temp 33.8 F)			Pressure 80 psia Gage pressure 65.3 psig (Sat temp 37.4 F)			Pressure 90 psia Gage pressure 75.3 psig (Sat temp 44.2 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*.7816*)	(*108.13*)	(*0.2235*)	(*.7301*)	(*108.31*)	(*0.2231*)	(*.6856*)	(*108.86*)	(*0.2224*)	(*.6109*)	(*109.48*)	(*0.221
40	.8023	109.67	.2268	.7421	109.47	.2249	.6904	109.26	.2232			
50	.8224	111.19	.2298	.7615	111.00	.2280	.7087	110.80	.2263	.6207	110.39	.223
60	.8424	112.71	.2328	.7807	112.52	.2310	.7269	112.32	.2293	.6372	111.92	.226
70	.8625	114.27	.2358	.7995	114.09	.2340	.7446	113.90	.2323	.6535	113.51	.229
80	.8823	115.84	.2387	.8182	115.67	.2369	.7625	115.48	.2352	.6697	115.12	.232
90	.9019	117.43	.2416	.8368	117.26	.2398	.7801	117.09	.2381	.6853	116.73	.235
100	.9216	119.02	.2444	.8553	118.86	.2427	.7978	118.69	.2410	.7018	118.35	.237
110	.9411	120.61	.2473	.8739	120.45	.2455	.8151	120.29	.2439	.7175	119.96	.240
120	.9605	121.23	.2501	.8923	122.07	.2483	.8327	121.91	.2467	.7334	121.59	.243
130	.9800	123.85	.2529	.9107	123.70	.2511	.8500	123.55	.2495	.7489	123.23	.246
140	.9991	125.48	.2556	.9289	125.34	.2539	.8673	125.19	.2522	.7645	124.89	.249
150	1.0187	127.12	.2583	.9470	126.99	.2566	.8844	126.84	.2550	.7800	126.55	.252
160	1.0379	128.79	.2610	.9650	128.65	.2593	.9014	128.51	.2577	.7954	128.23	.254
170	1.0569	130.47	.2637	.9831	130.33	.2620	.9184	130.19	.2604	.8108	129.92	.257
180	1.0762	132.15	.2664	1.0012	132.02	.2647	.9356	131.89	.2630	.8262	131.62	.260
190	1.0951	133.85	.2690	1.0188	133.72	.2673	.9524	133.59	.2657	.8411	133.34	.262
200	1.1139	135.56	.2716	1.0366	135.44	.2699	.9689	135.31	.2683	.8561	135.06	.265
210	1.1330	137.29	.2742	1.0544	137.17	.2725	.9857	137.04	.2709	.8711	136.80	.268
220	1.1519	139.03	.2768	1.0722	138.91	.2751	1.0025	138.79	.2735	.8863	138.55	.270
230	1.1707	140.78	.2794	1.0898	140.66	.2777	1.0192	140.54	.2761	.9013	140.31	.273
240	1.1894	142.54	.2819	1.1075	142.43	.2802	1.0358	142.32	.2787	.9163	142.08	.275
250	1.2082	144.32	.2844	1.1250	144.21	.2828	1.0524	144.10	.2812	.9312	143.87	.278
260	1.2268	146.12	.2870	1.1426	146.01	.2853	1.0690	145.90	.2837	.9460	145.67	.280
270	1.2456	147.93	.2895	1.1601	147.82	.2878	1.0854	147.72	.2862	.9608	147.50	.283
280	1.2642	149.76	.2919	1.1776	149.66	.2903	1.1019	149.55	.2887	.9756	149.34	.285
290	1.2829	151.60	.2944	1.1949	151.50	.2927	1.1183	151.40	.2912	.9902	151.19	.288
300	1.3013	153.45	.2969	1.2122	153.35	.2952	1.1343	153.25	.2936	1.0048	153.05	.290
310	1.3198	155.31	.2993	1.2296	155.22	.2976	1.1507	155.12	.2961	1.0194	154.92	.293
320	1.3384	157.20	.3017	1.2481	157.10	.3001	1.1671	157.00	.2985	1.0341	156.81	.295
330	1.3569	159.09	.3041	1.2655	158.99	.3025	1.1834	158.90	.3009	1.0487	158.71	.298
340	1.3753	160.99	.3066	1.2829	160.90	.3049	1.1996	160.80	.3033	1.0633	160.62	.300
350										1.0779	162.54	.303

Table 19. F-22 ($CHClF_2$) Monochlorodifluoromethane— Properties of Superheated Vapor (Continued)

mp	Pressure 100 psia Gage pressure 85.3 psig (Sat temp 50.4 F)			Pressure 110 psia Gage pressure 95.3 psig (Sat temp 56.1 F)			Pressure 120 psia Gage pressure 105.3 psig (Sat temp 61.5 F)			Pressure 130 psia Gage pressure 115.3 psig (Sat temp 66.5 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
at)	(.5505)	(110.03)	(0.2202)	(.5005)	(110.50)	(0.2192)	(.4586)	(110.91)	(0.2183)	(.4228)	(111.27)	(0.2178)
0	.5651	111.51	.2232	.5061	111.10	.2204						
0	.5802	113.11	.2262	.5202	112.70	.2235	.4698	112.28	.2210	.4272	111.84	.2187
0	.5953	114.74	.2292	.5343	114.35	.2266	.4833	113.95	.2241	.4398	113.53	.2217
0	.6101	116.37	.2322	.5480	115.99	.2296	.4961	115.60	.2271	.4520	115.20	.2248
0	.6249	118.00	.2351	.5617	117.64	.2325	.5089	117.27	.2301	.4641	116.88	.2278
0	.6394	119.62	.2380	.5753	119.27	.2354	.5215	118.91	.2330	.4761	118.54	.2307
0	.6539	121.26	.2408	.5886	120.93	.2383	.5343	120.58	.2359	.4879	120.23	.2337
0	.6682	122.92	.2437	.6020	122.59	.2411	.5466	122.26	.2388	.4996	121.92	.2365
0	.6824	124.58	.2465	.6151	124.27	.2440	.5589	123.94	.2416	.5112	123.61	.2394
0	.6965	126.25	.2492	.6281	125.94	.2467	.5711	125.63	.2444	.5227	125.31	.2422
0	.7107	127.94	.2520	.6412	127.64	.2495	.5832	127.34	.2472	.5341	127.03	.2450
0	.7246	129.64	.2547	.6541	129.35	.2522	.5952	129.06	.2499	.5454	128.76	.2478
0	.7386	131.35	.2574	.6669	131.07	.2549	.6072	130.79	.2526	.5566	130.50	.2505
0	.7523	133.07	.2601	.6796	132.80	.2576	.6190	132.53	.2553	.5677	132.25	.2532
0	.7660	134.80	.2627	.6922	134.54	.2603	.6307	134.27	.2580	.5786	134.01	.2559
0	.7796	136.55	.2654	.7048	136.29	.2629	.6424	136.03	.2607	.5895	135.77	.2586
0	.7933	138.30	.2680	.7174	138.05	.2655	.6541	137.80	.2633	.6005	137.55	.2612
0	.8071	140.07	.2706	.7299	139.82	.2681	.6657	139.58	.2659	.6114	139.33	.2638
0	.8206	141.85	.2731	.7424	141.61	.2707	.6772	141.37	.2685	.6221	141.13	.2664
0	.8343	143.64	.2757	.7550	143.40	.2733	.6888	143.17	.2711	.6330	142.94	.2690
0	.8477	145.45	.2782	.7673	145.22	.2758	.7003	144.99	.2736	.6437	144.76	.2716
0	.8612	147.28	.2807	.7796	147.06	.2784	.7118	146.84	.2762	.6543	146.61	.2741
0	.8746	149.13	.2832	.7919	148.91	.2809	.7231	148.69	.2787	.6649	148.84	.2766
0	.8879	150.98	.2857	.8041	150.77	.2834	.7344	150.59	.2812	.6754	150.34	.2791
0	.9011	152.84	.2882	.8162	152.64	.2858	.7456	152.43	.2837	.6859	152.22	.2816
0	.9144	154.72	.2906	.8284	154.52	.2883	.7569	154.32	.2861	.6965	154.12	.2841
0	.9277	156.62	.2931	.8406	156.42	.2907	.7681	156.22	.2886	.7069	156.03	.2866
0	.9410	158.52	.2955	.8529	158.33	.2932	.7793	158.14	.2910	.7174	157.94	.2890
0	.9541	160.43	.2979	.8650	160.25	.2956	.7906	160.06	.2934	.7277	159.87	.2914
0	.9674	162.36	.3003	.8771	162.18	.2980	.8018	162.00	.2959	.7381	161.81	.2939
0	.9806	164.31	.3027	.8891	164.13	.3004	.8129	163.95	.2983	.7485	163.77	.2963
0							.8240	165.92	.3007	.7588	165.74	.2987

mp	Pressure 140 psia Gage pressure 125.3 psig (Sat temp 71.3 F)			Pressure 150 psia Gage pressure 135.3 psig (Sat temp 75.8 F)			Pressure 160 psia Gage pressure 145.3 psig (Sat temp 80.1 F)			Pressure 180 psia Gage pressure 165.3 psig (Sat temp 88.2 F)		
	v	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
sat)	(.3919)	(111.60)	(0.2167)	(.3648)	(111.89)	(0.2160)	(.3411)	(112.06)	(0.2152)	(.3012)	(112.60)	(0.2138)
80	.4022	113.08	.2195	.3695	112.61	.2173						
90	.4141	114.79	.2226	.3810	114.35	.2205	.3518	113.90	.2184	.3031	112.93	.2145
00	.4257	116.48	.2256	.3922	116.07	.2235	.3627	115.64	.2215	.3135	114.73	.2177
10	.4369	118.16	.2286	.4030	117.77	.2265	.3733	117.36	.2246	.3236	116.51	.2208
20	.4483	119.86	.2315	.4139	119.49	.2296	.3838	119.10	.2276	.3331	118.29	.2239
30	.4593	121.57	.2344	.4245	121.21	.2325	.3939	120.84	.2306	.3427	120.07	.2269
40	.4704	123.27	.2373	.4349	122.93	.2354	.4039	122.58	.2335	.3520	121.85	.2299
50	.4812	124.99	.2402	.4453	124.66	.2382	.4138	124.33	.2364	.3611	123.63	.2329
60	.4921	126.72	.2430	.4556	126.40	.2411	.4236	126.08	.2392	.3701	125.41	.2358
70	.5027	128.46	.2458	.4657	128.15	.2439	.4332	127.84	.2420	.3791	127.19	.2386
80	.5132	130.20	.2485	.4757	129.91	.2466	.4429	129.60	.2448	.3879	128.98	.2415
90	.5236	131.96	.2512	.4856	131.67	.2494	.4522	131.38	.2476	.3966	130.78	.2443
00	.5340	133.73	.2539	.4953	133.45	.2521	.4615	133.17	.2503	.4051	132.59	.2470
10	.5444	135.50	.2566	.5051	135.24	.2548	.4709	134.96	.2530	.4137	134.41	.2498
20	.5546	137.29	.2593	.5149	137.03	.2574	.4801	136.76	.2557	.4221	136.23	.2525
30	.5649	139.08	.2619	.5245	138.83	.2601	.4892	138.57	.2583	.4303	138.06	.2551
40	.5749	140.89	.2645	.5341	140.64	.2624	.4983	140.40	.2610	.4386	139.90	.2578
50	.5850	142.70	.2671	.5436	142.47	.2653	.5073	142.33	.2636	.4468	141.75	.2604
60	.5951	144.54	.2697	.5530	144.31	.2679	.5162	144.08	.2662	.4550	143.61	.2630
70	.6051	146.39	.2722	.5625	146.17	.2704	.5251	145.94	.2687	.4630	145.49	.2656
80	.6150	148.26	.2747	.5718	148.04	.2730	.5340	147.82	.2713	.4711	147.38	.2682
90	.6248	150.13	.2773	.5811	149.92	.2755	.5428	149.70	.2738	.4791	149.27	.2707
00	.6347	152.02	.2798	.5903	151.81	.2780	.5516	151.60	.2763	.4870	151.17	.2732
10	.6446	153.91	.2822	.5997	153.71	.2805	.5604	153.51	.2788	.4951	153.09	.2757
20	.6544	155.83	.2847	.6090	155.63	.2829	.5692	155.43	.2813	.5029	155.03	.2782
30	.6641	157.75	.2871	.6181	157.56	.2854	.5777	157.36	.2838	.5106	156.97	.2807
40	.6739	159.68	.2896	.6278	159.59	.2878	.5864	159.31	.2862	.5184	158.92	.2832
50	.6835	161.63	.2920	.6364	161.45	.2903	.5950	161.26	.2886	.5263	160.89	.2856
60	.6933	163.59	.2944	.6454	163.41	.2927	.6036	163.23	.2911	.5340	163.87	.2881
70	.7029	165.57	.2968	.6545	165.39	.2951	.6121	165.21	.2935	.5417	164.87	.2905
80	.7125	167.56	.2992	.6635	167.39	.2975	.6207	167.22	.2959	.5493	166.87	.2929
90							.6292	169.22	.2982	.5570	168.87	.2953

Table 19. F-22 ($CHClF_2$) Monochlorodifluoromethane— Properties of Superheated Vapor (Concluded)

Temp F	Pressure 200 psia Gage pressure 185.3 psig (Sat temp 95.6 F)			Pressure 220 psia Gage pressure 205.3 psig (Sat temp 102.5 F)			Pressure 240 psia Gage pressure 225.3 psig (Sat temp 108.9 F)			Pressure 260 psia Gage pressure 245.3 psig (Sat temp 115.0 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
(*at sat*)	(*.2690*)	(*112.93*)	(*0.2124*)	(*.2425*)	(*113.17*)	(*0.2112*)	(*.2202*)	(*113.28*)	(*0.2099*)	(*.2013*)	(*113.42*)	(*0.208*
100	.2734	113.75	.2139									
110	.2829	115.61	.2172	.2498	114.62	.2138	.2207	113.51	.2103			
120	.2922	117.42	.2204	.2586	116.49	.2171	.2305	115.48	.2137	.2066	114.45	.210
130	.3013	119.25	.2236	.2674	118.37	.2203	.2388	117.43	.2171	.2147	116.47	.214
140	.3102	121.06	.2266	.2759	120.23	.2235	.2471	119.36	.2204	.2227	118.45	.217
150	.3189	122.89	.2296	.2843	122.11	.2265	.2551	121.29	.2236	.2303	120.44	.220
160	.3274	124.71	.2326	.2923	123.96	.2295	.2629	123.18	.2266	.2379	122.36	.223
170	.3358	126.52	.2355	.3004	125.81	.2325	.2705	125.06	.2297	.2454	124.29	.227
180	.3439	128.34	.2384	.3079	127.66	.2354	.2779	126.95	.2327	.2526	126.22	.230
190	.3521	130.17	.2412	.3155	129.52	.2383	.2851	128.85	.2356	.2596	128.16	.233
200	.3599	132.00	.2440	.3229	131.39	.2411	.2922	130.76	.2385	.2663	130.10	.235
210	.3679	133.84	.2467	.3304	133.25	.2439	.2992	132.64	.2413	.2728	132.01	.238
220	.3756	135.68	.2495	.3377	135.12	.2467	.3061	134.53	.2441	.2793	133.92	.241
230	.3834	137.53	.2522	.3450	136.99	.2494	.3128	136.42	.2469	.2857	135.84	.244
240	.3909	139.39	.2548	.3519	138.87	.2521	.3196	138.22	.2496	.2921	137.76	.247
250	.3985	141.25	.2575	.3590	140.76	.2548	.3262	140.24	.2523	.2984	139.70	.249
260	.4060	143.13	.2601	.3660	142.64	.2575	.3326	142.14	.2550	.3045	141.62	.252
270	.4134	145.02	.2627	.3729	144.55	.2601	.3392	144.06	.2576	.3107	143.56	.255
280	.4209	146.92	.2653	.3798	146.46	.2627	.3456	145.99	.2603	.3168	145.51	.258
290	.4281	148.83	.2679	.3866	148.38	.2653	.3520	147.93	.2629	.3229	147.46	.260
300	.4351	150.75	.2704	.3933	150.31	.2678	.3582	149.87	.2654	.3287	149.43	.263
310	.4428	152.68	.2729	.4001	152.25	.2704	.3645	151.82	.2680	.3346	151.39	.265
320	.4499	154.62	.2754	.4067	154.21	.2729	.3707	153.79	.2705	.3404	153.37	.268
330	.4572	156.57	.2779	.4133	156.17	.2754	.3769	155.77	.2731	.3462	155.36	.270
340	.4642	158.54	.2804	.4199	158.15	.2779	.3830	157.75	.2755	.3520	157.36	.273
350	.4713	160.51	.2829	.4264	160.13	.2804	.3892	159.75	.2780	.3577	159.36	.275
360	.4784	162.50	.2853	.4330	162.13	.2828	.3952	161.75	.2805	.3633	161.38	.278
370	.4855	164.50	.2877	.4394	164.14	.2852	.4012	163.77	.2829	.3690	163.41	.280
380	.4924	166.51	.2902	.4459	166.16	.2877	.4072	165.81	.2854	.3746	165.46	.283
390	.4994	168.53	.2926	.4524	168.19	.2901	.4132	167.85	.2878	.3802	167.51	.285
400	.5064	170.57	.2949	.4588	170.24	.2925	.4192	169.90	.2902	.3858	169.57	.288
410				.4652	172.31	.2949	.4251	171.98	.2926	.3913	171.66	.290
420										.3968	173.77	.292

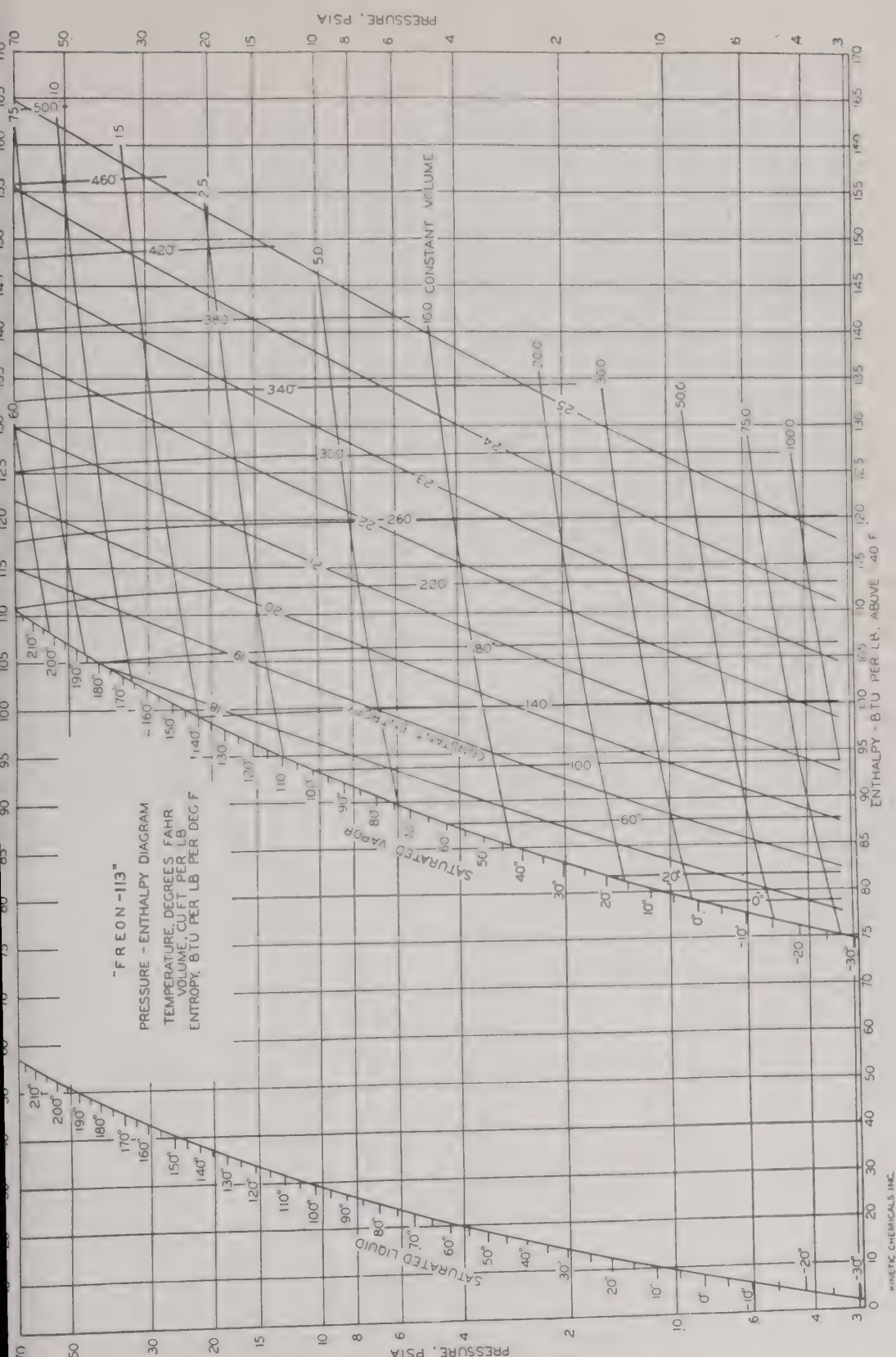

"FREON-113"
PRESSURE - ENTHALPY DIAGRAM
TEMPERATURE, DEGREES FAHR
VOLUME, CU FT PER LB
ENTROPY, BTU PER LB PER DEG F
PRESSURE, PSIA
ENTHALPY - BTU PER LB, ABOVE 40 F
SATURATED LIQUID
SATURATED VAPOR
CONSTANT VOLUME
KINETIC CHEMICALS INC.

Chart 11

Table 20. F-113 (CCl_2F-CCl_2F_2) Trichlorotrifluoromethane—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−30	0.2987	29.31*	0.00947	82.26	1.97	74.65	0.0047	0.1738
−20	.4288	29.05*	.00953	58.61	3.96	76.05	.0092	.1732
−10	.6046	28.69*	.00959	42.48	5.96	77.47	.0137	.1728
0	0.8377	28.21*	0.00966	31.31	7.98	78.89	0.0182	0.1725
2	.8924	28.10*	.00967	29.52	8.38	79.18	.0190	.1724
4	.9503	27.99*	.00968	27.84	8.78	79.46	.0199	.1724
5†	.9802	27.92*	.00969	27.04	8.98	79.60	.0203	.1723
6	1.011	27.86*	.00970	26.27	9.19	79.75	.0208	.1723
8	1.075	27.73*	.00971	24.81	9.59	80.03	.0216	.1723
10	1.142	27.60*	0.00972	23.45	10.00	80.32	0.0225	0.1723
12	1.213	27.45*	.00974	22.17	10.41	80.61	.0234	.1722
14	1.288	27.30*	.00975	20.97	10.81	80.89	.0242	.1722
16	1.366	27.14*	.00977	19.84	11.22	81.18	.0251	.1722
18	1.448	26.97*	.00978	18.79	11.62	81.46	.0259	.1722
20	1.534	26.80*	0.00979	17.81	12.03	81.75	0.0268	0.1722
22	1.624	26.61*	.00981	16.89	12.44	82.04	.0276	.1721
24	1.719	26.42*	.00982	16.02	12.85	82.33	.0285	.1721
26	1.818	26.22*	.00984	15.20	13.26	82.62	.0293	.1722
28	1.922	26.01*	.00985	14.43	13.67	82.91	.0302	.1722
30	2.031	25.79*	0.00987	13.71	14.08	83.20	0.0310	0.1722
32	2.145	25.55*	.00988	13.03	14.49	83.49	.0318	.1722
34	2.264	25.31*	.00990	12.39	14.91	83.78	.0327	.1722
36	2.388	25.06*	.00991	11.79	15.32	84.07	.0335	.1722
38	2.519	24.79*	.00993	11.22	15.74	84.36	.0343	.1722
40	2.655	24.52*	0.00994	10.68	16.16	84.65	0.0352	0.1723
42	2.797	24.23*	.00996	10.18	16.57	84.94	.0360	.1723
44	2.944	23.93*	.00997	9.703	16.99	85.24	.0368	.1723
46	3.098	23.61*	.00999	9.253	17.41	85.53	.0377	.1724
48	3.258	23.29*	.01000	8.830	17.82	85.82	.0385	.1724
50	3.427	22.94*	0.01002	8.426	18.24	86.11	0.0393	0.1725
52	3.602	22.59*	.01003	8.044	18.66	86.40	.0401	.1726
54	3.784	22.22*	.01005	7.682	19.08	86.69	.0410	.1726
56	3.973	21.83*	.01006	7.342	19.50	86.98	.0418	.1727
58	4.170	21.43*	.01008	7.018	19.93	87.28	.0426	.1727
60	4.374	21.02*	0.01010	6.713	20.35	87.57	0.0434	0.1728
62	4.586	20.59*	.01011	6.424	20.77	87.86	.0442	.1729
64	4.807	20.14*	.01013	6.149	21.19	88.15	.0450	.1729
66	5.036	19.67*	.01015	5.889	21.62	88.45	.0459	.1730
68	5.275	19.18*	.01016	5.640	22.05	88.74	.0467	.1731
70	5.523	18.68*	0.01018	5.404	22.48	89.04	0.0475	0.1731
72	5.780	18.16*	.01019	5.180	22.90	89.33	.0483	.1732
74	6.042	17.62*	.01021	4.971	23.33	89.62	.0491	.1733
76	6.320	17.06*	.01023	4.769	23.76	89.92	.0499	.1734
78	6.607	16.47*	.01025	4.574	24.19	90.21	.0507	.1735
80	6.902	15.87*	0.01026	4.392	24.63	90.51	0.0515	0.1736
82	7.208	15.25*	.01028	4.218	25.06	90.80	.0523	.1737
84	7.527	14.60*	.01030	4.051	25.49	91.09	.0531	.1738
86†	7.856	13.93*	.01031	3.893	25.93	91.39	.0539	.1739
88	8.194	13.24*	.01033	3.742	26.36	91.68	.0547	.1740
90	8.545	12.53*	0.01035	3.600	26.80	91.98	0.0555	0.1741
92	8.908	11.79*	.01037	3.463	27.24	92.28	.0563	.1742
94	9.281	11.03*	.01039	3.333	27.67	92.57	.0571	.1743
96	9.668	10.24*	.01040	3.208	28.11	92.86	.0578	.1744
98	10.07	9.42*	.01042	3.089	28.55	93.15	.0586	.1745
100	10.48	8.59*	0.01044	2.976	28.99	93.45	0.0594	0.1746
120	15.40	0.70	.01063	2.078	33.48	96.41	.0673	.1758
140	21.93	7.23	.01083	1.491	38.05	99.36	.0750	.1773
160	30.44	15.74	.01105	1.094	42.74	102.29	.0827	.1788
180	41.22	26.52	.01128	0.8193	47.53	105.19	.0903	.1804
200	54.66	39.96	0.01153	0.6241	52.45	108.07	0.0978	0.1821

* Inches of mercury below one atmosphere.
† Standard cycle temperatures.

Table 21. F-113 (CCl_2F-CCl_2F_2) Trichlorotrifluoroethane—Properties of Superheated Vapor

mp	Pressure 1.0 psia Gage pressure 27.88 in. vac (Sat temp 5.6 F)			Pressure 2.0 psia Gage pressure 25.85 in. vac (Sat temp 29.4 F)			Pressure 3.0 psia Gage pressure 23.81 in. vac (Sat temp 44.7 F)			Pressure 4.0 psia Gage pressure 21.78 in. vac (Sat temp 56.3 F)		
	v	h	s	v	h	s	v	h	s	v	h	s
sat)	*(26.54)*	*(79.69)*	*(0.1723)*	*(13.91)*	*(83.12)*	*(0.1722)*	*(9.587)*	*(85.34)*	*(0.1723)*	*(7.297)*	*(87.03)*	*(0.1727)*
50	29.10	86.20	0.1857	14.50	86.16	0.1783	9.637	86.12	0.1739			
60	29.67	87.70	.1886	14.79	87.66	.1812	9.831	87.62	.1769	7.351	87.58	0.1738
70	30.25	89.31	.1915	15.08	89.17	.1841	10.02	89.12	.1797	7.497	89.10	.1767
80	30.82	90.73	.1943	15.37	90.69	.1869	10.22	90.66	.1826	7.642	90.62	.1795
90	31.39	92.27	.1972	15.66	92.23	.1898	10.41	92.19	.1854	7.787	92.15	.1823
00	31.97	93.81	0.2000	15.95	93.77	0.1926	10.60	93.74	0.1882	7.932	93.70	0.1851
10	32.56	95.37	.2027	16.24	95.33	.1953	10.80	95.30	.1910	8.077	95.26	.1879
20	33.14	96.94	.2054	16.53	96.90	.1980	10.99	96.87	.1937	8.222	96.83	.1906
30	33.71	98.52	.2081	16.82	98.49	.2007	11.18	98.46	.1964	8.367	98.42	.1933
40	34.28	100.11	.2108	17.11	100.09	.2034	11.38	100.05	.1991	8.522	100.01	.1960
50	34.85	101.72	0.2135	17.39	101.69	0.2061	11.57	101.65	0.2017	8.667	101.62	0.1986
60	35.42	103.33	.2161	17.68	103.31	.2087	11.76	103.27	.2044	8.812	103.23	.2003
70	35.99	104.96	.2187	17.96	104.93	.2113	11.95	104.90	.2070	8.957	104.86	.2039
80	36.57	106.60	.2213	17.25	106.57	.2139	12.15	106.54	.2095	9.102	106.51	.2065
90	37.15	108.25	.2238	18.54	108.22	.2164	12.34	108.19	.2121	9.247	108.16	.2091
00	37.73	109.91	0.2264	18.83	109.88	0.2190	12.53	109.85	0.2147	9.387	109.82	0.2116
10	38.30	111.59	.2289	19.12	111.56	.2215	12.72	111.53	.2172	9.532	111.50	.2141
20	38.88	113.28	.2314	19.40	113.24	.2240	12.92	113.22	.2197	9.677	113.19	.2166
30	39.46	114.97	.2339	19.69	114.94	.2265	13.11	114.92	.2222	9.822	114.88	.2191
40	40.03	116.68	.2363	19.97	116.65	.2290	13.30	116.63	.2246	9.967	116.59	.2216

mp F	Pressure 6.0 psia Gage pressure 17.71 in. vac (Sat temp 73.7 F)			Pressure 8.0 psia Gage pressure 13.64 in. vac (Sat temp 86.9 F)			Pressure 10.0 psia Gage pressure 9.57 in. vac (Sat temp 97.6 F)			Pressure 12 psia Gage pressure 5.50 in. vac (Sat temp 106.8 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
sat)	*(5.004)*	*(89.58)*	*(0.1735)*	*(3.828)*	*(91.52)*	*(0.1730)*	*(3.110)*	*(93.11)*	*(0.1745)*	*(2.624)*	*(94.46)*	*(0.1750)*
00	5.260	93.62	0.1807	3.924	93.56	0.1776	3.124	93.48	0.1751			
10	5.357	95.18	.1835	3.999	95.12	.1804	3.183	95.04	.1779	2.640	94.97	0.1759
20	5.454	96.76	.1862	4.073	96.69	.1831	3.242	96.62	.1805	2.689	96.55	.1785
30	5.551	98.35	.1889	4.147	98.28	.1858	3.301	98.21	.1834	2.738	98.14	.1813
40	5.649	99.95	.1916	4.221	99.88	.1885	3.361	99.81	.1860	2.788	99.73	.1840
50	5.746	101.56	0.1943	4.295	101.48	0.1912	3.420	101.41	0.1887	2.838	101.34	0.1867
60	5.843	103.17	.1969	4.369	103.10	.1938	3.480	103.03	.1914	2.888	102.97	.1894
70	5.940	104.80	.1995	4.442	104.74	.1964	3.539	104.66	.1940	2.938	104.60	.1920
80	6.037	106.45	.2021	4.515	106.38	.1990	3.599	106.31	.1966	2.987	106.25	.1964
90	6.134	108.10	.2047	4.588	108.04	.2016	3.658	107.96	.1991	3.037	107.90	.1972
200	6.233	109.77	0.2072	4.661	109.71	0.2041	3.716	109.63	0.2017	3.086	109.57	0.1997
210	6.330	111.44	.2097	4.734	111.38	.2066	3.774	111.32	.2042	3.135	111.26	.2022
220	6.427	113.13	.2122	4.807	113.07	.2091	3.832	113.01	.2067	3.184	112.95	.2047
230	6.525	114.83	.2147	4.880	114.77	.2116	3.891	114.71	.2092	3.234	114.65	.2072
240	6.622	116.54	.2172	4.952	116.48	.2141	3.950	116.42	.2117	3.283	116.36	.2097
250	6.719	118.26	0.2197	5.024	118.20	0.2166	4.010	118.14	0.2141	3.332	118.08	0.2121
260	6.816	119.99	.2221	5.096	119.93	.2190	4.068	119.87	.2166	3.382	119.81	.2146
270	6.913	121.73	.2245	5.168	121.67	.2214	4.126	121.61	.2190	3.431	121.55	.2170
280	7.011	123.48	.2269	5.240	123.42	.2238	4.185	123.37	.2214	3.479	123.31	.2194
290	7.108	125.25	.2293	5.313	125.19	.2261	4.243	125.15	.2237	3.528	125.09	.2218

mp F	Pressure 20 psia Gage pressure 5.3 psig (Sat temp 134.6 F)			Pressure 30 psia Gage pressure 15.3 psig (Sat temp 159.1 F)			Pressure 40 psia Gage pressure 25.3 psig (Sat temp 177.9 F)			Pressure 50 psia Gage pressure 35.3 psig (Sat temp 193.5 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
sat)	*(1.625)*	*(98.56)*	*(0.1769)*	*(1.109)*	*(102.15)*	*(0.1787)*	*(0.8435)*	*(104.90)*	*(0.1802)*	*(0.6809)*	*(107.14)*	*(0.1816)*
200	1.827	109.32	0.1940	1.194	108.97	0.1894	0.8791	108.62	0.1860	0.6899	108.27	0.1833
210	1.857	111.01	.1966	1.215	110.67	.1919	.8953	110.32	.1885	.7031	109.97	.1858
220	1.887	112.70	.1991	1.236	112.37	.1945	.9114	112.03	.1911	.7163	111.69	.1884
230	1.917	114.40	.2016	1.257	114.08	.1970	.9274	113.74	.1936	.7296	113.41	.1909
240	1.947	116.12	.2041	1.278	115.80	.1995	.9435	115.47	.1951	.7430	115.14	.1934
250	1.977	117.84	0.2065	1.299	117.53	0.2019	0.9596	117.21	0.1986	0.7562	116.90	0.1959
260	2.007	119.58	.2090	1.320	119.28	.2044	.9756	118.97	.2010	.7693	118.67	.1984
270	2.037	121.33	.2114	1.341	121.03	.2068	0.9915	120.74	.2035	.7821	120.45	.2008
280	2.068	123.09	.2138	1.362	122.80	.2092	1.008	122.52	.2059	.7950	122.23	.2033
290	2.098	124.87	.2162	1.382	124.59	.2116	1.023	124.31	.2083	.8079	124.03	.2057
300	2.128	126.67	0.2185	1.402	126.39	0.2140	1.038	126.12	0.2107	0.8207	125.84	0.2081
310	2.158	128.47	.2209	1.422	128.20	.2163	1.054	127.94	.2131	.8335	127.66	.2104
320	2.187	130.28	.2232	1.442	130.02	.2187	1.070	129.76	.2154	.8463	129.48	.2128
330	2.217	132.10	.2255	1.462	131.85	.2210	1.085	131.59	.2178	.8591	131.31	.2152
340	2.247	133.94	.2278	1.483	133.69	.2233	1.101	133.44	.2201	.8716	133.16	.2175
350	2.277	135.78	0.2301	1.503	135.54	0.2256	1.117	135.29	0.2224	0.8844	135.02	0.2198
360	2.308	137.64	.2324	1.524	137.40	.2279	1.133	137.16	.2247	.8973	136.89	.2221
370	2.338	139.50	.2347	1.545	139.27	.2302	1.148	139.03	.2269	.9101	138.77	.2244
380	2.368	141.38	.2369	1.565	141.15	.2324	1.164	140.91	.2292	.9228	140.67	.2266
390	2.398	143.27	.2392	1.585	143.05	.2347	1.179	142.81	.2314	.9354	142.57	.2289

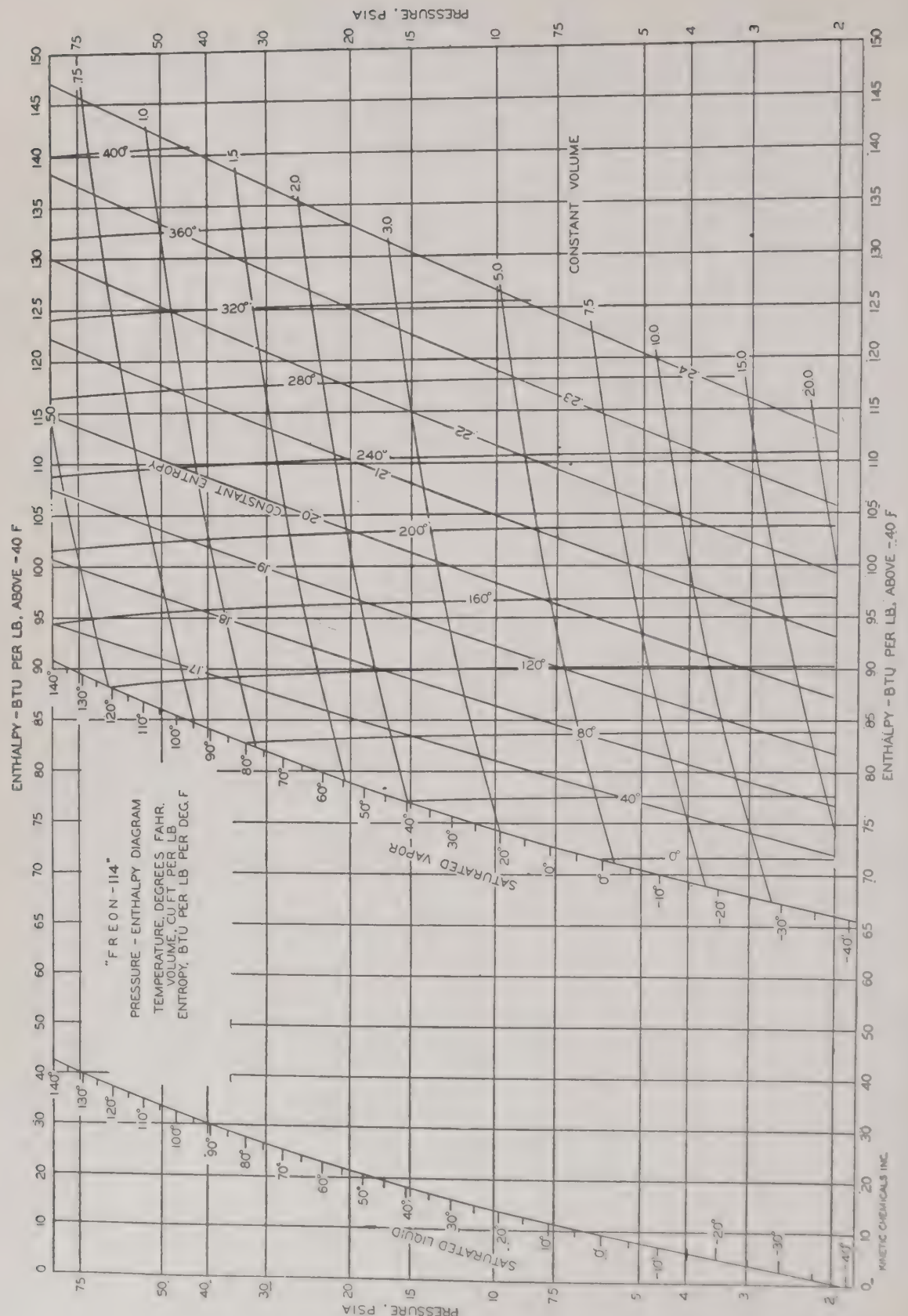

"FREON-114"
PRESSURE - ENTHALPY DIAGRAM
TEMPERATURE, DEGREES FAHR.
VOLUME, CU FT PER LB.
ENTROPY, BTU PER LB PER DEG. F
ENTHALPY - BTU PER LB. ABOVE -40 F
PRESSURE, PSIA
CONSTANT VOLUME
CONSTANT ENTROPY
SATURATED LIQUID
SATURATED VAPOR
KINETIC CHEMICALS INC.

Chart 12

Table 22. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−80	0.464	28.97*	105.603	51.26	−8.73	60.44	−0.0227	0.1595
−78	.50	28.90*	105.398	49.83	−8.30	60.71	− .0215	.1593
−76	.535	28.85*	105.193	44.87	−7.87	60.98	− .0203	.1592
−74	.58	28.73*	105.058	41.69	−7.44	61.25	− .0191	.1590
−72	.62	28.66*	104.924	38.92	−7.01	61.52	− .0179	.1589
−70	0.670	28.59*	104.790	36.40	−6.57	61.79	−0.0167	0.1587
−68	.72	28.46*	104.624	34.20	−6.14	62.06	− .0155	.1586
−66	.775	28.33*	104.458	31.74	−5.71	62.33	− .0143	.1585
−64	.833	28.23*	104.292	29.77	−5.28	62.60	− .0132	.1583
−62	.895	28.10*	104.126	27.79	−4.84	62.88	− .0121	.1582
−60	0.959	27.99*	103.960	26.06	−4.40	63.16	−0.0110	0.1580
−58	1.028	27.83*	103.792	24.55	−3.96	63.43	− .0098	.1579
−56	1.10	27.68*	103.622	22.94	−3.52	63.70	− .0087	.1578
−54	1.175	27.52*	103.452	21.50	−3.08	63.97	− .0075	.1577
−52	1.26	27.35*	103.282	20.16	−2.64	64.25	− .0064	.1576
−50	1.349	27.20*	103.113	18.96	−2.20	64.53	−0.0054	0.1575
−48	1.438	27.0*	102.938	17.85	−1.76	64.80	− .0043	.1574
−46	1.535	26.8*	102.766	16.80	−1.32	65.07	− .0032	.1573
−44	1.635	26.6*	102.594	15.75	−0.88	65.35	− .0021	.1572
−42	1.745	26.4*	102.422	14.87	−0.44	65.63	− .0010	.1571
−40	1.866	26.12*	102.25	14.02	0.00	65.91	0.0000	0.1571
−38	1.990	25.87*	102.08	13.20	0.45	66.19	.0011	.1570
−36	2.121	25.60*	101.90	12.44	0.91	66.47	.0021	.1569
−34	2.259	25.32*	101.72	11.73	1.36	66.74	.0032	.1568
−32	2.404	25.03*	101.55	11.07	1.81	67.02	.0042	.1567
−30	2.557	24.72*	101.37	10.45	2.27	67.30	0.0053	0.1567
−28	2.718	24.39*	101.19	9.877	2.72	67.58	.0063	.1567
−26	2.887	24.04*	101.01	9.338	3.17	67.86	.0074	.1566
−24	3.064	23.68*	100.83	8.833	3.63	68.14	.0084	.1565
−22	3.249	23.31*	100.65	8.362	4.08	68.42	.0095	.1565
−20	3.444	22.91*	100.47	7.921	4.54	68.70	0.0105	0.1565
−18	3.648	22.49*	100.29	7.508	4.99	68.98	.0116	.1565
−16	3.862	22.06*	100.11	7.121	5.44	69.26	.0126	.1564
−14	4.085	21.61*	99.92	6.757	5.90	69.54	.0136	.1564
−12	4.319	21.13*	99.74	6.416	6.35	69.82	.0146	.1564
−10	4.564	20.63*	99.56	6.095	6.81	70.10	0.0157	0.1564
− 8	4.819	20.11*	99.37	5.794	7.26	70.38	.0167	.1564
− 6	5.086	19.57*	99.19	5.510	7.72	70.66	.0177	.1564
− 4	5.365	19.00*	99.00	5.244	8.18	70.94	.0187	.1564
− 2	5.655	18.41*	98.81	4.992	8.63	71.22	.0197	.1565
0	5.958	17.79*	98.62	4.756	9.09	71.50	0.0207	0.1565
2	6.274	17.15*	98.44	4.533	9.54	71.78	.0217	.1565
4	6.603	16.48*	98.25	4.322	10.00	72.07	.0227	.1565
5†	6.772	16.14*	98.15	4.221	10.23	72.21	.0232	.1565
6	6.945	15.78*	98.06	4.123	10.46	72.35	.0236	.1566
8	7.301	15.06*	97.87	3.935	10.91	72.63	.0246	.1566
10	7.671	14.31*	97.68	3.758	11.37	72.91	0.0256	0.1566
12	8.057	13.52*	97.48	3.591	11.83	73.19	.0266	.1567
14	8.457	12.71*	97.29	3.432	12.29	73.47	.0275	.1567
16	8.873	11.86*	97.10	3.282	12.75	73.75	.0285	.1568
18	9.305	10.98*	96.90	3.140	13.20	74.04	.0295	.1568
20	9.753	10.07*	96.71	3.005	13.66	74.32	0.0304	0.1569
22	10.22	9.12*	96.51	2.877	14.12	74.60	.0314	.1569
24	10.70	8.14*	96.32	2.756	14.58	74.88	.0323	.1570
26	11.20	7.12*	96.12	2.641	15.05	75.17	.0333	.1571
28	11.72	6.07*	95.92	2.532	15.51	75.45	.0342	.1571

* Inches of mercury below one atmosphere.
† Standard ton temperatures.

Table 22. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane—Properties of Liquid and Saturated Vapor (Concluded)

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
30	12.25	4.99*	95.73	2.429	15.97	75.73	0.0352	0.1572
32	12.81	3.85*	95.53	2.330	16.43	76.01	.0361	.1573
34	13.38	2.69*	95.33	2.236	16.89	76.29	.0370	.1574
36	13.98	1.47*	95.13	2.147	17.36	76.58	.0380	.1575
38	14.59	0.22*	94.93	2.062	17.82	76.86	.0389	.1575
40	15.22	0.52	94.73	1.982	18.28	77.14	0.0398	0.1576
42	15.88	1.18	94.52	1.905	18.75	77.42	.0408	.1577
44	16.56	1.86	94.32	1.832	19.21	77.70	.0417	.1578
46	17.26	2.56	94.12	1.762	19.68	77.99	.0426	.1579
48	17.98	3.28	93.91	1.695	20.14	78.27	.0435	.1580
50	18.73	4.03	93.71	1.632	20.61	78.55	0.0444	0.1581
52	19.50	4.80	93.50	1.571	21.08	78.83	.0453	.1582
54	20.29	5.59	93.30	1.513	21.54	79.11	.0463	.1583
56	21.11	6.41	93.09	1.458	22.01	79.39	.0472	.1584
58	21.96	7.26	92.88	1.405	22.48	79.67	.0481	.1585
60	22.83	8.13	92.68	1.354	22.95	79.95	0.0490	0.1587
62	23.72	9.02	92.47	1.306	23.42	80.23	.0499	.1588
64	24.64	9.94	92.26	1.260	23.89	80.51	.0508	.1589
66	25.59	10.89	92.05	1.216	24.36	80.79	.0517	.1590
68	26.57	11.87	91.84	1.174	24.83	81.07	.0526	.1591
70	27.57	12.87	91.63	1.133	25.30	81.35	0.0534	0.1593
72	28.61	13.91	91.41	1.094	25.78	81.62	.0543	.1594
74	29.67	14.97	91.20	1.057	26.25	81.90	.0552	.1595
76	30.76	16.06	90.99	1.021	26.73	82.18	.0561	.1596
78	31.88	17.18	90.77	0.9869	27.20	82.46	.0570	.1597
80	33.04	18.34	90.56	0.9541	27.68	82.73	0.0579	0.1599
82	34.22	19.52	90.34	.9226	28.15	83.01	.0587	.1600
84	35.44	20.74	90.13	.8923	28.63	83.29	.0596	.1601
86†	36.69	21.99	89.91	.8632	29.11	83.56	.0605	.1603
88	37.97	23.27	89.69	.8353	29.58	83.84	.0613	.1604
90	39.29	24.59	89.47	0.8084	30.06	84.11	0.0622	0.1605
92	40.64	25.94	89.25	.7827	30.54	84.39	.0631	.1607
94	42.02	27.32	89.03	.7579	31.02	84.66	.0639	.1608
96	43.44	28.74	88.81	.7340	31.50	84.93	.0648	.1609
98	44.89	30.19	88.59	.7111	31.99	85.21	.0656	.1611
100	46.39	31.69	88.37	0.6890	32.47	85.48	0.0665	0.1612
102	47.92	33.22	88.15	.6677	32.95	85.75	.0674	.1614
104	49.48	34.78	87.93	.6472	33.43	86.02	.0682	.1615
106	51.09	36.39	87.70	.6274	33.92	86.29	.0691	.1617
108	52.73	38.03	87.48	.6084	34.40	86.56	.0699	.1618
110	54.41	39.71	87.25	0.5901	34.89	86.83	0.0708	0.1619
112	56.14	41.44	87.01	.5724	35.38	87.09	.0716	.1621
114	57.90	43.20	86.77	.5554	35.87	87.36	.0725	.1622
116	59.70	45.00	86.54	.5389	36.35	87.63	.0733	.1624
118	61.55	46.85	86.31	.5230	36.84	87.89	.0741	.1625
120	63.44	48.74	86.08	0.5077	37.33	88.16	0.0750	0.1627
122	65.37	50.67	85.85	.4929	37.83	88.42	.0758	.1628
124	67.35	52.65	85.61	.4787	38.32	88.69	.0767	.1630
126	69.37	54.67	85.37	.4649	38.81	88.95	.0775	.1631
128	71.43	56.73	85.13	.4515	39.31	89.21	.0783	.1633
130	73.54	58.84	84.89	0.4387	39.80	89.47	0.0792	0.1634
132	75.69	60.99	84.65	.4262	40.30	89.73	.0800	.1635
134	77.90	63.20	84.41	.4142	40.80	89.99	.0808	.1637
136	80.15	65.45	84.16	.4025	41.29	90.25	.0816	.1638
138	82.44	67.74	83.91	.3912	41.79	90.51	.0825	.1640
140	84.79	70.09	83.66	0.3803	42.29	90.76	0.0833	0.1641

* Inches of mercury below one atmosphere.
† Standard ton temperatures.

Table 23. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane—Properties of Superheated Vapor

emp F	Pressure 1.5 psia Gage pressure 26.87 in. vac (Sat temp −46.6 F)			Pressure 2.0 psia Gage pressure 25.85 in. vac (Sat temp −37.8 F)			Pressure 2.5 psia Gage pressure 24.83 in. vac (Sat temp −30.7 F)			Pressure 3.0 psia Gage pressure 23.81 in. vac (Sat temp −24.7 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
sat)	*(17.18)*	*(65.00)*	*(0.1575)*	*(13.14)*	*(66.21)*	*(0.1570)*	*(10.68)*	*(67.20)*	*(0.1567)*	*(9.011)*	*(68.04)*	*(0.1565)*
-40	17.46	65.93	0.1596	—	—	—	—	—	—	—	—	—
-30	17.89	67.35	.1630	13.39	67.33	0.1596	10.70	67.30	0.1570	—	—	—
-20	18.31	68.78	.1663	13.71	68.76	.1629	10.95	68.74	.1603	9.108	68.72	0.1581
-10	18.73	70.23	.1695	14.03	70.21	.1662	11.20	70.18	.1635	9.230	70.16	.1614
0	19.15	71.69	0.1727	14.34	71.67	0.1694	11.46	71.64	0.1667	9.532	71.62	0.1646
10	19.57	73.16	.1759	14.66	73.14	.1725	11.71	73.12	.1699	9.744	73.10	.1678
20	19.99	74.65	.1790	14.98	74.63	.1757	11.96	74.61	.1741	9.956	74.59	.1709
30	20.42	76.15	.1821	15.29	76.13	.1788	12.22	76.11	.1762	10.17	76.09	.1740
40	20.84	77.66	.1852	15.61	77.65	.1818	12.47	77.63	.1792	10.38	77.61	.1771
50	21.26	79.19	0.1882	15.92	79.18	0.1849	12.73	79.16	0.1823	10.59	79.14	0.1801
60	21.68	80.73	.1912	16.24	80.72	.1879	12.98	80.70	.1853	10.80	80.68	.1831
70	22.10	82.29	.1942	16.56	82.28	.1908	13.23	82.26	.1882	11.01	82.24	.1861
80	22.52	83.86	.1971	16.87	83.85	.1938	13.49	83.83	.1912	11.23	83.81	.1890
90	22.94	85.45	.2000	17.19	85.43	.1967	13.74	85.42	.1941	11.44	85.40	.1919
100	23.36	87.05	0.2029	17.51	87.03	0.1996	13.99	87.02	0.1970	11.65	87.00	0.1948
110	23.78	88.66	.2058	17.82	88.64	.2024	14.25	88.63	.1998	11.86	88.61	.1977
120	24.20	90.29	.2086	18.14	90.27	.2053	14.50	90.26	.2026	12.07	90.24	.2005
130	24.62	91.93	.2114	18.45	91.91	.2081	14.75	91.90	.2054	12.28	91.88	.2033
140	25.04	93.58	.2142	18.77	93.57	.2108	15.00	93.55	.2082	12.49	93.54	.2061
150	25.46	95.25	0.2170	19.08	95.23	0.2136	15.26	95.22	0.2110	12.70	95.21	0.2089
160	25.88	96.93	.2197	19.40	96.91	.2163	15.51	96.90	.2137	12.92	96.89	.2116
170	26.30	98.62	.2224	19.71	98.61	.2190	15.76	98.60	.2164	13.13	98.58	.2143
180	26.72	100.33	.2251	20.03	100.32	.2217	16.01	100.31	.2191	13.34	100.29	.2170
190	27.14	102.06	.2278	20.35	102.04	.2244	16.27	102.03	.2218	13.55	102.02	.2197
200	27.56	103.79	0.2304	20.66	103.78	0.2271	16.52	103.77	0.2245	13.76	103.76	0.2223
210	27.98	105.54	.2331	20.98	105.53	.2297	16.77	105.52	.2271	13.97	105.51	.2250
220	28.40	107.31	.2357	21.29	107.30	.2323	17.02	107.28	.2297	14.18	107.27	.2276
230	28.82	109.09	.2383	21.61	109.08	.2349	17.28	109.06	.2323	14.39	109.05	.2302
240	29.24	110.88	.2409	21.92	110.87	.2375	17.53	110.86	.2349	14.60	110.84	.2328
250	29.66	112.09	0.2434	22.24	112.67	0.2401	17.78	112.66	0.2375	14.81	112.65	0.2353
260				22.55	114.49	0.2426	18.03	114.48	0.2400	15.02	114.67	.2379
270										15.23	116.31	0.2404

emp F	Pressure 3.5 psia Gage pressure 22.80 in. vac (Sat temp −19.5 F)			Pressure 4.0 psia Gage pressure 21.78 in. vac (Sat temp −14.8 F)			Pressure 4.5 psia Gage pressure 20.76 in. vac (Sat temp −10.5 F)			Pressure 5.0 psia Gage pressure 19.74 in. vac (Sat temp −6.6 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
t sat)	*(7.806)*	*(66.77)*	*(0.1565)*	*(6.893)*	*(69.43)*	*(0.1565)*	*(6.177)*	*(70.02)*	*(0.1564)*	*(5.601)*	*(70.57)*	*(0.1564)*
-10	7.976	70.14	0.1596	6.967	70.12	0.1580	6.183	70.10	0.1566	—	—	—
0	8.158	71.60	0.1628	7.127	71.58	0.1612	6.326	71.56	0.1598	5.684	71.54	0.1585
10	8.340	73.08	.1660	7.287	73.06	.1644	6.468	73.04	.1630	5.813	73.02	.1617
20	8.522	74.57	.1691	7.446	74.55	.1675	6.610	74.53	.1661	5.941	74.51	.1649
30	8.704	76.07	.1722	7.606	76.05	.1706	6.752	76.04	.1692	6.069	76.02	.1680
40	8.886	77.59	.1753	7.765	77.57	.1737	6.894	77.56	.1723	6.197	77.54	.1711
50	9.068	79.12	0.1783	7.925	79.10	0.1767	7.036	79.09	0.1753	6.325	79.07	0.1741
60	9.250	80.67	.1813	8.085	80.65	.1797	7.178	80.63	.1783	6.453	80.61	.1771
70	9.431	82.23	.1843	8.244	82.21	.1827	7.320	82.19	.1813	6.581	82.17	.1801
80	9.613	83.80	.1872	8.403	83.78	.1856	7.462	83.76	.1843	6.709	83.75	.1830
90	9.795	85.38	.1901	8.562	85.37	.1885	7.604	85.35	.1872	6.837	85.33	.1859
100	9.976	86.98	0.1930	8.721	86.97	0.1914	7.745	86.95	0.1901	6.964	86.93	0.1888
110	10.16	88.60	.1959	8.880	88.58	.1943	7.887	88.57	.1929	7.092	88.55	.1917
120	10.34	90.23	.1987	9.039	90.21	.1971	8.028	90.20	.1957	7.219	90.19	.1945
130	10.52	91.87	.2015	9.198	91.85	.1999	8.170	91.84	.1985	7.346	91.82	.1973
140	10.70	93.52	.2043	9.356	93.51	.2027	8.311	93.49	.2013	7.474	93.48	.2001
150	10.88	95.19	0.2071	9.515	95.18	0.2055	8.452	95.16	0.2041	7.601	95.15	0.2029
160	11.06	96.87	.2098	9.673	96.86	.2082	8.593	96.84	.2068	7.728	96.83	.2056
170	11.24	98.57	.2125	9.832	98.56	.2109	8.734	98.54	.2096	7.855	98.53	.2083
180	11.43	100.28	.2152	9.990	100.27	.2136	8.875	100.25	.2123	7.983	100.24	.2110
190	11.61	102.00	.2179	10.15	101.99	.2163	9.015	101.98	.2149	8.110	101.96	.2137
200	11.79	103.74	0.2205	10.31	103.73	0.2190	9.156	103.72	0.2176	8.236	103.70	0.2163
210	11.97	105.49	.2232	10.47	105.48	.2216	9.297	105.47	.2202	8.363	105.46	.2190
220	12.15	107.26	.2258	10.62	107.25	.2242	9.438	107.24	.2228	8.490	107.22	.2216
230	12.33	109.04	.2284	10.78	109.03	.2268	9.579	109.02	.2254	8.617	109.00	.2242
240	12.51	110.83	.2310	10.94	110.82	.2294	9.720	110.81	.2280	8.744	110.80	.2268
250	12.69	112.64	0.2335	11.10	112.63	0.2320	9.861	112.62	0.2306	8.871	112.61	0.2294
260	12.87	114.46	.2361	11.26	114.45	.2345	10.00	114.44	.2331	8.998	114.43	.2319
270	13.05	116.30	.2386	11.42	116.28	.2371	10.14	116.27	.2357	9.125	116.26	.2344
280	13.23	118.15	0.2411	11.57	118.13	0.2396	10.28	118.12	0.2382	9.251	118.11	.2370
290										9.378	119.98	0.2395

Table 23. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane— Properties of Superheated Vapor (Continued)

Temp F	Pressure 5.5 psia Gage pressure 18.73 in. vac (Sat temp −3.1 F)			Pressure 6.0 psia Gage pressure 17.71 in. vac (Sat temp +0.3 F)			Presure 6.5 psia Gage pressure 16.69 in. vac (Sat temp 3.4 F)			Pressure 7.0 psia Gage pressure 15.67 in. (Sat temp 6.3 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*5.125*)	(*71.07*)	(*0.1564*)	(*4.726*)	(*71.54*)	(*0.1565*)	(*4.386*)	(*71.98*)	(*0.1565*)	(*4.094*)	(*72.39*)	(*0.15*
0	5.159	71.52	0.1574									
10	5.276	73.00	.1606	4.829	72.98	0.1595	4.451	72.96	0.1586	4.127	72.94	0.15
20	5.393	74.49	.1637	4.937	74.47	.1627	4.551	74.45	.1618	4.220	74.43	.16
30	5.510	76.00	.1668	5.044	75.98	.1658	4.650	75.96	.1649	4.312	75.94	.16
40	5.626	77.52	.1699	5.151	77.50	.1689	4.749	77.48	.1679	4.404	77.46	.16
50	5.743	79.05	0.1730	5.258	79.03	0.1719	4.848	79.01	.1710	4.496	78.99	0.17
60	5.860	80.60	.1760	5.365	80.58	.1749	4.947	80.56	.1740	4.588	80.54	.17
70	5.976	82.16	.1789	5.472	82.14	.1779	5.046	82.12	.1770	4.680	82.10	.17
80	6.092	83.73	.1819	5.579	83.71	.1809	5.145	83.70	.1799	4.772	83.68	.17
90	6.209	85.32	.1848	5.686	85.30	.1838	5.244	85.28	.1828	4.864	85.27	.18
100	6.325	86.92	0.1877	5.793	86.90	0.1867	5.342	86.89	0.1857	4.956	86.87	0.18
110	6.441	88.53	.1905	5.899	88.52	.1895	5.441	88.50	.1886	5.048	88.49	.18
120	6.557	90.16	.1934	6.006	90.15	.1924	5.539	90.13	.1914	5.139	90.12	.19
130	6.673	91.81	.1962	6.112	91.79	.1952	5.638	91.78	.1942	5.231	91.76	.19
140	6.789	93.46	.1990	6.219	93.45	.1980	5.736	93.43	.1970	5.322	93.42	.19
150	6.905	95.13	0.2017	6.235	95.12	0.2007	5.834	95.10	0.1998	5.414	95.09	0.19
160	7.021	96.82	.2045	6.431	96.80	.2035	5.932	96.79	.2025	5.505	96.77	.20
170	7.136	98.52	.2072	6.537	98.50	.2062	6.031	98.49	.2052	5.596	98.47	.20
180	7.252	100.23	.2099	6.644	100.21	.2089	6.129	100.29	.2079	5.687	100.18	.20
190	7.368	101.95	.2126	6.750	101.94	.2115	6.227	101.92	.2106	5.778	101.91	.20
200	7.483	103.69	9.2152	6.856	103.68	0.2142	6.325	103.66	0.2133	5.870	103.65	0.21
210	7.599	105.44	.2179	6.962	105.43	.2168	6.423	105.42	.2159	5.961	105.40	.21
220	7.714	107.21	.2205	7.068	107.20	.2195	6.521	107.19	.2185	6.052	107.17	.21
230	7.830	108.99	.2231	7.173	108.98	.2221	6.619	108.97	.2211	6.143	108.95	.22
240	7.945	110.78	.2257	7.279	110.77	.2246	6.716	110.76	.2237	6.234	110.75	.22
250	8.061	112.59	0.2282	7.385	112.58	0.2272	6.814	112.57	0.2263	6.325	112.56	0.22
260	8.176	114.42	.2308	7.491	114.40	.2298	6.912	114.39	.2288	6.416	114.38	.22
270	8.291	116.25	.2333	7.597	116.24	.2323	7.010	116.23	.2314	6.507	116.22	.23
280	8.407	118.10	.2358	7.703	118.09	.2348	7.108	118.08	.2339	6.597	118.07	.23
290	8.522	119.96	0.2383	7.809	119.95	.2373	7.205	119.94	.2364	6.688	119.93	.23
300				7.914	121.83	0.2398	7.303	121.82	0.2389	6.779	121.81	0.23

Temp F	Pressure 7.5 psia Gage pressure 14.65 in. vac (Sat temp 9.1 F)			Pressure 8.0 psia Gage pressure 13.64 in. vac (Sat temp 11.7 F)			Pressure 9 psia Gage pressure 11.60 in. vac (Sat temp 16.6 F)			Pressure 10 psia Gage pressure 9.57 in. (Sat temp 21.1 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*3.839*)	(*72.78*)	(*0.1566*)	(*3.615*)	(*73.15*)	(*0.1567*)	(*3.240*)	(*73.84*)	(*0.1568*)	(*2.937*)	(*74.47*)	(*0.156*
10	3.846	72.92	0.1569									
20	3.933	74.41	.1600	3.682	74.39	0.1593	3.263	74.35	0.1578			
30	4.019	75.92	.1631	3.763	75.90	.1624	3.336	75.86	.1609	2.994	75.82	0.15
40	4.105	77.44	.1662	3.844	77.42	.1655	3.408	77.38	.1640	3.060	77.35	.16
50	4.191	78.98	0.1693	3.925	78.96	0.1685	3.480	78.92	0.1671	3.125	78.88	0.16
60	4.277	80.53	.1723	4.006	80.51	.1715	3.553	80.47	.1701	3.190	80.43	.16
70	4.363	82.09	.1753	4.086	82.07	.1745	3.625	82.03	.1731	3.255	82.00	.17
80	4.449	83.66	.1782	4.167	83.64	.1774	3.697	83.61	.1760	3.320	83.58	.17
90	4.535	85.25	.1811	4.248	85.23	.1803	3.769	85.20	.1789	3.385	85.17	.17
100	4.621	86.85	0.1840	4.328	86.84	0.1832	3.840	86.80	0.1818	3.450	86.77	0.18
110	4.707	88.47	.1869	4.409	88.46	.1861	3.912	88.42	.1847	3.515	88.39	.18
120	4.793	90.10	.1897	4.489	90.09	.1889	3.984	90.05	.1875	3.579	90.02	.18
130	4.878	91.75	.1925	4.570	91.73	.1917	4.055	91.70	.1903	3.644	91.67	.18
140	4.964	93.40	.1953	4.650	93.39	.1945	4.127	93.36	.1931	3.708	93.33	.19
150	5.049	95.07	0.1981	4.730	95.06	0.1973	4.198	95.03	0.1959	3.773	95.00	0.19
160	5.134	96.76	.2008	4.810	96.75	.2000	4.270	96.72	.1987	3.837	96.69	.19
170	5.220	98.46	.2035	4.890	98.45	.2028	4.341	98.42	.2014	3.902	98.39	.20
180	5.305	100.17	.2062	4.970	100.16	.2055	4.412	100.13	.2041	3.966	100.10	.20
190	5.390	101.90	.2089	5.050	101.88	.2081	4.483	101.86	.2067	4.030	101.83	.20
200	5.475	103.64	0.2116	5.130	103.62	0.2108	4.555	103.60	0.2094	4.095	103.57	0.20
210	5.560	105.39	.2142	5.210	105.38	.2134	4.626	105.35	.2121	4.159	105.33	.21
220	5.645	107.16	.2168	5.290	107.15	.2161	4.697	107.12	.2147	4.223	107.10	.21
230	5.730	108.94	.2194	5.370	108.93	.2187	4.768	108.91	.2173	4.287	108.88	.21
240	5.815	110.74	.2220	5.449	110.73	.2213	4.839	110.70	.2199	4.351	110.68	.21
250	5.900	112.55	0.2246	5.529	112.54	0.2238	4.910	112.51	0.2224	4.415	112.49	0.22
260	5.985	114.37	.2271	5.609	114.36	.2264	4.981	114.33	.2250	4.479	114.31	.22
270	6.070	116.21	.2297	5.689	116.19	.2289	5.052	116.17	.2275	4.543	116.15	.22
280	6.155	118.06	.2322	5.768	118.05	.2314	5.123	118.02	.2300	4.607	118.00	.22
290	6.240	119.92	.2347	5.848	119.91	.2339	5.194	119.89	.2325	4.671	119.87	.23
300	6.325	121.80	0.2372	5.927	121.79	0.2364	5.265	121.77	0.2350	4.735	121.75	0.23
310				6.007	123.68	0.2389	5.336	123.66	0.2375	4.799	123.64	
320										4.862	125.55	

Table 23. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane— Properties of Superheated Vapor (Continued)

Temp F	Pressure 11 psia Gage pressure 7.53 in. vac (Sat temp 25.2 F)			Pressure 12 psia Gage pressure 5.50 in. vac (Sat temp 29.1 F)			Pressure 13 psia Gage pressure 3.46 in. vac (Sat temp 32.7 F)			Pressure 14 psia Gage pressure 1.42 in. vac (Sat temp 36.1 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
(t sat)	(2.686)	(75.05)	(0.1570)	(2.477)	(75.64)	(0.1572)	(2.298)	(76.11)	(0.1573)	(2.144)	(76.59)	(0.1575)
30	2.714	75.78	0.1585	2.481	75.74	0.1575						
40	2.774	77.31	.1616	2.536	77.27	.1606	2.335	77.23	0.1596	2.162	77.19	0.1587
50	2.834	78.85	0.1647	2.591	78.81	0.1636	2.386	78.77	0.1626	2.210	78.73	0.1617
60	2.893	80.40	.1677	2.646	80.36	.1666	2.436	80.32	.1656	2.257	80.29	.1647
70	2.952	81.96	.1707	2.700	81.93	.1696	2.487	81.89	.1686	2.304	81.86	.1677
80	3.012	83.54	.1736	2.755	83.51	.1726	2.538	83.47	.1716	2.351	83.44	.1707
90	3.071	85.13	.1765	2.809	85.10	.1755	2.588	85.07	.1745	2.399	85.03	.1736
100	3.130	86.74	0.1794	2.864	86.71	0.1784	2.639	86.67	0.1774	2.446	86.64	0.1765
110	3.189	88.36	.1823	2.918	88.33	.1813	2.689	88.29	.1803	2.493	88.26	.1794
120	3.248	89.99	.1851	2.972	89.96	.1841	2.739	89.93	.1831	2.539	89.90	.1822
130	3.307	91.64	.1879	3.027	91.61	.1869	2.789	91.58	.1859	2.586	91.55	.1850
140	3.366	93.30	.1907	3.081	93.27	.1897	2.839	93.24	.1887	2.633	93.21	.1878
150	3.425	94.97	0.1935	3.135	94.94	0.1925	2.889	94.91	0.1915	2.679	94.88	0.1906
160	3.483	96.66	.1963	3.189	96.63	.1952	2.939	96.60	.1943	2.726	96.57	.1934
170	3.542	98.36	.1990	3.243	98.33	.1979	2.989	98.31	.1970	2.772	98.28	.1961
180	3.601	100.08	.2017	3.297	100.05	.2006	3.039	100.02	.1997	2.818	99.99	.1988
190	3.659	101.81	.2044	3.351	101.78	.2033	3.089	101.75	.2024	2.865	101.72	.2015
200	3.718	103.55	0.2070	3.404	103.52	0.2060	3.139	103.49	0.2050	2.911	103.47	0.2041
210	3.776	105.30	.2097	3.458	105.28	.2086	3.188	105.25	.2077	2.957	105.23	.2068
220	3.835	107.07	.2123	3.512	107.05	.2113	3.238	107.02	.2103	3.004	107.00	.2094
230	3.893	108.86	.2149	3.565	108.83	.2139	3.288	108.81	.2129	3.050	108.78	.2120
240	3.952	110.65	.2175	3.619	110.63	.2164	3.337	110.61	.2155	3.096	110.58	.2146
250	4.010	112.46	0.2201	3.672	112.44	0.2190	3.387	112.42	0.2181	3.142	112.39	0.2172
260	4.068	114.29	.2226	3.726	114.27	.2216	3.436	114.24	.2206	3.188	114.22	.2197
270	4.127	116.13	.2251	3.780	116.11	.2241	3.486	116.08	.2232	3.234	116.06	.2223
280	4.185	117.98	.2277	3.833	117.96	.2266	3.535	117.93	.2257	3.280	117.91	.2248
290	4.243	119.85	.2302	3.887	119.82	.2291	3.585	119.80	.2282	3.326	119.78	.2273
300	4.301	121.73	0.2327	3.940	121.70	0.2316	3.634	121.68	0.2307	3.372	121.66	0.2298
310	4.359	123.62	.2351	3.993	123.60	.2341	3.683	123.58	.2332	3.418	123.56	.2323
320	4.417	125.53	0.2376	4.047	125.51	0.2366	3.733	125.49	.2356	3.464	125.47	.2347
330							3.782	127.41	0.2381	3.510	127.39	0.2372

Temp F	Pressure 16 psia Gage pressure 1.3 psig (Sat temp 42.3 F)			Pressure 18 psia Gage pressure 3.3 psig (Sat temp 48.0 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp 53.3 F)			Pressure 22 psia Gage pressure 7.3 psig (Sat temp 58.1 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
(t sat)	(1.892)	(77.47)	(0.1577)	(1.694)	(78.27)	(0.1580)	(1.534)	(79.00)	(0.1583)	(1.403)	(79.68)	(0.1585)
50	1.924	78.66	0.1601	1.701	78.58	0.1586						
60	1.965	80.21	.1631	1.738	80.14	.1616	1.557	80.06	0.1603	1.408	79.98	0.1591
70	2.007	81.78	.1661	1.776	81.71	.1646	1.591	81.64	.1633	1.439	81.56	.1621
80	2.049	83.37	.1690	1.813	83.30	.1676	1.624	83.22	.1663	1.470	83.15	.1651
90	2.090	84.96	.1720	1.850	84.90	.1705	1.658	84.82	.1692	1.501	84.75	.1681
100	2.131	86.57	0.1749	1.887	86.51	0.1734	1.692	86.44	0.1721	1.532	86.37	0.1709
110	2.173	88.20	.1778	1.924	88.13	.1763	1.725	88.07	.1750	1.562	88.00	.1738
120	2.214	89.83	.1806	1.961	89.77	.1792	1.759	89.71	.1779	1.593	89.64	.1767
130	2.255	91.48	.1834	1.998	91.42	.1820	1.792	91.36	.1807	1.623	91.30	.1795
140	2.296	93.15	.1862	2.035	93.09	.1848	1.825	93.03	.1835	1.654	92.97	.1823
150	2.337	94.83	0.1890	2.071	94.77	0.1876	1.858	94.71	0.1863	1.684	94.65	0.1851
160	2.378	96.52	.1918	2.108	96.46	.1903	1.891	96.40	.1890	1.714	96.34	.1879
170	2.419	98.22	.1945	2.144	98.16	.1931	1.924	98.11	.1918	1.745	98.05	.1906
180	2.460	99.94	.1972	2.181	99.88	.1958	1.957	99.83	.1945	1.775	99.77	.1933
190	2.500	100.67	.1999	2.217	101.62	.1984	1.990	101.56	.1972	1.805	101.51	.1960
200	2.541	103.41	0.2025	2.253	103.36	0.2011	2.023	103.31	0.1998	1.835	103.26	0.1987
210	2.582	105.17	.2052	2.290	105.12	.2038	2.056	105.07	.2025	1.865	105.02	.2013
220	2.622	106.95	.2078	2.326	106.90	.2064	2.089	106.84	.2051	1.895	106.79	.2040
230	2.663	108.73	.2104	2.362	108.68	.2090	2.122	108.63	.2077	1.925	108.58	.2066
240	2.704	110.53	.2130	2.398	110.48	.2116	2.155	110.43	.2103	1.955	110.39	.2092
250	2.744	112.35	0.2156	2.435	112.30	0.2142	2.187	112.25	0.2129	1.985	112.20	0.2117
260	2.785	114.17	.2181	2.471	114.13	.2167	2.220	114.08	.2155	2.014	114.03	.2143
270	2.825	116.01	.2207	2.507	115.97	.2193	2.252	115.92	.2180	2.044	115.88	.2168
280	2.866	117.87	.2232	2.543	117.82	.2218	2.285	117.78	.2205	2.074	117.73	.2194
290	2.906	119.74	.2257	2.579	119.69	.2243	2.317	119.65	.2230	2.104	119.60	.2219
300	2.946	121.62	0.2282	2.615	121.57	0.2268	2.350	121.53	0.2255	2.133	121.49	0.2244
310	2.986	123.51	.2307	2.651	123.47	.2293	2.382	123.43	.2280	2.163	123.39	.2269
320	3.027	125.42	.2332	2.687	125.38	.2317	2.415	125.34	.2305	2.192	125.30	.2293
330	3.067	127.35	.2356	2.723	127.30	.2342	2.447	127.26	.2329	2.222	127.22	.2318
340	3.107	129.28	0.2380	2.759	129.24	0.2366	2.480	129.20	.2354	2.252	129.16	.2342
350							2.512	131.15	0.2378	2.281	131.11	0.2367

Table 23. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane— Properties of Superheated Vapor (Continued)

Temp F	Pressure 24 psia Gage pressure 9.3 psig (Sat temp 62.6 F)			Pressure 26 psia Gage pressure 11.3 psig (Sat temp 66.8 F)			Pressure 28 psia Gage pressure 13.3 psig (Sat temp 70.8 F)			Pressure 30 psia Gage pressure 15.3 psig (Sat temp 74.6 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*1.292*)	(*80.31*)	(*0.1588*)	(*1.198*)	(*80.90*)	(*0.1590*)	(*1.117*)	(*81.46*)	(*0.1593*)	(*1.046*)	(*81.98*)	(*0.1595*)
70	1.313	81.49	0.1610	1.206	81.41	0.1600						
80	1.341	83.08	.1640	1.233	83.00	.1630	1.139	82.93	0.1620	1.058	82.85	0.1611
90	1.370	84.68	.1670	1.259	84.61	.1659	1.164	84.54	.1650	1.082	84.47	.1641
100	1.398	86.30	0.1699	1.286	86.23	0.1688	1.189	86.16	0.1679	1.105	86.09	0.1670
110	1.427	87.93	.1727	1.312	87.86	.1717	1.213	87.79	.1708	1.128	87.73	.1699
120	1.455	89.58	.1756	1.338	89.51	.1764	1.238	89.44	.1737	1.151	89.38	.1728
130	1.483	91.23	.1784	1.364	91.17	.1774	1.262	91.10	.1765	1.174	91.04	.1756
140	1.511	92.90	.1812	1.390	92.84	.1802	1.287	92.78	.1793	1.197	92.71	.1784
150	1.539	94.58	0.1840	1.416	94.52	0.1830	1.311	94.46	0.1821	1.220	94.40	0.1812
160	1.567	96.28	.1868	1.442	96.22	.1858	1.335	96.16	.1849	1.242	96.10	.1840
170	1.595	97.99	.1895	1.468	97.93	.1885	1.359	97.88	.1876	1.265	97.82	.1867
180	1.623	99.71	.1922	1.494	99.66	.1912	1.383	99.60	.1903	1.288	99.54	.1895
190	1.651	101.45	.1949	1.520	101.40	.1939	1.407	101.34	.1930	1.310	101.28	.1922
200	1.678	103.20	0.1976	1.545	103.15	0.1966	1.431	103.09	0.1957	1.333	103.04	0.1949
210	1.706	104.96	.2003	1.571	104.91	.1993	1.455	104.86	.1984	1.355	104.81	.1975
220	1.733	106.74	.2029	1.596	106.69	.2019	1.479	106.64	.2010	1.378	106.59	.2002
230	1.761	108.53	.2055	1.622	108.48	.2045	1.503	108.43	.2036	1.400	108.38	.2028
240	1.788	110.34	.2081	1.647	110.29	.2071	1.527	110.24	.2062	1.422	110.19	.2054
250	1.816	112.15	0.2107	1.673	112.11	0.2097	1.550	112.06	0.2088	1.444	112.01	0.2079
260	1.843	113.98	.2133	1.698	113.94	.2123	1.574	113.89	.2114	1.467	113.84	.2105
270	1.871	115.83	.2158	1.724	115.78	.2148	1.598	115.74	.2139	1.489	115.69	.2131
280	1.898	117.69	.2183	1.749	117.64	.2174	1.622	117.60	.2164	1.511	117.55	.2156
290	1.925	119.56	.2208	1.774	119.51	.2199	1.645	119.47	.2190	1.533	119.42	.2181
300	1.953	121.44	0.2233	1.800	121.40	0.2224	1.669	121.36	0.2215	1.555	121.31	0.2206
310	1.980	123.34	.2258	1.825	123.30	.2249	1.692	123.26	.2240	1.577	123.21	.2231
320	2.007	125.25	.2283	1.850	125.21	.2273	1.716	125.17	.2264	1.599	125.13	.2256
330	2.034	127.18	.2307	1.875	127.14	.2298	1.739	127.10	.2289	1.621	127.06	.2280
340	2.061	129.12	.2332	1.900	129.08	.2322	1.762	129.04	.2313	1.643	129.00	.2305
350	2.088	131.07	0.2356	1.925	131.04	0.2346	1.786	131.04	0.2337	1.665	130.96	0.2329
360	2.116	133.04	0.2380	1.951	133.01	0.2371	1.809	132.97	.2362	1.687	132.93	.2353
370							1.833	134.95	0.2386	1.708	134.91	0.2377

Temp F	Pressure 32 psia Gage pressure 17.3 psig (Sat temp 78.2 F)			Pressure 34 psia Gage pressure 19.3 psig (Sat temp 81.6 F)			Pressure 36 psia Gage pressure 21.3 psig (Sat temp 84.9 F)			Pressure 38 psia Gage pressure 23.3 psig (Sat temp 88.0 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*0.9836*)	(*82.49*)	(*0.1598*)	(*0.9284*)	(*82.96*)	(*0.1600*)	(*0.8792*)	(*83.41*)	(*0.1602*)	(*0.8347*)	(*83.84*)	(*0.1604*)
80	0.9874	82.78	0.1603									
90	1.010	84.39	.1633	0.9458	84.32	0.1625	0.8891	84.24	0.1617	0.8384	84.17	0.1610
100	1.032	86.02	0.1662	0.9666	85.95	0.1654	0.9089	85.87	0.1646	0.8573	85.80	0.1639
110	1.053	87.66	.1691	0.9873	87.59	.1683	.9286	87.52	.1675	.8761	87.44	.1668
120	1.075	89.31	.1720	1.008	89.24	.1712	.9482	89.17	.1704	.8948	89.10	.1697
130	1.097	90.97	.1748	1.028	90.91	.1740	.9677	90.84	.1733	.9134	90.77	.1726
140	1.118	92.65	.1776	1.049	92.59	.1768	0.9871	92.52	.1761	.9320	92.46	.1754
150	1.140	94.34	0.1804	1.069	94.28	0.1796	1.007	94.21	0.1789	0.9505	94.15	0.1782
160	1.161	96.04	.1832	1.090	95.98	.1824	1.026	95.92	.1817	.9689	95.86	.1810
170	1.183	97.76	.1859	1.110	97.70	.1852	1.045	97.64	.1844	0.9872	97.58	.1837
180	1.204	99.49	.1887	1.130	99.43	.1879	1.064	99.37	.1872	1.005	99.31	.1865
190	1.225	101.23	.1914	1.150	101.17	.1906	1.083	101.11	.1899	1.024	101.06	.1892
200	1.246	102.98	0.1940	1.170	102.93	0.1933	1.102	102.87	0.1926	1.042	102.82	0.1919
210	1.268	104.75	.1967	1.190	104.70	.1959	1.121	104.64	.1952	1.060	104.59	.1945
220	1.289	106.53	.1993	1.210	106.48	.1986	1.140	106.43	.1979	1.078	106.38	.1972
230	1.310	108.33	.2020	1.230	108.28	.2012	1.159	108.23	.2005	1.096	108.18	.1998
240	1.331	110.14	.2046	1.250	110.09	.2038	1.178	110.04	.2031	1.113	109.99	.2024
250	1.352	111.96	0.2072	1.269	111.91	0.2064	1.197	111.86	0.2057	1.131	111.81	0.2050
260	1.372	113.79	.2097	1.289	113.75	.2090	1.215	113.70	.2083	1.149	113.65	.2076
270	1.393	115.64	.2123	1.309	115.60	.2115	1.234	115.55	.2108	1.167	115.50	.2101
280	1.414	117.50	.2148	1.328	117.46	.2141	1.253	117.41	.2133	1.185	117.37	.2127
290	1.435	119.38	.2173	1.348	119.33	.2166	1.271	119.29	.2159	1.203	119.25	.2152
300	1.455	121.27	0.2198	1.368	121.22	0.2191	1.290	121.18	0.2184	1.220	121.14	0.2177
310	1.476	123.17	.2223	1.387	123.13	.2216	1.308	123.08	.2209	1.238	123.04	.2202
320	1.497	125.09	.2248	1.407	125.05	.2241	1.327	125.00	.2233	1.255	124.96	.2227
330	1.517	127.02	.2272	1.426	126.98	.2265	1.345	126.93	.2258	1.273	126.89	.2251
340	1.538	128.96	.2297	1.446	128.92	.2289	1.364	128.88	.2283	1.290	128.84	.2276
350	1.559	130.92	0.2321	1.465	130.88	0.2314	1.382	130.84	0.2307	1.308	130.80	0.2300
360	1.579	132.89	.2345	1.485	132.85	.2338	1.401	132.81	.2331	1.325	132.77	.2323
370	1.600	134.87	0.2370	1.504	134.83	.2362	1.419	134.79	.2355	1.343	134.75	.2348
380				1.524	136.83	0.2386	1.437	136.79	0.2379	1.360	136.75	0.2372

Table 23. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane— Properties of Superheated Vapor (Continued)

mp F	Pressure 40 psia Gage pressure 25.3 psig (Sat temp 91.0 F)			Pressure 45 psia Gage pressure 30.3 psig (Sat temp 98.1 F)			Pressure 50 psia Gage pressure 35.3 psig (Sat temp 104.6 F)			Pressure 55 psia Gage pressure 40.3 psig (Sat temp 110.7 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
sat)	*(0.7948)*	*(84.25)*	*(0.1606)*	*(0.7096)*	*(85.23)*	*(0.1611)*	*(0.6408)*	*(86.10)*	*(0.1616)*	*(0.5841)*	*(86.92)*	*(0.1620)*
100	0.8108	85.73	0.1632	0.7125	85.54	0.1616						
110	.8288	87.37	.1661	.7288	87.19	.1646	0.6487	87.00	0.1631			
120	.8467	89.03	.1690	.7450	88.86	.1675	.6636	88.68	.1660	0.5967	88.49	0.1647
130	.8646	90.70	.1719	.7611	90.53	.1703	.6784	90.36	.1689	.6104	90.18	.1676
140	.8824	92.39	.1747	.7772	92.22	.1732	.6931	92.05	.1718	.6240	91.88	.1705
150	0.9001	94.09	0.1775	0.7932	93.92	0.1760	0.7077	93.76	0.1746	0.6375	93.59	0.1733
160	.9176	95.80	.1803	.8091	95.64	.1788	.7223	95.48	.1774	.6509	95.31	.1761
170	.9351	97.52	.1831	.8250	97.37	.1816	.7367	97.21	.1802	.6643	97.05	.1789
180	.9524	99.25	.1858	.8407	99.11	.1843	.7511	98.95	.1829	.6776	98.80	.1816
190	.9697	101.00	.1885	.8563	100.86	.1870	.7654	100.71	.1856	.6908	100.56	.1844
200	0.9870	102.76	0.1912	0.8719	102.62	0.1897	0.7796	102.48	0.1883	0.7039	102.33	0.1871
210	1.004	104.53	.1939	.8874	104.04	.1924	.7937	104.26	.1910	.7170	104.12	.1898
220	1.021	106.32	.1965	.9028	106.19	.1950	.8077	106.05	.1937	.7300	105.91	.1924
230	1.039	108.12	.1992	.9182	107.99	.1977	.8217	107.86	.1963	.7429	107.72	.1951
240	1.056	109.93	.2018	.9335	109.81	.2003	.8357	109.68	.1989	.7558	109.55	.1977
250	1.073	111.76	0.2044	0.9488	111.64	0.2029	0.8496	111.51	0.2015	0.7686	111.38	0.2003
260	1.090	113.60	.2069	.9641	113.48	.2055	.8635	113.35	.2041	.7813	113.23	.2029
270	1.107	115.45	.2095	.9793	115.33	.2080	.8773	115.21	.2067	.7940	115.09	.2054
280	1.124	117.32	.2120	0.9945	117.20	.2106	.8911	117.08	.2092	.8066	116.97	.2080
290	1.141	119.20	.2146	1.010	119.08	.2131	.9049	118.97	.2118	.8192	118.85	.2105
300	1.157	121.09	0.2171	1.025	120.98	0.2156	0.9186	120.87	0.2143	0.8318	120.75	0.2130
310	1.174	123.00	.2196	1.040	122.89	.2181	.9323	122.78	.2168	.8444	122.67	.2155
320	1.191	124.92	.2220	1.055	124.81	.2206	.9460	124.70	.2193	.8569	124.59	.2180
330	1.208	126.85	.2245	1.070	126.75	.2230	.9597	126.64	.2217	.8694	126.53	.2205
340	1.224	128.80	.2270	1.085	128.70	.2255	.9733	128.59	.2242	.8819	128.49	.2230
350	1.241	130.76	0.2294	1.100	130.66	0.2279	0.9868	130.55	0.2266	0.8943	130.45	0.2254
360	1.258	132.73	.2318	1.115	132.63	.2304	1.000	132.53	.2290	.9067	132.43	.2278
370	1.274	134.72	.2342	1.130	134.62	.2328	1.014	134.52	.2315	.9191	134.42	.2303
380	1.291	136.72	.2366	1.144	136.62	.2352	1.027	136.53	.2339	.9314	136.43	.2327
390	1.308	138.73	0.2390	1.159	138.63	0.2376	1.041	138.54	.2362	.9437	138.45	.2351
400							1.054	140.57	0.2386	0.9560	140.48	0.2374
410										0.9683	142.53	0.2398

emp F	Pressure 60 psia Gage pressure 45.3 psig (Sat temp 116.3 F)			Pressure 65 psia Gage pressure 50.3 psig (Sat temp 121.6 F)			Pressure 70 psia Gage pressure 55.3 psig (Sat temp 126.6 F)			Pressure 75 psia Gage pressure 60.3 psig (Sat temp 131.4 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
t sat)	*(0.5364)*	*(87.67)*	*(0.1624)*	*(0.4958)*	*(88.37)*	*(0.1628)*	*(0.4607)*	*(89.03)*	*(0.1632)*	*(0.4302)*	*(89.65)*	*(0.1635)*
120	0.5411	88.30	0.1635									
130	.5539	90.00	.1664	0.5059	89.81	0.1652	0.4645	89.62	0.1642			
140	.5666	91.70	.1693	.5179	91.52	.1681	.4759	91.34	.1670	0.4395	91.15	0.1660
150	0.5792	93.42	0.1721	0.5298	93.25	0.1710	0.4873	93.07	0.1699	0.4502	92.89	0.1689
160	.5918	95.15	.1749	.5415	94.98	.1738	.4985	94.81	.1727	.4608	94.63	.1717
170	.6043	96.89	.1777	.5532	96.73	.1766	.5095	96.56	.1755	.4713	96.39	.1745
180	.6166	98.64	.1805	.5648	98.49	.1794	.5204	98.33	.1783	.4817	98.16	.1773
190	.6288	100.41	.1832	.5763	100.26	.1821	.5312	100.10	.1811	.4920	99.94	.1801
200	0.6410	102.18	0.1859	0.5876	102.04	0.1848	0.5419	101.88	0.1838	0.5022	101.73	0.1828
210	.6530	103.97	.1886	.5989	103.83	.1875	.5525	103.68	.1865	.5122	103.53	.1855
220	.6650	105.78	.1913	.6101	105.63	.1902	.5630	105.49	.1892	.5222	105.35	.1882
230	.6770	107.59	.1939	.6213	107.45	.1928	.5735	107.31	.1918	.5322	107.17	.1909
240	.6889	109.41	.1965	.6324	109.28	.1955	.5840	109.15	.1945	.5421	109.01	.1935
250	0.7008	111.25	0.1991	0.6435	111.12	0.1981	0.5944	110.99	0.1971	0.5519	110.86	0.1962
260	.7126	113.10	.2017	.6545	112.98	.2007	.6048	112.85	.1997	.5617	112.72	.1988
270	.7244	114.97	.2043	.6655	114.85	.2033	.6151	114.72	.2023	.5715	114.59	.2014
280	.7362	116.85	.2069	.6765	116.73	.2058	.6254	116.60	.2048	.5812	116.48	.2040
290	.7479	118.74	.2094	.6874	118.62	.2084	.6356	118.50	.2074	.5908	118.38	.2065
300	0.7595	120.64	0.2119	0.6983	120.52	0.2109	0.6458	120.41	0.2099	0.6004	120.29	0.2090
310	.7711	122.56	.2144	.7091	122.44	.2134	.6559	122.33	.2124	.6099	122.22	.2115
320	.7827	124.49	.2169	.7199	124.37	.2159	.6660	124.26	.2149	.6194	124.15	.2140
330	.7943	126.43	.2194	.7307	126.32	.2184	.6761	126.21	.2174	.6289	126.10	.2165
340	.8058	128.38	.2219	.7414	128.28	.2208	.6862	128.17	.2199	.6383	128.06	.2190

Table 23. F-114 ($CClF_2$-$CClF_2$) Dichlorotetrafluoroethane—Properties of Superheated Vapor (Concluded)

Temp. F	Presssure 60 psia Gage pressure 15.3 psig (Sat temp 116.3 F)			Pressure 65 psia Gage pressure 50.3 psig (Sat temp 121.6 F)			Pressure 70 psia Gage pressure 55.3 psig (Sat temp 126.6 F)			Pressure 75 psia Gage pressure 60.3 psig (Sat temp 131.4 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*0.5364*)	(*87.67*)	(*0.1624*)	(*0.4958*)	(*88.37*)	(*0.1628*)	(*0.4607*)	(*89.08*)	(*0.1632*)	(*0.4302*	(*89.65*)	(*0.163*
350	0.8173	130.35	0.2243	0.7521	130.25	0.2233	0.6962	130.14	0.2223	0.6477	130.04	0.221
360	.8287	132.33	.2267	.7627	132.23	.2257	.7062	132.13	.2248	.6571	132.03	.223
370	.8401	134.33	.2292	.7733	134.23	.2281	.7161	134.13	.2272	.6664	134.03	.226
380	.8515	136.34	.2316	.7839	136.24	.2305	.7260	136.14	.2296	.6757	136.04	.228
390	.8629	138.36	.2340	.7945	138.26	.2329	.7359	138.17	.2320	.6850	138.07	.231
400	0.8742	140.39	0.2363	0.8050	140.30	0.2353	0.7458	140.21	0.2344	0.6943	140.11	0.233
410	0.8855	142.44	0.2387	.8155	142.35	.2377	.7556	142.26	.2368	.7036	142.16	.235
420				0.8260	144.41	0.2401	0.7654	144.32	0.2391	.7128	144.23	.238
430										0.7220	146.31	0.240

Temp F	Pressure 80 psia Gage pressure 65.3 psig (Sat temp 135.9 F)			Pressure 85 psia Gage pressure 70.3 psig (Sat temp 140.2 F)		
t	*v*	*h*	*s*	*v*	*h*	*s*
(*at sat*)	(*0.4033*)	(*90.24*)	(*0.1638*)	(*0.3794*)	(*90.78*)	(*0.1641*)
140	0.4076	90.96	0.1650	—	—	—
150	0.4178	92.70	0.1679	0.3890	92.51	0.1670
160	.4279	94.46	.1708	.3988	94.27	.1699
170	.4379	96.22	.1736	.4084	96.04	.1727
180	.4478	97.99	.1764	.4179	97.83	.1755
190	.4576	99.78	.1792	.4273	99.62	.1783
200	0.4674	101.58	0.1819	0.4366	101.42	0.1811
210	.4770	103.38	.1846	.4458	103.23	.1838
220	.4865	105.20	.1873	.4548	105.05	.1865
230	.4959	107.03	.1900	.4638	106.89	.1892
240	.5053	108.87	.1927	.4728	108.73	.1918
250	0.5146	110.72	0.1953	0.4817	110.59	0.1944
260	.5239	112.59	.1979	.4905	112.46	.1971
270	.5331	114.47	.2005	.4993	114.34	.1996
280	.5423	116.36	.2031	.5081	116.23	.2022
290	.5514	118.26	.2056	.5168	118.14	.2048
300	0.5605	120.17	0.2081	0.5254	120.05	0.2073
310	.5695	122.10	.2107	.5340	121.98	.2099
320	.5785	124.04	.2132	.5426	123.93	.2124
330	.5875	125.99	.2157	.5511	125.88	.2149
340	.5964	127.96	.2181	.5595	127.85	.2173
350	0.6053	129.93	0.2206	0.5679	129.83	0.2198
360	.6142	131.92	.2230	.5763	131.82	.2222
370	.6230	133.93	.2255	.5847	133.83	.2247
380	.6318	135.94	.2279	.5930	135.85	.2271
390	.6406	137.97	.2303	.6013	137.88	.2295
400	0.6494	140.02	0.2327	0.6096	139.92	0.2319
410	.6581	142.07	.2351	.6179	141.98	.2343
420	.6668	144.14	.2374	.6262	144.05	.2366
430	.6775	146.22	.2398	.6344	146.13	.2390
440	—	—	—	.6426	148.23	.2413

Table 24. Isobutane—Properties of Liquid and Saturated Vapor

Temp F	Pressure†		Liquid, density**	Vapor, sp vol**	Enthalpy, datum 0 F Btu per lb		Entropy, datum 0 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−20	7.50	14.6*	38.35	11.0	−9.0	156.5	−0.020	0.356
−15	8.30	13.0*	38.15	9.90	−7.0	157.0	− .015	.354
−10	9.28	11.0*	37.95	8.91	−4.5	158.5	− .010	.353
− 5	10.4	8.8*	37.80	7.99	−2.5	159.5	− .005	.351
0	11.6	6.3*	37.60	7.17	0.0	160.5	0.000	0.350
+ 5	13.1	3.3*	37.40	6.41	+2.5	162.0	0.005	0.348
10	14.6	0.2*	37.20	5.75	5.0	163.5	.011	.348
15	16.3	1.6	37.00	5.18	7.5	164.5	.016	.347
20	18.2	3.5	36.80	4.68	10.0	166.0	.021	.346
25	20.2	5.5	36.60	4.24	13.0	167.5	.027	.346
30	22.3	7.6	36.40	3.86	15.5	169.0	0.032	0.346
35	24.6	9.9	36.20	3.52	18.0	170.5	.038	.346
40	26.9	12.2	36.00	3.22	21.0	172.0	.044	.346
45	29.5	14.8	35.80	2.96	24.0	174.0	.049	.346
50	32.5	17.8	35.60	2.71	27.0	175.5	.055	.346
55	35.5	20.8	35.40	2.49	30.0	177.5	0.061	0.347
60	38.7	24.0	35.20	2.28	33.0	179.0	.067	.348
65	42.2	27.5	35.00	2.10	36.5	181.0	.073	.349
70	45.8	31.1	34.80	1.94	39.5	183.0	.079	.350
75	49.7	35.0	34.60	1.79	43.0	185.0	.086	.351
80	53.9	39.2	34.35	1.66	46.5	187.0	0.092	0.352
85	58.6	43.9	34.10	1.54	50.0	189.0	.098	.353
90	63.3	48.6	33.90	1.42	53.5	191.0	.105	.356
95	68.4	53.7	33.70	1.32	57.5	193.5	.112	.358
100	73.7	59.0	33.45	1.23	61.0	195.5	.118	.359
105	79.3	64.6	33.25	1.14	65.0	198.0	0.125	0.360
110	85.1	70.4	33.00	1.07	69.0	200.0	.132	.362
115	91.4	76.7	32.80	0.990	73.0	202.5	.139	.364
120	98.0	83.3	32.50	.926	77.0	204.5	.147	.367
125	104.8	90.1	32.30	.867	81.5	207.5	.154	.369
130	112.0	97.3	32.00	0.811	86.0	209.0	0.161	0.371
135	119.3	104.6	31.80	.760	90.5	211.0	.169	.375
140	126.8	112.1	31.50	.710	95.0	213.2	.176	.377
145	136.0	121.3	31.30	.662	99.6	215.2	.183	.379
150	145.0	130.3	31.03	.620	104.5	217.5	.190	.382
155	155.0	140.3	30.73	0.580	109.3	219.7	0.197	0.384
160	165.0	150.3	30.43	.542	114.2	221.7	.204	.387
165	175.0	160.3	30.14	.505	119.3	224.0	.212	.389
170	186.0	171.3	29.85	.475	124.7	225.7	.219	.392
175	198.0	183.3	29.54	.448	129.9	228.0	.226	.395
180	210.0	195.3	29.23	0.420	135.0	230.0	0.233	0.397

† Based on material from Dana, Jenkins, Burdick and Timm, published originally in *Refrigerating Engineering*, June 1926, vol. 12, no. 12, page 402.

* Inches of mercury below one standard atmosphere (29.92 in.).

** Specific volume and density beyond 135 F, B. H. Sage and W. N. Lacey, *Industrial and Engineering Chemistry*, June 1938.

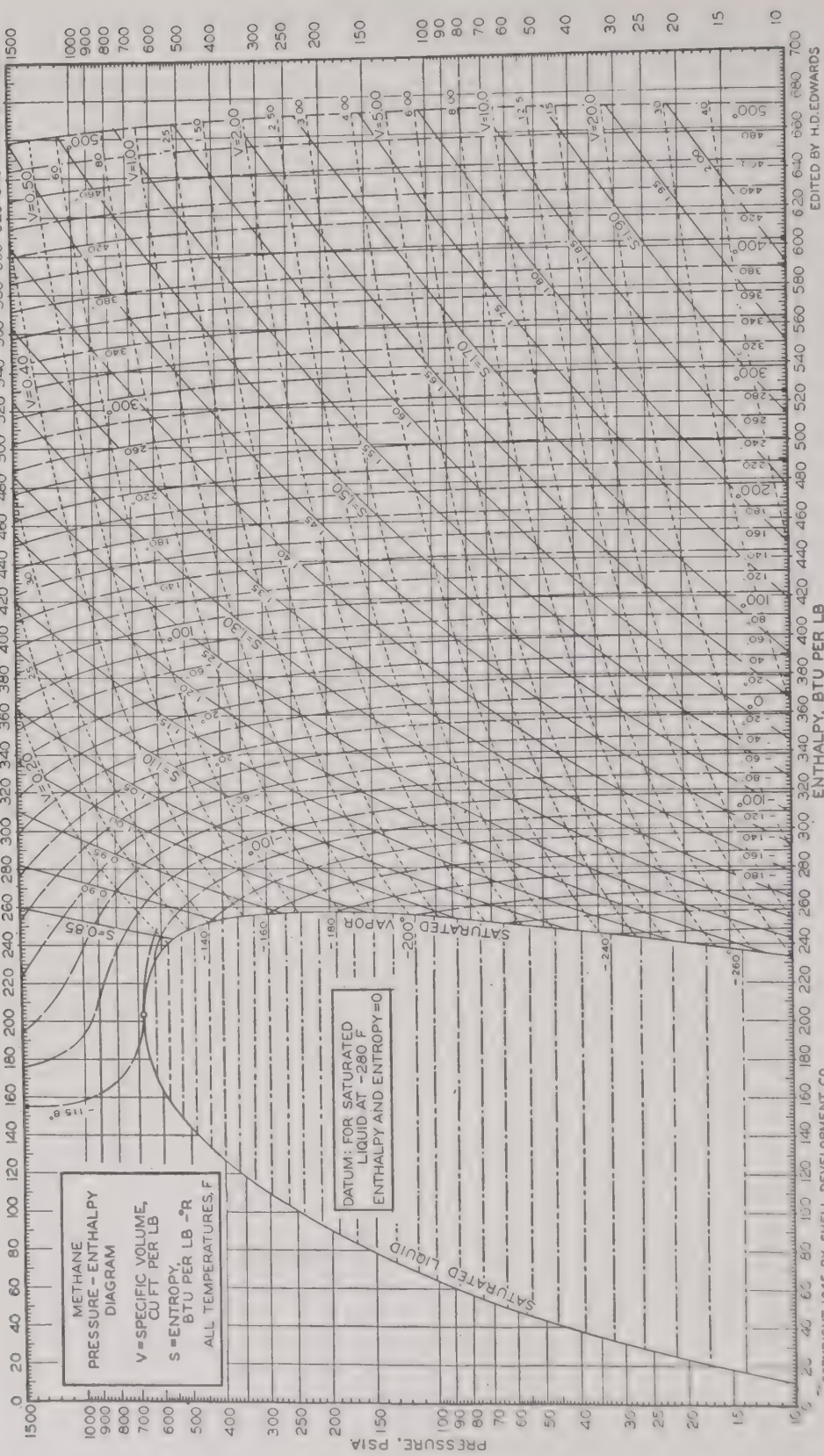

METHANE
PRESSURE - ENTHALPY
DIAGRAM
V = SPECIFIC VOLUME, CU FT PER LB
S = ENTROPY, BTU PER LB - °R
ALL TEMPERATURES, F
DATUM: FOR SATURATED LIQUID AT -280 F ENTHALPY AND ENTROPY = 0
SATURATED VAPOR
SATURATED LIQUID
PRESSURE, PSIA
ENTHALPY, BTU PER LB.
EDITED BY H.D. EDWARDS
COPYRIGHT 1945 BY SHELL DEVELOPMENT CO.

Chart 13

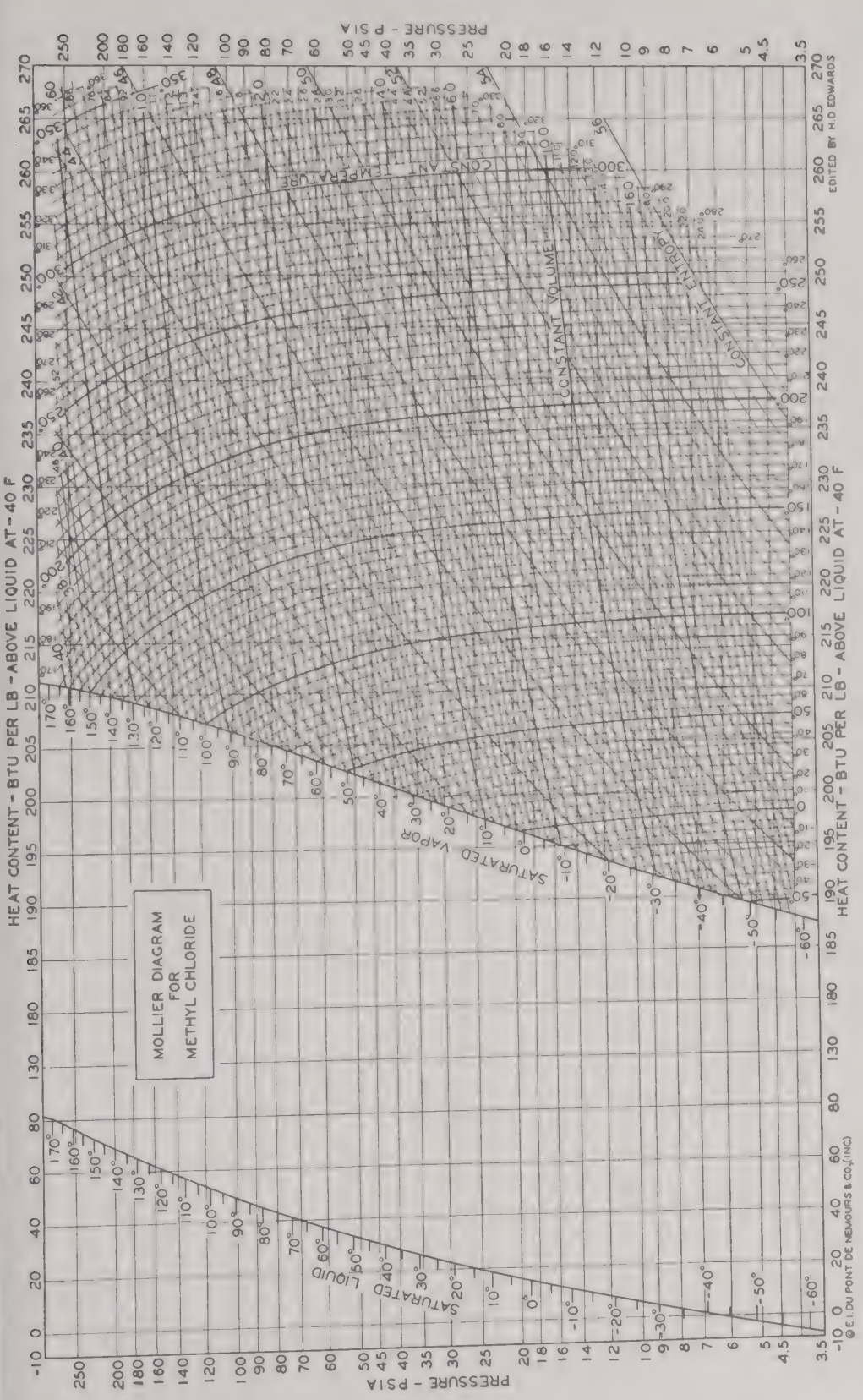

MOLLIER DIAGRAM FOR METHYL CHLORIDE
HEAT CONTENT - BTU PER LB - ABOVE LIQUID AT - 40 F
PRESSURE - P S I A
CONSTANT TEMPERATURE
CONSTANT VOLUME
CONSTANT ENTROPY
SATURATED LIQUID
SATURATED VAPOR
EDITED BY H.O. EDWARDS
© E. I. DU PONT DE NEMOURS & CO.(INC)

Chart 14

Table 25. Methyl Chloride—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F	
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−80	1.953	25.94*	0.01493	41.08	−13.888	184.75	−0.0351	0.4882
−70	2.715	24.32*	.01508	29.84	−10.521	186.25	− .0261	.4790
−60	3.799	22.19*	.01523	22.09	− 7.039	187.74	− .0172	.4703
−50	5.155	19.43*	.01538	16.64	− 3.532	189.19	− .0085	.4620
−40	6.878	15.92*	0.01553	12.72	0.000	190.66	0.0000	0.4544
−30	9.036	11.52*	.01568	9.873	3.562	192.08	.0084	.4472
−20	11.71	6.090*	.01583	7.761	7.146	193.49	.0166	.4405
−10	14.96	0.266	.01598	6.176	10.75	194.87	.0247	.4343
0	18.90	4.201	0.01613	4.969	14.39	196.23	0.0327	0.4284
2	19.77	5.077	.01616	4.763	15.12	196.51	.0343	.4273
4	20.68	5.985	.01619	4.568	15.85	196.78	.0359	.4262
5	21.15	6.455	.01622	4.471	16.21	196.92	.0367	.4257
6	21.62	6.924	.01625	4.379	16.58	197.05	.0375	.4251
8	22.59	7.896	.01628	4.206	17.31	197.31	.0390	.4240
10	23.60	8.903	0.01631	4.038	18.04	197.58	0.0406	0.4229
12	24.64	9.943	.01634	3.878	18.77	197.83	.0422	.4218
14	25.72	11.02	.01637	3.726	19.51	198.09	.0437	.4208
16	26.83	12.13	.01640	3.581	20.25	198.34	.0453	.4198
18	27.97	13.28	.01644	3.443	20.98	198.59	.0468	.4187
20	29.16	14.46	0.01647	3.312	21.73	198.84	0.0484	0.4177
22	30.38	15.69	.01650	3.186	22.47	199.08	.0499	.4166
24	31.64	16.95	.01654	3.067	23.21	199.32	.0514	.4156
26	32.95	18.25	.01658	2.952	23.95	199.56	.0530	.4146
28	34.29	19.60	.01662	2.843	24.70	199.79	.0545	.4136
30	35.68	20.98	0.01665	2.739	25.44	200.03	0.0560	0.4126
32	37.11	22.41	.01669	2.640	26.18	200.26	.0575	.4117
34	38.58	23.88	.01673	2.546	26.93	200.49	.0590	.4107
36	40.09	25.39	.01677	2.455	27.67	200.72	.0605	.4098
38	41.65	26.95	.01681	2.369	28.42	200.95	.0621	.4088
40	43.25	28.56	0.01684	2.286	29.17	201.17	0.0636	0.4079
42	44.91	30.21	.01688	2.206	29.92	201.40	.0651	.4070
44	46.61	31.91	.01692	2.130	30.67	201.62	.0665	.4061
46	48.35	33.66	.01696	2.057	31.42	201.84	.0680	.4052
48	50.15	35.45	.01700	1.987	32.17	202.06	.0695	.4043
50	51.99	37.29	0.01704	1.920	32.93	202.28	0.0710	0.4034
52	53.88	39.18	.01708	1.856	33.68	202.49	.0725	.4025
54	55.83	41.13	.01712	1.794	34.44	202.71	.0740	.4017
56	57.83	43.13	.01716	1.735	35.19	202.91	.0754	.4008
58	59.88	45.19	.01720	1.679	35.95	203.13	.0769	.3999
60	62.00	47.30	0.01724	1.624	36.71	203.33	0.0784	0.3991
62	64.17	49.47	.01728	1.572	37.47	203.54	.0798	.3983
64	66.39	51.70	.01732	1.522	38.23	203.74	.0813	.3974
66	68.67	53.98	.01736	1.473	39.00	203.95	.0827	.3966
68	71.01	56.32	.01740	1.427	39.76	204.15	.0842	.3958
70	73.41	58.71	0.01744	1.382	40.52	204.34	0.0856	0.3950
72	75.86	61.17	.01748	1.339	41.29	204.53	.0870	.3941
74	78.37	63.68	.01752	1.298	42.06	204.72	.0885	.3933
76	80.94	66.25	.01756	1.258	42.82	204.90	.0899	.3925
78	83.57	68.87	.01760	1.220	43.59	205.09	.0913	.3918
80	86.26	71.56	0.01764	1.183	44.36	205.27	0.0928	0.3910
82	89.01	74.31	.01768	1.148	45.13	205.45	.0942	.3902
84	91.82	77.13	.01773	1.114	45.90	205.62	.0956	.3894
86	94.70	80.00	.01778	1.081	46.67	205.80	.0970	.3887
88	97.64	82.94	.01782	1.049	47.44	205.96	.0984	.3879
90	100.6	85.95	0.01786	1.018	48.21	206.13	0.0998	0.3872
92	103.7	89.02	.01791	.9889	48.99	206.30	.1012	.3867
94	106.9	92.16	.01796	.9603	49.77	206.46	.1026	.3857
96	110.1	95.37	.01800	.9333	50.54	206.62	.1041	.3850
98	113.4	98.65	.01804	.9069	51.32	206.78	.1055	.3843
100	116.7	102.0	0.01808	0.8814	52.09	206.94	0.1069	0.3836
120	154.2	139.5	.01859	.6710	59.93	208.39	.1206	.3768
140	199.6	184.9	.01915	.5189	67.87	209.58	.1341	.3705
160	253.5	238.8	.01978	.4070	75.90	210.56	.1473	.3646

* Inches of mercury below one atmosphere.

Table 26. Methyl Chloride—Properties of Superheated Vapor

emp F	Pressure 8 psia (Sat temp −34.5 F)			Pressure 12 psia (Sat temp −19.0 F)			Pressure 16 psia (Sat temp −7.2 F)			Pressure 20 psia (Sat temp 2.5 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
sat)	*(11.06)*	*(191.44)*	*(0.4604)*	*(7.587)*	*(193.64)*	*(0.4399)*	*(5.804)*	*(196.28)*	*(0.4326)*	*(4.710)*	*(197.58)*	*(0.4270)*
0	12.01	197.58	0.4643	7.943	197.06	0.4476	5.907	196.54	0.4355			
10	12.29	199.42	.4682	8.131	198.94	.4516	6.050	198.45	.4396	4.801	197.95	.4300
20	12.57	201.26	.4721	8.318	200.82	.4555	6.192	200.37	.4435	4.917	199.90	.4341
30	12.84	203.12	.4760	8.503	202.70	.4594	6.333	202.27	.4474	5.032	201.82	.4380
40	13.11	204.97	.4798	8.688	204.58	.4632	6.473	204.18	.4513	5.146	203.75	.4420
50	13.39	206.90	0.4836	8.873	206.51	0.4670	6.613	206.10	.4552	5.260	205.71	0.4458
60	13.66	208.82	.4873	9.056	208.43	.4708	6.753	208.85	0.4589	5.373	207.66	.4496
65	13.79	209.79	.4891	9.147	209.41	.4726	6.823	209.04	.4608	5.429	208.66	.4515
70	13.93	210.76	.4910	9.239	210.39	.4745	6.893	210.03	.4627	5.486	209.66	.4534
80	14.21	212.70	.4946	9.422	212.35	.4782	7.033	212.00	.4664	5.599	211.65	.4572
90	14.48	214.68	.4982	9.605	214.34	.4818	7.173	214.01	.4701	5.711	213.67	.4608
100	14.75	216.66	0.5018	9.788	216.34	0.4855	7.311	216.01	.4737	5.823	215.69	0.4645
110	15.02	218.68	.5054	9.971	218.37	.4890	7.449	218.06	0.4773	5.935	217.75	.4681
120	15.29	220.70	.5089	10.15	220.40	.4926	7.587	220.10	.4809	6.046	219.80	.4717
130	15.56	222.75	.5124	10.34	222.47	.4961	7.725	222.18	.4844	6.157	221.90	.4753
140	15.83	224.81	.5159	10.52	224.54	.4996	7.863	224.26	.4879	6.268	223.99	.4788
150	16.10	226.90	0.5193	10.70	226.64	0.5030	8.001	226.83	.4914	6.379	226.12	0.4823
160	16.37	228.99	.5227	10.88	228.74	.5065	8.137	228.49	0.4949	6.489	228.34	.4858
170	16.64	231.12	.5261	11.06	230.88	.5099	8.273	230.64	.4983	6.599	230.40	.4892
180	16.91	233.25	.5295	11.24	233.02	.5133	8.409	232.79	.5017	6.709	232.56	.4927
190	17.18	235.41	.5329	11.42	235.19	.5166	8.545	234.97	.5051	6.819	234.75	.4961

emp F	Pressure 30 psia (Sat temp 21.4 F)			Pressure 40 psia (Sat temp 35.9 F)			Pressure 50 psia (Sat temp 47.8 F)			Pressure 60 psia (Sat temp 58.1 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
t sat)	*(3.224)*	*(199.00)*	*(0.4169)*	*(2.461)*	*(200.73)*	*(0.4097)*	*(1.992)*	*(202.09)*	*(0.4043)*	*(1.676)*	*(203.17)*	*(0.3998)*
60	3.532	206.70	.4324	2.608	205.69	.4196	2.054	204.65	.4049	1.684	203.49	.4008
65	3.571	207.71	.4343	2.638	206.73	.4216	2.079	205.71	.4114	1.705	204.60	.4028
70	3.610	208.73	.4362	2.668	207.77	.4236	2.104	206.77	.4134	1.727	205.71	.4049
80	3.687	210.77	.4400	2.729	209.84	.4275	2.154	208.59	.4174	1.770	207.92	.4089
90	3.764	212.82	.4438	2.788	211.94	.4313	2.203	211.03	.4213	1.812	210.11	.4129
100	3.840	214.87	0.4475	2.847	214.03	0.4351	2.252	213.18	0.4252	1.854	212.30	0.4169
110	3.916	216.96	.4512	2.906	216.17	.4389	2.300	215.35	.4290	1.895	214.50	.4208
120	3.992	219.06	.4548	2.964	218.32	.4426	2.348	217.62	.4328	1.936	216.69	.4246
130	4.067	221.18	.4585	3.022	220.46	.4463	2.396	219.70	.4366	1.977	218.93	.4283
140	4.142	223.30	.4620	3.080	222.60	.4499	2.443	221.88	.4402	2.017	221.16	.4322
150	4.217	225.45	0.4656	3.138	224.78	0.4535	2.490	224.10	0.4439	2.057	223.41	0.4359
160	4.292	227.61	.4691	3.195	226.96	.4570	2.537	226.32	.4475	2.097	225.66	.4396
170	4.367	229.79	.4726	3.252	229.18	.4606	2.584	228.55	.4511	2.137	227.92	.4432
180	4.442	231.97	.4761	3.309	231.38	.4641	2.630	230.79	.4546	2.176	230.19	.4468
190	4.517	234.19	.4795	3.366	233.62	.4675	2.676	233.05	.4581	2.215	232.48	.4505
200	4.591	236.40	0.4829	3.423	235.86	0.4710	2.722	235.32	0.4616	2.254	234.77	0.4538
210	4.665	238.65	.4862	3.479	238.13	.4744	2.768	237.61	.4650	2.293	237.08	.4573
220	4.739	240.90	.4896	3.535	240.40	.4778	2.813	239.90	.4684	2.331	239.40	.4608
230	4.813	243.20	.4929	3.591	242.72	.4811	2.853	242.24	.4718	2.369	241.75	.4642
240	4.887	245.50	.4962	3.647	245.04	.4845	2.903	244.58	.4752	2.407	244.11	.4676

emp F	Pressure 80 psia (Sat temp 75.3 F)			Pressure 100 psia (Sat temp 89.6 F)			Pressure 120 psia (Sat temp 101.9 F)			Pressure 140 psia (Sat temp 112.8 F)		
t	v	h	s	v	h	s	v	h	s	v	h	s
t sat)	*(1.272)*	*(205.84)*	*(0.3928)*	*(1.025)*	*(206.11)*	*(0.3872)*	*(0.858)*	*(208.10)*	*(0.3827)*	*(0.838)*	*(207.91)*	*(0.3790)*
150	1.516	222.00	0.4229	1.191	220.55	0.4124	0.974	219.04	0.4034	0.818	217.48	0.3954
160	1.547	224.32	.4267	1.217	222.94	.4163	0.996	221.51	.4074	.838	220.03	.3995
170	1.578	226.64	.4304	1.243	225.33	.4201	1.019	223.97	.4113	.858	222.56	.4036
180	1.608	228.96	.4341	1.268	227.71	.4239	1.041	226.42	.4152	.877	225.09	.4076
190	1.638	231.30	.4377	1.293	230.10	.4276	1.062	228.88	.4190	.896	227.61	.4115
200	1.668	233.65	0.4413	1.318	232.50	0.4312	1.083	231.33	0.4228	.915	230.13	.4154
210	1.698	236.01	.4448	1.343	234.91	.4349	1.104	233.79	.4265	.934	232.65	.4192
220	1.728	238.37	.4483	1.367	237.32	.4384	1.125	236.25	.4301	.953	235.17	.4229
230	1.758	240.77	.4518	1.391	239.76	.4420	1.146	238.74	.4337	.971	237.70	.4266
240	1.788	243.16	.4552	1.415	242.20	.4455	1.167	241.22	.4373	.989	240.23	.4302
250	1.817	245.56	0.4586	1.439	244.63	0.4489	1.187	243.69	0.4408	1.007	242.73	0.4338
260	1.846	247.95	.4620	1.463	247.06	.4523	1.207	246.16	.4443	1.024	245.24	.4373
270	1.875	250.37	.4654	1.487	249.51	.4557	1.227	248.64	.4477	1.042	247.76	.4407
280	1.904	252.79	.4687	1.511	251.96	.4591	1.247	251.12	.4511	1.060	250.28	.4442
290	1.933	255.24	.4720	1.534	254.44	.4624	1.267	253.64	.4545	1.077	252.82	.4476
300	1.961	257.69	0.4752	1.557	256.92	0.4657	1.287	256.15	0.4578	1.094	255.36	0.4510
310	1.989	260.17	.4784	1.580	259.43	.4689	1.307	258.67	.4611	1.111	257.91	.4543
320	2.017	262.64	.4816	1.603	261.93	.4722	1.327	261.20	.4643	1.128	260.46	.4576
330	2.045	265.14	.4848	1.626	264.45	.4754	1.346	263.75	.4676	1.145	263.04	.4609
340	2.073	267.64	.4879	1.649	266.97	.4786	1.365	266.29	.4708	1.162	265.61	.4641

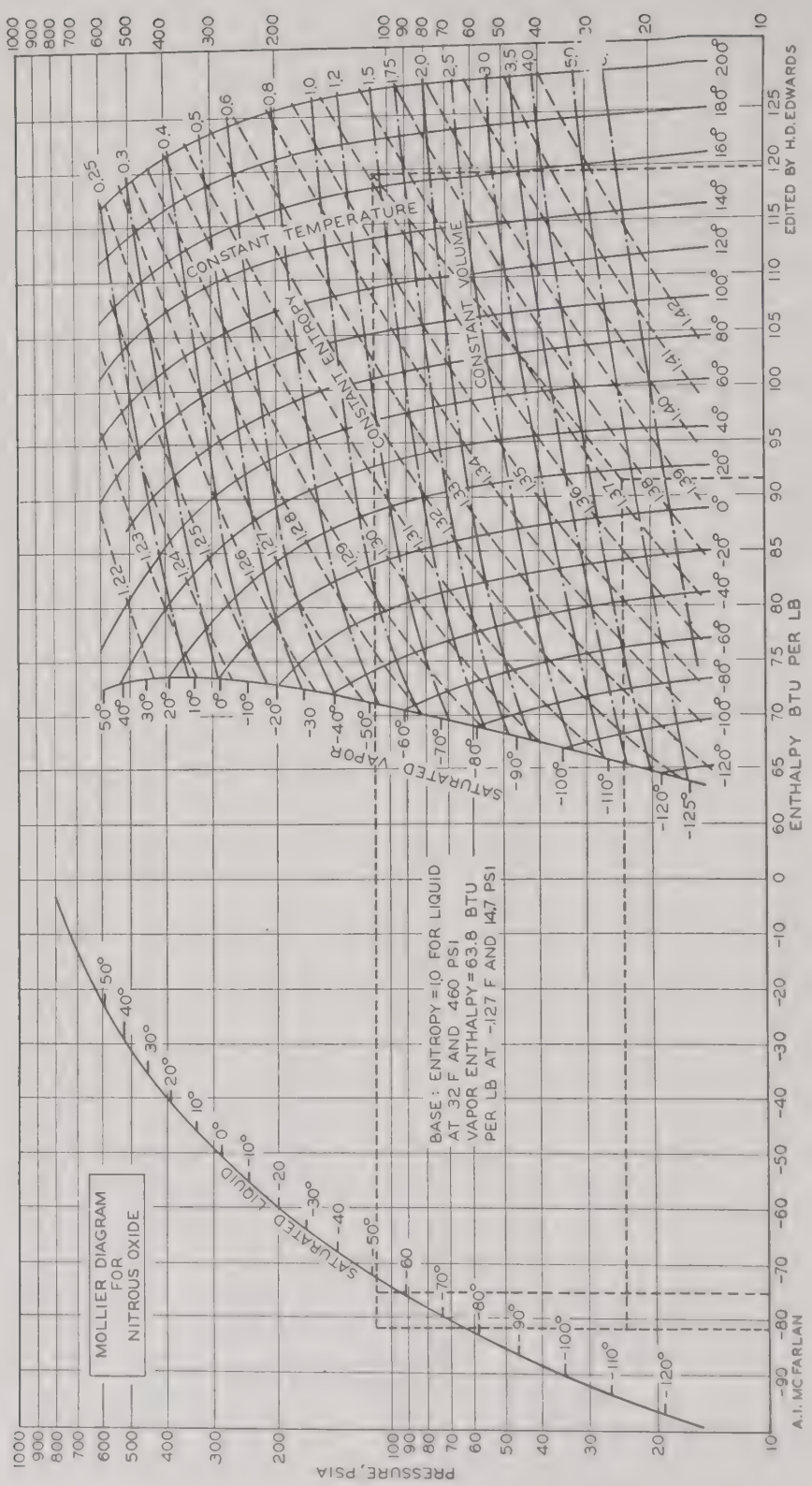

MOLLIER DIAGRAM FOR NITROUS OXIDE
BASE: ENTROPY = 1.0 FOR LIQUID AT 32 F AND 460 PSI
VAPOR ENTHALPY = 63.8 BTU PER LB AT -127 F AND 14.7 PSI
CONSTANT TEMPERATURE
CONSTANT ENTROPY
CONSTANT VOLUME
SATURATED LIQUID
SATURATED VAPOR
PRESSURE, PSIA
ENTHALPY BTU PER LB
A.I. MCFARLAN
EDITED BY H.D. EDWARDS

Chart 15

Table 27. Nitrous Oxide—Properties of Liquid and Saturated Vapor

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum 80 F Btu per lb		Entropy, datum 80 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−127	14.7	0.0	80.0	5.5	−98.5	63.8		1.33
−120	19.0	4.3	79.0	4.25	−96.6	64.5		1.312
−110	26.0	11.3	77.5	3.18	−93.8	65.7		1.3105
−100	35.0	20.3	76.0	2.38	−90.7	66.7		1.3029
− 90	46.0	31.3	75.0	1.82	−87.2	68.0		1.2966
− 80	58.0	42.3	74.0	1.43	−83.7	68.9		1.2887
− 70	74.0	59.3	72.5	1.15	−79.8	70.0		1.2835
− 60	91.0	76.3	70.5	0.935	−75.8	70.7		1.2775
− 50	112.0	97.3	69.3	0.752	−71.8	71.5		1.2722
− 40	137.0	122.3	68.0	0.606	−67.5	72.2		1.2657
− 30	167.0	152.3	67.0	0.513	−63.2	72.7		1.2595
− 20	203.0	188.3	65.5	0.425	−58.7	73.2		1.2530
− 10	240.0	225.3	64.7	0.364	−54.2	73.5		1.2455
0	283.0	268.3	63.1	0.303	−49.4	73.6		1.2365
10	335.0	320.3	61.2	0.262	−44.8	73.7		1.2300
20	387.0	372.3	59.2	0.217	−39.8	73.6		1.2243
32	460.0	447.3	57.0	0.1785	−33.3	73.5	1.00	1.2180
40	520.0	505.3	54.7	0.160	−28.2	73.3		
50	590.0	575.3	52.3	0.138	−21.1	72.6		
60	675.0	660.3	49.2	0.119	−14.2	72.0		
70	760.0	745.3	46.5	0.106	− 7.0	70.7		
80	865.0	850.3	40.0	0.08	0.0	69.0		
97	1069.0	1054.3	26.5	0.0377	66.5	66.5		

Table 28. Specific Volume of Aqua-Ammonia Solutions*

Cu ft per lb of subcooled or saturated liquid

	Weight Concentration Ammonia in Water											
Temp F	0.00	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.00	Temp F
20	.0160	.0166	.0172	.0177	.0182	.0188	.0196	.0206	.0218	.0232	.0247	20
40	.0160	.0167	.0172	.0178	.0183	.0190	.0199	.0209	.0222	.0236	.0253	40
60	.0160	.0167	.0173	.0179	.0185	.0192	.0201	.0213	.0226	.0242	.0260	60
80	.0161	.0168	.0174	.0180	.0186	.0195	.0204	.0216	.0231	.0248	.0267	80
100	.0161	.0169	.0175	.0182	.0189	.0197	.0208	.0221	.0236	.0254	.0275	100
120	.0162	.0170	.0176	.0184	.0191	.0200	.0212	.0226	.0242	.0262	.0284	120
140	.0163	.0171	.0178	.0186	.0194	.0204	.0216	.0230	.0249	.0269	.0294	140
160	.0164	.0172	.0180	.0188	.0197	.0207	.0223	.0237	.0256	.0279	.0306	160
180	.0165	.0174	.0182	.0191	.0200	.0212	.0226	.0244	.0265	.0289	.0320	180
200	.0166	.0175	.0184	.0193	.0204	.0216	.0232	.0252	.0275	.0304	.0338	200
220	.0168	.0177	.0186	.0197	.0210	.0225	.0242	.0264	.0288	.0320	.0361	220

* Prepared by Jennings and Shannon

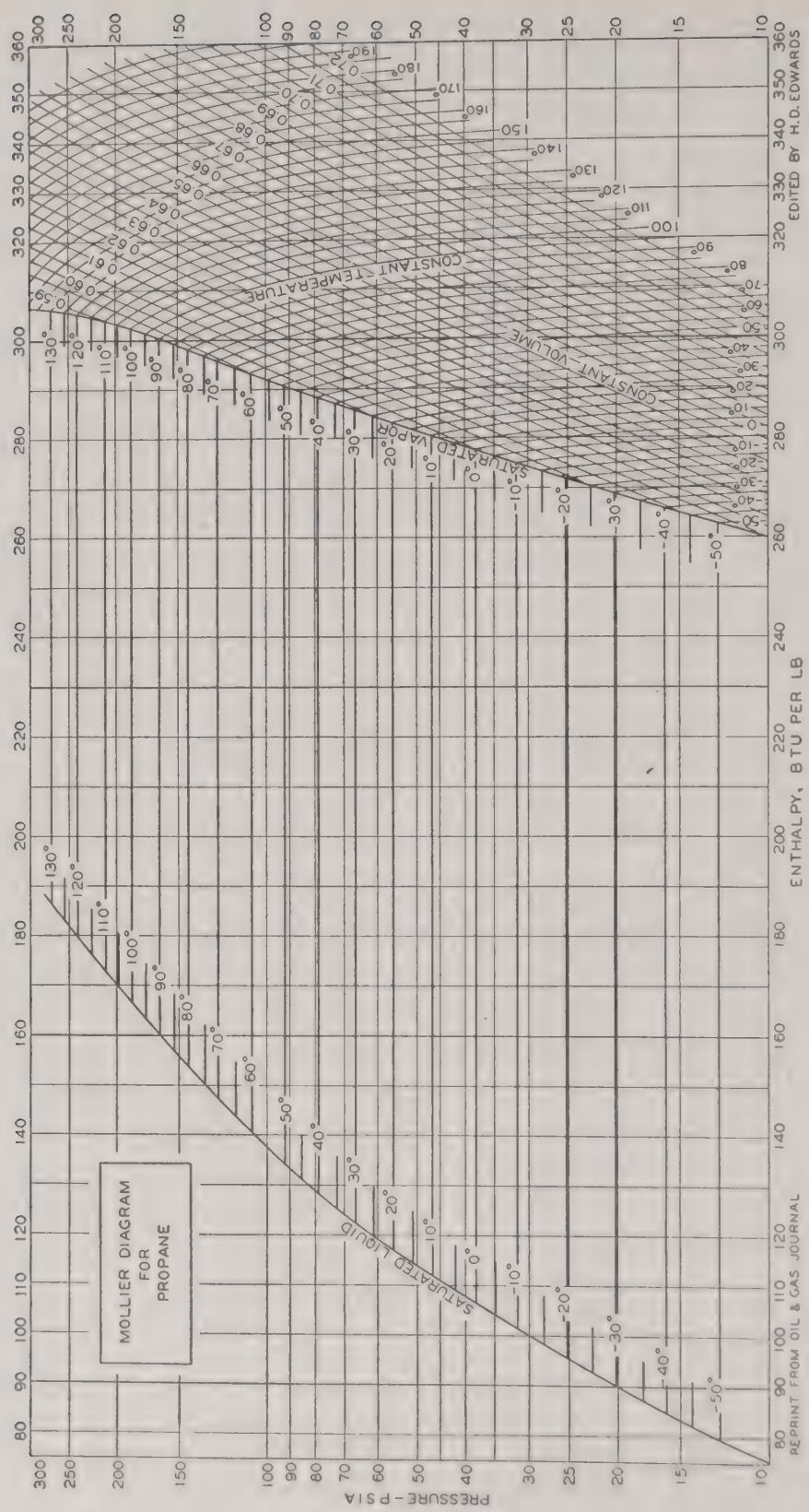

MOLLIER DIAGRAM FOR PROPANE
PRESSURE - PSIA
ENTHALPY, BTU PER LB
SATURATED LIQUID
SATURATED VAPOR
CONSTANT VOLUME
CONSTANT TEMPERATURE
REPRINT FROM OIL & GAS JOURNAL
EDITED BY H.D. EDWARDS

Chart 16

Table 29. Propane—Properties of Liquid and Saturated Vapor

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy,** datum −200 F Btu per lb		Entropy,** datum −200 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
−75	6.37	17.0*	37.59	14.5	65.5	255.5	0.142	0.624
−70	7.37	14.9*	37.40	12.9	68.0	257.0	.148	.623
−65	8.48	12.7*	37.20	11.3	71.5	258.0	.154	.621
−60	9.72	10.1*	37.00	9.93	74.0	259.5	.160	.620
−55	11.1	7.3*	36.80	8.70	77.0	261.0	.167	.618
−50	12.6	4.3*	36.60	7.74	79.5	262.7	0.173	0.617
−45	14.4	0.6*	36.39	6.89	82.3	264.2	.179	.615
−40	16.2	1.5	36.19	6.13	85.0	265.8	.185	.614
−35	18.1	3.4	35.99	5.51	87.5	267.2	.190	.613
−30	20.3	5.6	35.78	4.93	90.2	268.9	.196	.612
−25	22.7	8.0	35.58	4.46	92.8	270.3	0.202	0.610
−20	25.4	10.7	35.37	4.00	95.6	271.8	.208	.608
−15	28.3	13.6	35.16	3.60	98.3	273.2	.214	.607
−10	31.4	16.7	34.96	3.26	101.0	274.9	.220	.607
− 5	34.7	20.0	34.75	2.97	103.8	276.2	.226	.606
0	38.2	23.5	34.54	2.71	106.2	277.7	0.231	0.605
+ 5	41.9	27.2	34.33	2.48	108.8	279.0	0.236	0.604
10	46.0	31.3	34.12	2.27	113.3	280.5	.246	.603
15	50.6	35.9	33.90	2.07	114.0	281.8	.248	.602
20	55.5	40.8	33.67	1.90	116.8	283.1	.254	.601
25	60.9	46.2	33.43	1.74	119.7	284.4	.260	.600
30	66.3	51.6	33.20	1.60	122.3	285.7	0.266	0.599
35	72.0	57.3	32.97	1.48	125.0	287.0	.272	.598
40	78.0	63.3	32.73	1.37	128.0	288.3	.278	.597
45	84.6	69.9	32.49	1.27	131.1	289.5	.285	.596
50	91.8	77.1	32.24	1.18	134.2	290.7	.292	.596
55	99.3	84.6	32.00	1.10	137.2	292.0	0.298	0.596
60	107.1	92.4	31.75	1.01	140.6	293.2	.306	.595
65	115.4	100.7	31.50	0.945	143.8	294.5	.313	.594
70	124.0	109.3	31.24	.883	147.5	295.8	.321	.594
75	133.2	118.5	30.97	.825	150.3	296.9	.327	.594
80	142.8	128.1	30.70	0.770	154.0	298.1	0.335	0.593
85	153.1	138.4	30.42	.722	157.0	299.2	.342	.593
90	164.0	149.0	30.15	.673	160.3	300.3	.349	.593
95	175.0	160.0	29.87	.632	163.4	301.3	.356	.592
100	187.0	172.0	29.58	.591	166.8	302.4	.363	.592
105	200.0	185.0	29.27	0.553	169.8	303.2	0.370	0.592
110	212.0	197.0	28.96	.520	172.8	304.0	.376	.591
115	226.0	211.0	28.63	.488	176.2	304.7	.383	.590
120	240.0	225.0	28.30	.459	179.8	305.2	.391	.589
125	254.0	239.0	27.97	.432	183.5	305.8	.399	.588
130	272.0	257.3	27.64	0.404	186.8	306.1	0.406	0.587
135	288.0	273.3	27.32	.382	190.0	306.3	.413	.586
140	305.0	290.3	27.00	.360	194.0	306.5	.422	.585

Based on material from Dana, Jenkins, Burdick and Timm, published originally in *Refrigerating Engineering*, June 1926, vol. 12, no. 12, page 403.

* Inches of mercury below one standard atmosphere (29.92 in.).

** From Mollier Diagrams for Propane, W. C. Edminster, Standard Oil Co. (Indiana).

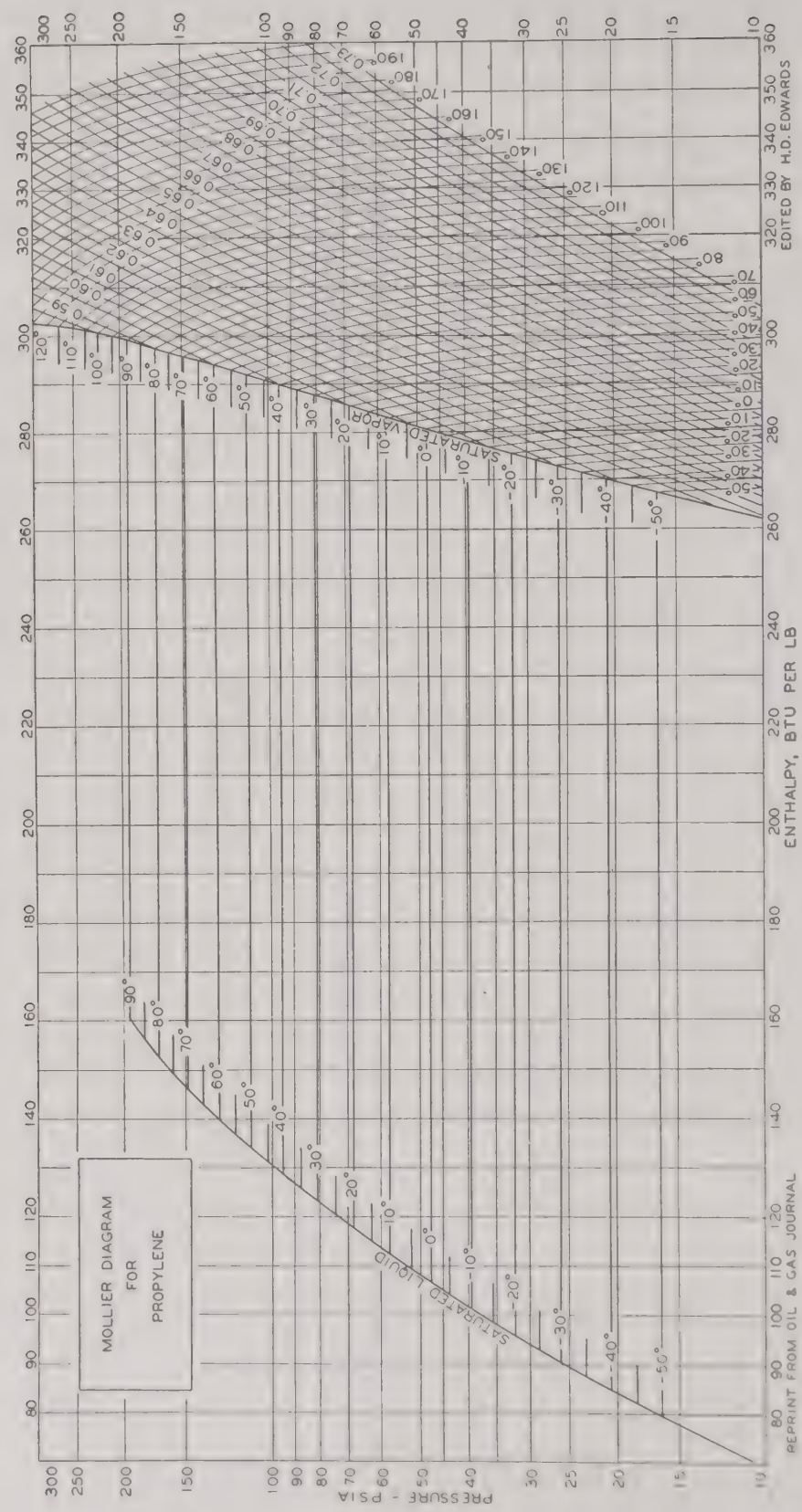
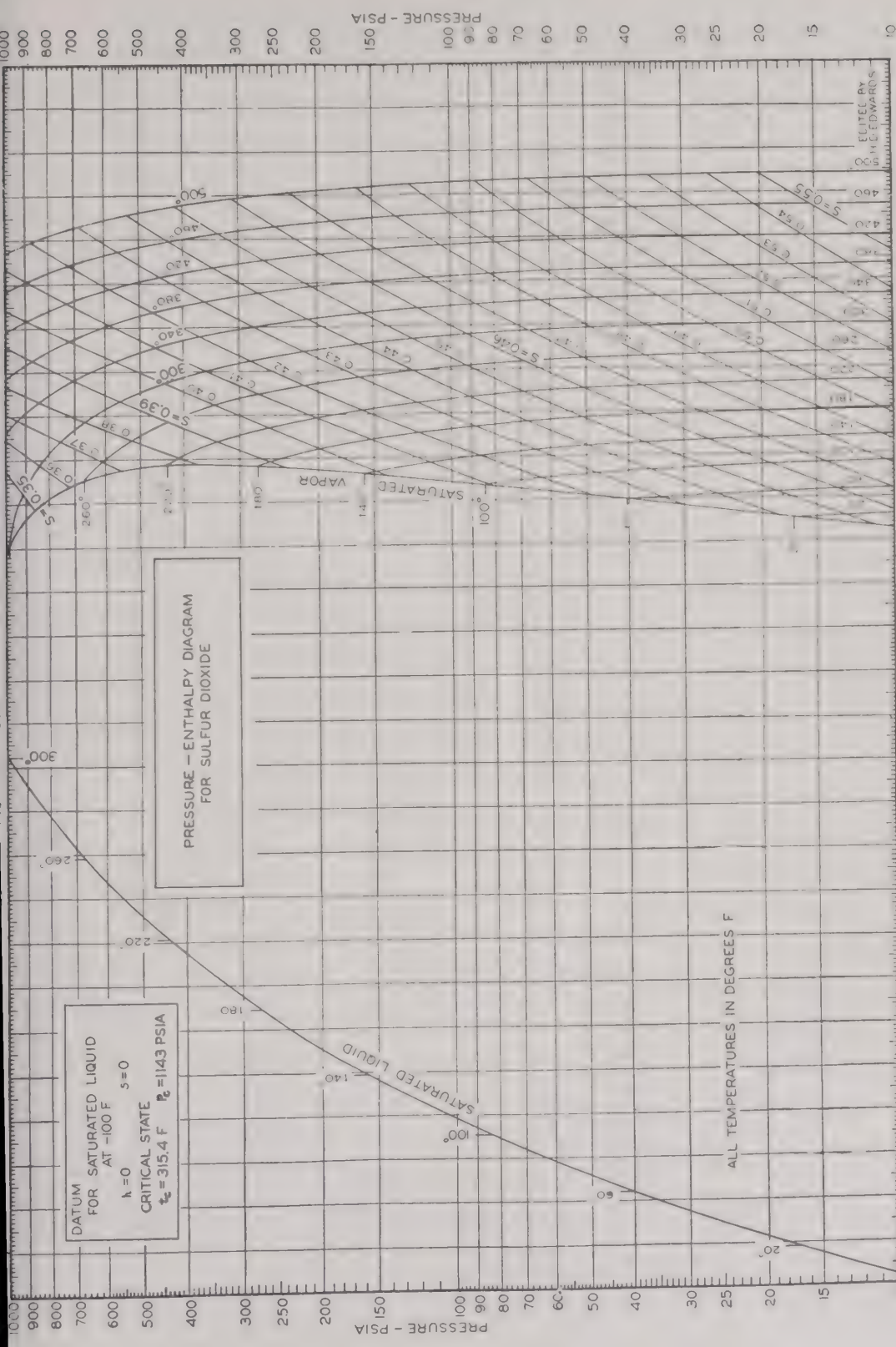

MOLLIER DIAGRAM
FOR
PROPYLENE
SATURATED LIQUID
SATURATED VAPOR
PRESSURE - PSIA
ENTHALPY, BTU PER LB
EDITED BY H.D. EDWARDS
REPRINT FROM OIL & GAS JOURNAL

Chart 17


PRESSURE – ENTHALPY DIAGRAM
FOR SULFUR DIOXIDE
DATUM
FOR SATURATED LIQUID
AT -100 F
h = 0 s = 0
CRITICAL STATE
t_c = 315.4 F P_c = 1143 PSIA
SATURATED LIQUID
SATURATED VAPOR
ALL TEMPERATURES IN DEGREES F
PRESSURE - PSIA
ENTHALPY - BTU PER LB
COPYRIGHT 1945 BY SHELL DEVELOPMENT CO
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REPRINT, TRANS. AM. INST. CHEM. ENGR. VOL. 41 NO. 3, 1945.

Chart 18

Table 30. Sulfur Dioxide—Properties of Liquid and Saturated Vapor

Temp F	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum −40 F Btu per lb		Entropy, datum −40 F Btu per lb F		
t	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Evap	Vapor s_g
−40	3.136	23.53*	0.01044	22.42	0.00	178.61	0.00000	0.42562	0.42562
−30	4.331	21.10*	.01053	16.56	2.93	179.90	.00674	.41190	.41864
−20	5.883	17.94*	.01063	12.42	5.98	181.07	.01366	.39826	.41192
−10	7.863	13.91*	.01072	9.44	9.16	182.13	.02075	.38469	.40544
0	10.35	8.85*	0.01082	7.280	12.44	183.07	0.02795	0.37122	0.39917
2	10.91	7.71*	.01084	6.923	13.12	183.25	.02941	.36853	.39794
4	11.50	6.51*	.01086	6.584	13.78	183.41	.03084	.36586	.39670
5†	11.81	5.88*	0.01087	6.421	14.11	183.49	0.03155	0.36454	0.39609
6	12.12	5.25*	.01088	6.266	14.45	183.57	.03228	.36319	.39547
8	12.75	3.97*	.01090	5.967	15.13	183.73	.03373	.36053	.39426
10	13.42	2.61*	0.01092	5.682	15.80	183.87	0.03519	0.35787	0.39306
11	13.77	1.89*	.01093	5.548	16.14	183.94	.03592	.35654	.39246
12	14.12	1.18*	.01094	5.417	16.48	184.01	.03664	.35521	.39185
13	14.48	0.45*	.01095	5.289	16.81	184.07	.03737	.35388	.39125
14	14.84	0.14	.01096	5.164	17.15	184.14	.03808	.35257	.39065
15	15.21	0.51	0.01097	5.042	17.49	184.21	0.03880	0.35125	0.39005
16	15.59	0.89	.01098	4.926	17.84	184.28	.03953	.34993	.38946
17	15.98	1.28	.01099	4.812	18.18	184.34	.04026	.34861	.38887
18	16.37	1.67	.01100	4.701	18.52	184.40	.04098	.34729	.38827
19	16.77	2.07	.01101	4.593	18.86	184.46	.04169	.34598	.38767
20	17.18	2.48	0.01102	4.487	19.20	184.52	0.04241	0.34466	0.38707
21	17.60	2.90	.01104	4.386	19.55	184.58	.04313	.34335	.38648
22	18.03	3.33	.01105	4.287	19.90	184.64	.04385	.34204	.38589
23	18.46	3.76	.01106	4.190	20.24	184.69	.04457	.34073	.38530
24	18.89	4.19	.01107	4.096	20.58	184.74	.04528	.33943	.38471
25	19.34	4.64	0.01108	3.994	20.92	184.79	0.04600	0.33812	0.38412
26	19.80	5.10	.01109	3.915	21.26	184.84	.04671	.33683	.38354
27	20.26	5.56	.01110	3.829	21.61	184.89	.04743	.33553	.38296
28	20.73	6.03	.01112	3.744	21.96	184.94	.04814	.33422	.38236
29	21.21	6.51	.01113	3.662	22.30	184.98	.04886	.33292	.38178
30	21.70	7.00	0.01114	3.581	22.64	185.02	0.04956	0.33163	0.38119
31	22.20	7.50	.01115	3.503	22.98	185.06	.05027	.33034	.38061
32	22.71	8.01	.01116	3.437	23.33	185.10	.05099	.32904	.38003
33	23.23	8.53	.01118	3.355	23.68	185.14	.05171	.32774	.37945
34	23.75	9.05	.01119	3.283	24.03	185.18	.05242	.32645	.37887
35	24.28	9.58	0.01120	3.212	24.38	185.22	0.05312	0.32517	0.37829
40	27.10	12.40	.01126	2.887	26.12	185.37	.05668	.31873	.37541
45	30.15	15.45	.01132	2.601	27.86	185.48	.06020	.31234	.37254
50	33.45	18.75	.01138	2.348	29.61	185.56	.06370	.30599	.36969
55	37.05	22.35	0.01144	2.124	31.36	185.60	0.06715	0.29971	0.36686
60	40.93	26.23	.01150	1.926	33.10	185.59	.07060	.29345	.36405
65	45.13	30.43	.01156	1.749	34.84	185.54	.07401	.28724	.36125
70	49.62	34.92	.01163	1.590	36.58	185.46	.07736	.28110	.35846
75	54.47	39.77	.01169	1.448	38.32	185.34	.08070	.27498	.35568
80	59.68	44.98	0.01176	1.321	40.05	185.17	0.08399	0.26897	0.35296
81	60.77	46.07	.01177	1.297	40.39	185.13	.08462	.26772	.35234
82	61.88	47.18	.01179	1.274	40.73	185.09	.08525	.26652	.35177
83	63.01	48.31	.01180	1.253	41.08	185.05	.08589	.26532	.35121
84	64.14	49.44	.01181	1.229	41.43	185.01	.08653	.26412	.35065
85	65.28	50.58	0.01183	1.207	41.78	184.97	0.08718	0.26291	0.35009
86†	66.45	51.75	.01184	1.185	42.12	184.92	.08783	.26171	.34954
87	67.64	52.94	.01185	1.164	42.46	184.87	.08847	.26052	.34899
88	68.84	54.14	.01187	1.144	42.80	184.82	.08910	.25993	.34843
89	70.04	55.34	.01188	1.124	43.15	184.77	.08974	.25813	.34787
90	71.25	56.55	0.01190	1.104	43.50	184.72	0.09038	0.25693	0.34731
95	77.60	62.90	.01196	1.011	45.20	184.43	.09349	.25103	.34452
100	84.52	69.82	.01204	0.9262	46.90	184.10	.09657	.24516	.34173
110	99.76	85.06	.01219	.7804	50.26	183.31	.10254	.23357	.33611
120	120.93	106.23	.01236	.6598	53.58	182.36	.10829	.22217	.33046
140	158.61	143.91	.01272	.4758	60.04	179.94	.11893	.19990	.31883

* Inches of mercury below one standard atmosphere.
† Standard cycle temperatures.
Note—For values used in making chart No. 18 see Transactions of American Institute of Chemical Engineers, vol. 41, no. 3, June 25, 1945.

Table 31. Sulfur Dioxide—Properties of Superheated Vapor

Pressure 10 psia Gage pressure 9.6 in. vac (Sat temp −1.34 F)			Pressure 15 psia Gage pressure 0.3 psig (Sat temp 14.43 F)			Pressure 20 psia Gage pressure 5.3 psig (Sat temp 26.44 F)			Pressure 25 psia Gage pressure 10.3 psig (Sat temp 36.33 F)		
v	h	s	v	h	s	v	h	s	v	h	s
(7.52)	*(182.95)*	*(0.4000)*	*(5.110)*	*(185.17)*	*(0.39091)*	*(3.878)*	*(184.86)*	*(0.38329)*	*(3.123)*	*(185.26)*	*(0.37764)*
7.939	186.7	0.40802	5.192	185.4	0.39270						
8.030	188.4	.41159	5.333	187.3	.39672						
8.316	190.1	.41505	5.470	189.2	.40054	4.035	187.8	0.38959	3.181	186.1	0.37927
8.500	191.8	.41837	5.604	191.0	.40424	4.145	189.8	.39346	3.273	188.4	.38372
8.681	193.5	.42161	5.734	192.8	.40777	4.251	191.8	.39719	3.363	190.6	.38795
8.860	195.2	0.42480	5.862	195.6	0.41116	4.354	193.7	0.40080	3.451	192.7	0.39198
9.038	196.9	.42795	5.988	196.4	.41443	4.454	195.6	.40429	3.536	194.7	.39582
9.214	198.6	.43104	6.112	198.2	.41765	4.552	197.5	.40758	3.618	196.7	.39945
9.389	200.3	.43407	6.233	199.9	.42076	4.648	199.3	.41093	3.696	198.6	.40291
9.563	202.0	.43705	6.353	210.6	.42383	4.742	201.1	.41415	3.772	200.5	.40625
9.736	203.7	0.43997	6.471	203.3	0.42682	4.834	202.9	0.41726	3.848	202.4	0.40949
9.908	205.4	.44283	6.588	205.6	.42976	4.925	204.7	.42027	3.923	204.2	.41261
10.08	207.1	.44565	6.705	206.7	.43264	5.015	206.5	.42322	3.998	206.0	.41568
10.25	208.8	.44842	6.821	208.4	.43548	5.104	208.2	.42613	4.073	207.8	.41866
10.42	210.5	.45116	6.937	210.1	.43825	5.193	209.9	.42898	4.145	209.6	.42158
10.59	212.2	0.45296	7.052	211.8	0.44097	5.281	211.6	0.43176	4.216	211.4	0.42439
10.76	213.8	.45651	7.167	213.5	.44366	5.369	213.3	.43449	4.287	213.2	.42717
10.93	215.4	.45913	7.282	215.2	.44630	5.456	215.0	.43716	4.358	215.0	.42988
11.10	217.0	.46171	7.396	216.9	.44889	5.542	216.7	.43977	4.428	216.7	.43253
						5.629	218.4	.44234	4.498	218.4	.43413

Pressure 40 psia Gage pressure 25.3 psig (Sat temp 58.83 F)			Pressure 50 psia Gage pressure 35.3 psig (Sat temp 70.40 F)			Pressure 60 psia Gage pressure 45.3 psig (Sat temp 80.29 F)			Pressure 70 psia Gage pressure 55.3 psig (Sat temp 88.97 F)		
v	h	s	v	h	s	v	h	s	v	h	s
1.970	*(185.60)*	*(0.36470)*	*(1.577)*	*(185.45)*	*(0.35826)*	*(1.3144)*	*(185.16)*	*(0.35272)*	*(1.125)*	*(184.77)*	*(0.34789)*
2.246	196.1	0.38415	1.650	193.9	0.37369	1.448	191.4	0.36403	1.181	187.6	0.35443
2.304	198.3	0.38810	1.825	196.4	0.37815	1.500	194.3	0.36906	1.228	191.6	0.36020
2.360	200.4	.39183	1.872	198.8	.38234	1.548	197.0	.37375	1.272	194.8	.36545
2.413	202.5	.39541	1.917	201.1	.38627	1.590	199.5	.37810	1.313	197.6	.37028
2.465	204.6	.39881	1.961	203.3	.38998	1.629	201.9	.38217	1.352	200.3	.37478
2.515	206.5	.40209	2.003	205.4	.39353	1.669	204.2	.38603	1.389	202.9	.37897
2.565	208.5	0.40525	2.044	207.5	0.39691	1.708	206.5	0.38963	1.424	205.3	0.38291
2.614	210.4	.40831	2.084	209.6	.40015	1.744	208.6	.39310	1.457	207.6	.38662
2.662	212.3	.41127	2.123	211.6	.40327	1.778	210.7	.39639	1.489	209.9	.39014
2.709	214.2	.41416	2.161	213.4	.40628	1.808	212.8	.39956	1.521	212.0	.39348
2.755	216.0	.41694	2.199	215.4	.40919	1.842	214.8	.40260	1.551	214.1	.39670
2.800	217.9	0.41966	2.237	217.3	0.41200	1.874	216.8	0.40554	1.580	216.1	0.39978
2.845	219.7	.42233	2.274	219.2	.41477	1.907	218.7	.40839	1.608	218.1	.40275
2.889	221.5	.42494	2.311	221.1	.41748	1.939	220.7	.41118	1.636	220.1	.40564
2.933	223.3	.42751	2.347	223.0	.42015	1.970	222.6	.41391	1.664	222.1	.40845
2.977	225.1	.43007	2.383	224.9	.42275	2.000	224.5	.41657	1.691	224.1	.41120
3.021	227.0	0.43262	2.418	226.7	0.42535	2.032	226.4	0.41917	1.718	226.0	0.41389

Pressure 80 psia Gage pressure 65.3 psig (Sat temp 96.88 F)			Pressure 100 psia Gage pressure 85.3 psig (Sat temp 110.15 F)			Pressure 120 psia Gage pressure 105.3 psig (Sat temp 121.52 F)			Pressure 140 psia Gage pressure 125.3 psig (Sat temp 131.64 F)		
v	h	s	v	h	s	v	h	s	v	h	s
(0.9809)	*(184.33)*	*(0.34567)*	*(0.7786)*	*(183.30)*	*(0.33603)*	*(0.6430)*	*(182.19)*	*(0.33954)*	*(0.5451)*	*(181.04)*	*(0.32388)*
1.163	198.6	0.36819	0.8928	194.6	0.35528	0.7085	190.1	0.34264	0.5734	185.1	0.33089
1.199	201.3	0.37270	0.9255	197.9	0.36061	0.7403	193.9	0.34904	0.6055	189.7	0.33777
1.232	203.9	.37692	0.9561	200.9	.36558	0.7700	197.4	.35484	0.6345	193.6	.34442
1.263	206.4	.38093	0.9848	203.7	.37009	0.7972	200.6	.36012	0.6613	196.3	.35041
1.292	208.7	.38461	1.012	206.4	.37431	0.8288	203.7	.36494	0.6861	200.8	.35588
1.320	211.0	.38813	1.038	209.0	.37829	0.8470	206.7	.36936	0.7092	204.0	.36088
1.347	213.3	0.39150	1.062	211.5	0.38203	0.8699	209.4	0.37348	0.7309	207.1	0.36548
1.374	215.5	.39471	1.086	213.8	.38556	0.8916	212.0	.37737	0.7513	210.0	.36976
1.400	217.5	.39780	1.109	216.1	.38892	0.9124	214.5	.38104	0.7707	212.7	.37379
1.426	219.6	.40079	1.131	218.4	.39214	0.9342	217.0	.38451	0.7892	215.4	.37758
1.451	221.6	.40369	1.152	220.5	.39524	0.9515	219.3	.38785	0.8070	217.9	.38118

Table 32. Properties of Liquid and Saturated Vapor of Ethylamine, Methylamine, Methylene Chloride, Methyl Formate

Temp F t	Pressure		Liquid, density	Vapor, sp vol	Enthalpy, datum 0 F Btu per lb		Entropy, datum 0 F Btu per lb F	
	psia	psig	lb/cu ft $1/v_f$	cu ft/lb v_g	Liquid h_f	Vapor h_g	Liquid s_f	Vapor s_g
Ethylamine ($C_2H_5NH_2$)								
−58	.355	29.20*		270.8	−7.82	284.78	−0.0263	0.7094
−40	0.740	28.42*		134.5	0.00	290.93	.0000	.6931
−22	1.408	27.06*		72.87	10.93	296.84	.0253	.6788
− 4	2.546	24.75*		41.71	23.01	303.05	.0498	.6664
5	3.342	23.12*	0.02212	32.32	27.61	306.04	.0619	.6609
23	5.590	18.76*		20.00	38.91	311.92	0.0857	0.6512
41	8.960	11.72*		12.88	50.36	317.70	.1091	.6430
68	16.896	2.20		7.156	68.18	326.50	.1437	.6332
86	24.72	10.02		5.039	80.50	332.46	.1664	.6281
113	41.49	26.79		3.136	99.52	341.44	.1999	.6223
Methylamine (CH_3NH_2)								
−58	1.322	27.24*		98.93	−12.4	374.8	−0.0503	0.9135
−40	2.532	25.14*		56.72	0.0	381.7	.0000	.9092
−22	4.551	20.67*		32.56	12.7	388.4	.0294	.8877
− 4	7.78	14.12*		19.84	25.7	395.0	.0586	.8689
5	10.03	9.89*	0.02273	15.54	32.6	398.5	.0731	.8603
23	15.99	1.29		10.01	45.6	404.3	0.1015	0.8443
41	24.49	9.79		6.796	59.2	410.1	.1292	.8300
68	43.52	28.82		3.985	80.2	418.8	.1698	.8115
86	61.53	46.83		2.962	94.5	424.4	.1963	.8008
113	98.76	84.06		1.867	116.9	432.6	.2360	.7873
Methyl Formate ($C_2H_4O_2$)								
0	1.50	26.87*		54.0	0	232.5	0.0000	0.5075
20	2.70	24.43*		31.0	10.3	236.6	.0219	.4934
40	4.66	20.45*		18.9	20.6	240.7	.0432	.4837
60	7.61	14.44*		12.0	30.9	244.8	.0633	.4748
80	12.07	24.55*		7.98	41.2	248.8	.0825	.4670
100	18.26	3.56		5.38	51.5	252.9	0.1015	0.4615
120	27.24	12.54		3.74	61.8	257.0	.1192	.4559
140	38.41	23.71		2.65	72.1	261.0	.1375	.4525
Methylene Chloride (Carrene) (CH_2Cl_2)								
10	1.38	27.12*		42.55	3.4	164.4	0.0072	0.3502
20	1.92	26.01*		31.40	6.8	165.6	.0151	.3461
30	2.56	24.71*		23.90	10.2	166.9	.0222	.3425
40	3.38	23.04*		18.60	13.6	168.0	.0285	.3377
60	5.52	18.72*		11.68	20.4	170.1	.0410	.3292
80	8.81	12.02*		7.50	27.2	172.0	0.0520	0.3202
100	13.25	2.97*		5.14	34.0	173.7	.0620	.3113
120	19.20	4.50		3.65	40.8	175.0	.0714	.3031
140	26.79	12.09		2.69	47.6	176.0	.0795	.2935

* Inches of mercury below one standard atmosphere.
See Table 6.

8. REFRIGERANT CHARACTERISTICS

IN THE early days of mechanical refrigeration, few chemical compounds were available that were suitable as refrigerants and the types of refrigerating equipment were necessarily limited to those that could be used with those few refrigerants. As the industry grew, as new uses for refrigeration appeared, and as different types of equipment were designed, the desirability of new refrigerants more suitable to those uses and equipment designs became more evident.

The use of mechanical refrigeration in the home and in retail businesses called for a low-pressure refrigerant to permit lighter, smaller, and lower priced equipment; sulfur dioxide and later ethyl chloride and methyl chloride made their appearance. The need for a refrigerant having high vapor density, low ratio of compression, and other characteristics suitable for a centrifugal compressor led to the introduction of methylene chloride.

Finally, physical chemists exploring the field of the hydrocarbons of the methane and ethane series, found that by juggling the atoms of these hydrocarbons, substituting chlorine and/or fluorine atoms for hydrogen atoms, they could synthesize almost any type of refrigerant needed for a specific use or for a specific type of equipment. These halogenated hydrocarbons are supplied to the industry under the trade name Freon.

And yet the search goes on, with the trend toward non-flammable, non-toxic, odorless refrigerants of greater stability, even though these characteristics may be obtained at the expense of efficiency, horsepower-per-ton, and coefficient of performance, for these differ so little among the commonly used refrigerants that considerations of safety, stability, boiling point, condensing pressure and other properties suitable to a specific application are apt to be more compelling, especially for the smaller, automatic equipment. In the larger installations attended by operating personnel, the factors affecting operating costs and the size and original cost of the equipment, including piping, may outweigh toxicity, flammability, odor, and even corrosion factors. No one refrigerant may be considered as a universal ideal, and many angles must be considered in determining the suitability of a refrigerant for a specific application or for a specific type of equipment.

Refrigerant **suitability** for a specific application or use should be determined by consideration of (1) thermodynamic, physical and chemical properties; (2) reaction with moisture; (3) leak detection; (4) action on materials; (5) safety considerations and codes; (6) cost; (7) type of compressor; and (8) methods of handling.

I. Thermodynamic, Physical and Chemical Properties (Tables 1, 2 and 3)

1. Boiling temperature and pressure. Inasmuch as the evaporator temperature determines the evaporator pressure, it is desirable to select a refrigerant whose saturation pressure at minimum evaporator operating temperature is slightly above zero gage, in order to maintain a small positive differential between suction (and crankcase) pressure, and atmospheric pressure. Probability of leakage of refrigerant from the system or of air and/or moisture into the low-pressure side of the system, especially at the compressor shaft seal, is less with the smaller pressure differentials. This factor will require less consideration if the motor-compressor unit is of the hermetically-sealed type, but cannot be ignored even in the sealed construction.

2. Freezing temperature. The refrigerant used must have a freezing temperature well below the minimum temperature at

Table 1. Comparative Refrigerant Characteristics

Performance based on 5 F evaporation and 86 F condensation

Refrigerant	Chemical symbol	Molecular weight	Boiling temperature at 0 psig, F	Freezing temperature at 0 psig, F	Critical temperature, F	Critical pressure, psia	Evaporator pressure at 5 F, psig	Condensing pressure at 86 F, psig	Ratio of compression 86 F, 5 F, psia	Net refrigerating effect of liquid 86 F, 5 F, Btu/lb	Refrigerant circulated per ton 86 F, 5 F, Lb/min	Liquid circulated per ton 86 F, 5 F, Cu in./min	Specific volume of vapor, 5 F, cu ft/lb	Compressor displacement per ton 86 F, 5 F, cfm	Horsepower per ton 86 F, 5 F	Coefficient of performance 86 F, 5 F	Temperature of compressor discharge F
Air	—	—	−318.0	—	−221.0	547.0	—	—	5.00	28.5	7.02	—	11.7	82.3	2.82	1.67	—
Methane	CH_4	16.03	−258.9	−297.0	−115.8	673.0	—	—	5.00	53.8	3.71	—	21.2	78.7	1.203	3.91	—
Ethylene	C_2H_4	28.03	−155.0	−272.0	48.85	731.8	404.3	—	5.00	45.0	4.45	—	.275	1.4	1.532	2.57	—
Nitrous oxide	N_2O	44.02	−127.0	−152.0	96.5	1050.0	318.3	915.3	2.80	85.2	2.35	71.2	.308	.73	1.31	3.59	—
Ethane	C_2H_6	30.04	−127.5	−278.0	90.1	708.3	221.3	661.1	2.87	58.6	3.41	409.0	.533	1.82	2.18	2.16	—
Carbon dioxide	CO_2	44.00	−109.3	− 69.9	87.8	1071.0	316.8	1028.3	3.15	55.5	3.61	167.0	.266	.96	1.827	2.59	160
Propylene	C_3H_6	42.05	− 53.7	−301.0	196.5	667.2	37.0	167.0	4.95	189.2	1.06	—	—	—	1.060	4.46	107
Propane	C_3H_8	44.06	− 44.2	−309.8	202.0	661.5	27.2	140.5	3.70	121.0	1.65	94.0	2.48	4.09	1.020	4.62	97
Freon-22	$CHClF_2$	86.48	− 41.4	−256.0	204.8	716.0	28.3	159.8	4.06	69.3	2.89	68.1	1.25	3.60	1.016	4.65	131
Ammonia	NH_3	17.03	− 28.0	−107.9	271.2	1651.0	19.6	154.5	4.94	474.6	.421	19.6	8.15	3.46	.981	4.75	210
Freon-12	CCl_2F_2	120.90	− 21.6	−247.0	233.0	580.0	11.8	93.2	4.08	51.1	3.91	83.8	1.48	5.81	1.005	4.69	102
Methyl chloride	CH_3Cl	50.48	− 10.8	−144.0	289.4	969.0	6.5	80.0	4.48	150.2	1.33	40.9	4.47	5.95	1.020	4.62	178
Isobutane	C_4H_{10}	58.12	10.3	−229.0	272.7	537.0	3.3*	44.8	4.54	111.5	1.79	91.1	6.41	11.50	1.083	4.36	80
Sulfur dioxide	SO_2	64.06	14.0	−103.9	314.8	1141.5	5.9*	51.8	5.63	142.8	1.40	28.6	6.42	9.10	.969	4.86	195
Methylamine	CH_5N	31.05	20.3	−134.5	314.0	1082.0	9.0*	63.2	6.12	304.5	.66	28.2	15.54	1.03	.978	4.81	—
Butane	C_4H_{10}	58.12	31.3	−211.0	306.0	550.1	13.2*	26.9	5.01	123.5	1.62	79.1	9.98	16.16	1.011	4.67	86
Freon-114	$C_2Cl_2F_4$	170.93	38.4	−137.0	294.3	474.0	16.1*	22.0	5.42	43.1	4.64	89.3	4.22	19.59	.997	4.74	71
Freon-21	$CHCl_2F$	102.93	48.0	−211.0	353.3	750.0	19.3*	16.5	5.97	89.4	2.24	45.8	9.12	20.46	.932	5.11	140
Ethyl chloride	C_2H_5Cl	64.52	54.5	−217.7	369.0	764.0	20.5*	12.4	5.83	142.3	1.41	44.4	16.10	22.62	.906	5.19	106
Ethylamine	C_2H_7N	45.06	61.8	−115.0	362.0	815.0	23.1*	10.3	7.25	243.0	.823	323.0	32.32	26.4	.855	5.55	—
Freon-11	CCl_3F	137.38	74.7	−168.0	388.4	635.0	24.0*	3.6	6.24	67.5	2.96	56.0	12.27	36.3	.943	5.00	113
Methyl formate	$C_2H_4O_2$	60.04	89.2	−147.5	418.0	607.0	26.3*	1.8*	7.75	189.2	1.06	29.9	47.25	49.9	—	—	149
Ethyl ether	$C_4H_{10}O$	74.08	94.3	−177.3	522.1	380.8	26.9*	4.9*	8.20	126.3	1.58	62.9	35.00	55.4	.822	5.74	—
Methylene chloride	CH_2Cl_2	84.93	103.6	−142.0	421.0	670.0	27.4*	8.4*	9.07	134.6	1.49	30.9	49.90	74.3	.968	4.87	205
Freon-113	$C_2Cl_3F_3$	187.39	117.6	− 31.0	417.4	495.0	27.9*	13.9*	8.02	53.7	3.73	66.4	27.04	100.9	.981	4.84	81
Dichlorethylene	$C_2H_2Cl_2$	96.93	118.0	− 70.0	470.0	795.0	28.3*	15.8*	8.42	114.3	1.75	38.3	63.6	111.2	.973	4.83	—
Trichlorethylene	C_2HCl_3	131.37	187.0	−124.3	520.0	728.0	29.6*	26.2*	11.65	91.7	2.18	41.6	229.4	502.0	.980	4.82	—
Water (40 F and 86 F)	H_2O	18.02	212.0	32.0	706.1	3226.0	29.7*	28.7*	5.06	1025.3	.195	5.4	2444.0	476.6	1.125	4.20	282

* Inches of mercury vacuum.

which the evaporator may possibly operate.

3. Condensing temperature and pressure. In installations designed for a specific location with known temperature limits of the condensing medium, the refrigerant to be used may be more accurately chosen, than if, as is generally true, the equipment is made for nation-wide use, and subject to wide temperature limits of the condensing medium. For this latter type of equipment the maximum condensing temperature likely to be encountered must be a governing factor in the selection of the refrigerant.

In any case, it is desirable to use a refrigerant whose condensing pressure at extremes of design and safety conditions is low enough to permit the use of lightweight material in the high-pressure portion of the system, including non-ferrous sheet metals or pipe of light gages. This applies to the condenser, receiver, liquid line, expansion valve, liquid line solenoid, or other accessories subject to condensing pressure. Such construction is desirable from considerations of safety, original and operating costs, and conservation of space.

The condensing pressure will be a determining factor in the choice of type of condenser, air-cooled or water-cooled. Thus it may automatically eliminate a refrigerant from applications in which the air-cooled condenser is a practical and economic necessity.

There is also increased likelihood of leakage of refrigerant from the system and greater danger in handling or working around a refrigerant having a relatively high condensing pressure.

4. Discharge temperature. Refrigerants that have relatively high compressor discharge temperatures are apt to cause oil break-down and formation of sludges unless highly refined, expensive oils are used.

5. Critical temperature. The refrigerant used must have a critical temperature well above the maximum condensing temperature that may be experienced, even under extreme conditions. The critical temperatures of most of the commonly used refrigerants (except carbon dioxide) are high enough that this factor may be disregarded for most types and locations of equipment.

6. Critical pressure. The refrigerant must have a critical pressure well above the likely maximum condensing pressure. High power consumption for compression results if the critical pressure is but little higher than the condensing pressure corresponding to normal water or air temperature. As is true of the critical temperature, this factor is satisfactory for all of the commonly used refrigerants other than carbon dioxide.

7. Compression ratio. A refrigerant having a low ratio of compression is desirable as to first cost of the equipment and cost of operation. The original cost of the compressor will be increased by a high ratio of compression because of the higher precision in manufacture necessary to minimize losses from clearance volume and from leakage between the cylinder walls and pistons. As these factors affect volumetric efficiency, the overall cost of operation is correspondingly affected.

High compression ratios affect the design, construction and operating characteristics of expansion valves, liquid line solenoid valves and the like, enabling them to operate at the high difference in pressure on the two sides of the valve.

Rotary and centrifugal type compressors should use only refrigerants having comparatively low ratios of compression to minimize vapor slippage past the rotor.

8. Latent heat of vaporization. Other factors being equal, a high latent heat of vaporization is desirable, since it affects the net refrigerating effect, the amount of refrigerant circulated, and the size and cost of piping and equipment. The value of a high latent heat of vaporization is sometimes over-rated, however, for its bearing on the selection of a refrigerant is affected by other important factors, such as specific heat of the liquid, and by the density of the liquid and of the vapor. In fact, the small machine (household and small commercial sizes) favors a refrigerant with a moderate to low latent heat of vaporization, to prevent the expansion valve or refrigerant float valves from having such small orifices as to make very difficult the accurate metering of liquid into the evaporator.

9. Specific heat of the liquid. The latent heat of vaporization is the "refrigerating ability" of the refrigerant. It is reduced because some of it is required to cool the

Table 2. Comparative Refrigerant Characteristics at Various Evaporating and Condensing Temperatures

	Evaporating pressure, psig	Condensing pressure, psig	Ratio of compression	Net refrigerating effect, Btu/lb	Refrigerant circulated per ton, lb/min	Specific volume of vapor, cu ft/lb	Compressor displacement per ton, cu ft/min
—130 F Evaporating, —40 F Condensing							
Ethylene	16.16	196.7	6.8	141.4	1.42	3.72	5.27
Nitrous oxide	1.02″	145.3	11.3			5.94	
Ethane	2.24″	98.2	8.3	155.1	1.29	8.16	10.52
Carbon dioxide	18.84″	131.0	27.1	130.6	1.53	14.74	22.60
Freon-22	28.51″	.61	22.2	89.7	2.25	58.21	130.0
—76 F Evaporating, 5 F Condensing							
Ethylene	95.0	399.8	3.8	117.0	1.71	1.14	1.95
Nitrous oxide	60.3	318.3	4.4			1.22	
Ethane	40.4	221.3	4.2	131.1	1.53	2.30	3.51
Propane	17.4″	35.9	8.2	147.7	1.35	14.80	20.02
Freon-22	18.87″	28.3	8.1	84.2	2.38	8.56	20.37
Ammonia	23.47″	19.6	10.9	534.7	.37	75.00	28.05
Freon-12	23.54″	11.8	8.5	59.9	3.34	10.02	33.47
Methyl chloride	25.34″	6.5	9.4	169.2	1.18	36.00	42.55
—40 F Evaporating, 68 F Condensing							
Nitrous oxide	145.3	730.3	4.7			.600	
Ethane	98.2	532.2	4.9	72.8	2.75	1.136	3.12
Carbon dioxide	131.1	816.9	5.7	75.9	2.60	.611	1.59
Propylene (approx)	21.0	145.0	6.9	122.0	1.65		
Propane	1.5	104.9	13.4	168.6	1.19	6.13	7.28
Freon-22	.61	118.3	8.7	70.1	2.85	3.28	9.35
Ammonia	8.7″	109.6	11.9	479.3	.42	24.86	10.38
Freon-12	11.0″	67.5	8.3	50.1	3.99	3.91	15.60
Methyl chloride	15.9″	56.3	10.3	150.9	1.33	12.72	16.85
Sulfur dioxide	23.6″	32.6	14.5	143.4	1.40	22.2	31.00
20 F Evaporating, 80 F Condensing							
Propane	40.8	128.1	2.57	128.5	1.55	1.900	2.96
Freon-22	43.3	145.0	2.76	72.96	2.74	.937	2.57
Ammonia	33.5	138.3	3.18	485.8	.41	5.910	2.44
Freon-12	21.0	84.0	2.76	56.2	3.66	1.121	3.99
Methyl chloride	14.5	71.6	2.96	154.5	1.29	3.312	4.28
Isobutane	3.5	39.2	3.97	119.5	1.67	4.680	7.84
Sulfur dioxide	2.5	45.0	3.48	144.5	1.38	4.487	6.22
Methylamine	.2	40.5	3.71	313.6	.64		
Butane	6.3″	22.9	3.24	133.0	1.51	7.230	10.93
Freon-114	10.1″	18.3	3.44	46.6	4.29	3.000	12.87
40 F Evaporating, 100 F Condensing							
Propane	63.3	172.0	2.40	121.5	1.65	1.37	2.25
Freon-22	69.0	197.9	2.54	68.1	2.94	.656	1.93
Ammonia	58.6	197.2	2.89	467.8	.428	3.294	1.30
Freon-12	37.0	116.9	2.54	51.6	3.88	.792	3.07
Methyl chloride	27.9	104.3	2.80	149.4	1.34	2.32	3.11
Isobutane	12.2	59.0	3.74	111.0	1.80	3.22	5.80

Table 2. Comparative Refrigerant Characteristics at Various Evaporating and Condensing Temperatures (Continued)

	Evaporating pressure psig	Condensing pressure psig	Ratio of compression	Net refrigerating effect Btu/lb	Refrigerant circulated per ton lb/min	Specific volume of vapor cu ft/lb	Compressor displacement per ton cu ft/min
40 F Evaporating, 100 F Condensing							
Sulfur dioxide	12.4	69.8	3.12	138.5	1.44	2.89	4.17
Methylamine	9.3	62.8	3.23	304.8	.66		
Butane	3.0	37.5	2.95	128.0	1.56	4.88	7.63
Freon-114	.5	31.7	3.05	44.7	4.47	1.98	4.85
Freon-21	4.8″	25.3	3.28	90.0	2.22	4.13	9.18
Freon-11	15.6″	8.9	3.36	68.8	2.90	5.45	15.8
Methyl formate	20.45″	3.6	3.91	189.2	1.06	18.90	20.0
Methylene chloride	23.04″	2.97″	3.92	134.0	1.49	18.60	27.8
Freon-113	24.52″	8.59″	3.95	55.7	3.60	10.68	38.7
Water	29.67″	27.99″	7.75	1011.3	.198	2438	482.0

″ Inches of mercury vacuum.

warm liquid ahead of the expansion valve down to evaporator temperature. What is left is known as the net refrigerating effect. A low specific heat of the liquid is desirable, for the less latent heat of vaporization used in cooling the liquid, the more will be left as net refrigerating effect.

10. Density of the liquid. A refrigerant having high liquid density may first appear to be desirable because of savings in size and cost of receiver, liquid line pipe or tubes and other vessels containing liquid refrigerant. Such savings are largely offset, however, by the increased viscosity that parallels increase in density, requiring increased sizes of pipe and orifices to maintain low pressure drops.

In installations in which the evaporator is placed several floors above the condensing unit, the formation of flash gas in the liquid line is a factor that must be considered; the greater the density of the liquid refrigerant the lower the level at which this will occur.

Since refrigerant is sold by the pound, the refrigerant charge for an installation will, other factors being equal, be more for a refrigerant having high liquid density. This factor may be of minor concern in small, self-contained equipment, in which the refrigerant charge is small and the liquid line short and of small diameter.

11. Viscosity of the liquid. Pressure drop in long liquid lines may be sufficient to require larger and more costly piping in order to prevent excess flash gas and reduced capacity of the expansion valve. In large installations these factors may require serious consideration.

12. Specific heat of the vapor. If the heat capacity or specific heat of the vapor is high, the heat that is added to the saturated vapor, superheating it in the evaporator and suction line, will raise its temperature less than if the specific heat is low. As its temperature rises, the superheated vapor expands and its density decreases. This is unfavorable to the compressor capacity, for the cylinders pump more refrigerant vapor if it is saturated, cool and dense than if it is superheated, warmed and less dense. Therefore, a high specific heat of vapor is desirable. The gains effected by use of heat exchangers may offset superheating losses of the compressor only, and result in added efficiency to the system as a whole.

13. Density of the vapor. One might assume that high vapor density would permit the use of smaller tubes, piping and orifices in the pressure side of the system, but the reverse is true, due to high pressure drop resulting from high vapor density. Moderately low vapor densities are, therefore, desirable in ordinary compression systems, except those using centrifugal

compressors, for which refrigerants having high vapor densities are desirable.

14. Viscosity of the vapor. To minimize pressure drop and consequently the sizes and costs of tubes and piping, a low vapor viscosity is desirable.

15. Thermal conductivity. An ability to conduct the heat quickly from the evaporator or to the condenser in either the liquid or vapor phases is a desirable characteristic of a refrigerant, since it increases evaporator and condenser performance. Therefore high thermal conductivities of refrigerant vapor or liquid are desirable.

16. Oil solubility. Some refrigerants are highly miscible with compressor oil, consequently picking up oil from the compressor and carrying it through the system. Some oil passing the pistons and discharge valves is desirable for lubrication, and to seal the valve surfaces, yet oil serves no useful purpose in the rest of the system, with the possible exception of some types of refrigerant flow controls.

17. Horsepower per ton. With a theoretically perfect cycle, all refrigerants would be equal as to power required per ton of refrigeration. In actual practice there is very little difference in horsepower per ton between the commonly used refrigerants except in the case of carbon dioxide. Thus this factor is much less important than most of the other characteristics.

In the smaller machines the unit ordinarily used instead of horsepower per ton is Btu per watt, which is of inverse character to horsepower per ton.

18. Stability. Ideally, any refrigerant used in a compression system should be absolutely stable and inert, that is, it should have no chemical reaction with any material—oil, gas (including air), metal, moisture or other substance of any nature with which it might come in contact, nor be chemically affected by any extremes of temperature to which it might be subjected.

19. Dielectric strength of vapor. This characteristic is important only in the "hermetic" units, in which the refrigerant comes into direct contact with the field coils of the motor driving the compressor. If the dielectric strength of the refrigerant vapor at suction pressure and the temperature are low, possibility of short-circuits between electrical conductors is increased, although in most designs no bare conductors are exposed to the refrigerant, all being fully insulated. Table 3 shows the relative dielectric strengths of the vapor at atmospheric pressures of several of the refrigerants that are frequently used in hermetic motor-compressor units.

20. Leakage tendency. The smaller the molecule of the refrigerant, the harder it is to prevent it from leaking. The size of the molecule is proportional to the square root of the molecular weight. Pressure, capillarity, viscosity, density and rate of diffusion are also factors that affect the tendency of a gas to leak, but these being equal, the refrigerant with a high molecular weight has a lower leak tendency.

21. Washing effect. The hydro-carbon refrigerants are all excellent oil and grease solvents, which makes it especially important that castings particularly, but other materials as well, be degreased and very carefully cleansed after machining and fabrication. Otherwise the continual washing of the parts by the refrigerant or refrigerant-oil mixture circulating in the system will dissolve the oils on the inner surfaces of the system or soaked into the pores of castings and will loosen scale, dirt and other impurities which will clog screens, foul orifices and aggravate sludge formation.

The action of the refrigerant on gasket material and the nature of the impregnation of seal surfaces may lead to difficulty in "holding" the refrigerant.

22. Odor. An easily detectable odor has an advantage in leak detection in that maintenance personnel will become aware of leakage and make the necessary repair, or that the user of untended equipment will know when to call the repairman. An odor can serve also as a warning to occupants to avoid the refrigerant. On the other hand, a pronounced odor, especially one of an irritating nature, may cause hysteria and panic in congested public places or institutions.

There are but few refrigerants that have a pronounced odor that is easily detected. These include ammonia and sulfur dioxide, and, to a lesser degree, ethyl chloride and ethane. Most of the hydrocarbons, including the halogens, methyl chloride, methyl-

Table 3. Applications, Leak Methods and Oil Miscibility

	Application for which the refrigerant is used or is suitable							Applicable leak test methods[4,6]	Miscibility with oil	Dielectric strength of vapor[5]
	Industrial with attendance[1]	Commercial up to 25 hp[1]	Air conditioning[1]	Household[1]	Evaporator temperature range[2]	Type of compressor[3]	Air or water-cooled			
Methane	Ltd.	None	None	None	Ult.	Recip.	Water	2, 4	Yes	1.00
Ethylene	Ltd.	Few	None	None	Ult.	Recip.	Water	2, 4	Yes	1.21
Nitrous oxide	Ltd.	Few	Suit.	None	Ult.	Recip.	Water	2, 4	No	1.14
Ethane	Ltd.	Few	None	None	Ult.	Recip.	Water	2, 4	Yes	
Carbon dioxide	Ext.	Few	Ltd.	None	Low & Med.	Recip.	Water	2, 4	No	.88
Propylene	Ltd.	Ltd.	None	None	Ult. & Low	Recip.	Water	2, 4	Yes	
Propane	Ltd.	Ltd.	None	None	Ult. & Low	Recip.	Water	2, 4	Yes	
Freon-22	Ltd.	Ltd.	None	Suit.	Ult. & Low	Recip.	Water	1, 3, 5	Yes	1.31
Ammonia	Ext.	Ext.	Ltd.	Ext.	Low to High	Recip.	Water	1, 3, 6	No	.82
Freon-12	Ext.	Ext.	Ext.	Ext.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3, 5	Yes	2.40
Methyl chloride	Ltd.	Ext.	Ltd.	Ext.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3, 5	Yes	1.06
Isobutane	Few	Ltd.	None	Ltd.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3	Yes	
Sulfur dioxide	Few	Ext.	Few	Ext.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3, 6	No	1.90
Methylamine	Few	Suit.	None	Suit.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3	Yes	
Butane	Few	Suit.	None	Ltd.	Med. & High	Rec.-Rot.	Wat. or Air	1, 3	Yes	
Freon-114	None	None	None	Ext.	Med. & High	Rotary	Air	1, 3, 5	Yes	2.80
Freon-21	Few	None	None	None	Med. & High	Rotary	Air	1, 3, 5	Yes	1.33
Ethyl chloride	None	Few	None	Ltd.	Med. & High	Rotary	Air	1, 3, 5	Yes	1.00
Ethylamine	Suit.	None	None	None	High	Rot.-Cen.	Air	1	Yes	
Freon-11	Suit.	Few	Ext.	Ext.	High	Rot.-Cen.	Water	1, 5	Yes	3.00
Methyl formate	Suit.	None	None	None	High	Centrif.	Water	2, 5	Yes	
Ethyl ether	Suit.	None	None	None	High	Centrif.	Water	2	Yes	
Methylene chloride	Suit.	None	Ext.	None	High	Centrif.	Water	2, 5	Yes	1.11
Freon-113	Suit.	None	Ext.	None	High	Centrif.	Water	2	Yes	2.60
Dichlorethylene	None	None	None	None	High	Centrif.	Water	2, 5	Yes	
Trichlorethylene	None	None	None	None	High	Centrif.	Water	2, 5	Yes	
Water	Ltd.	None	Ext.	None	High	Cen.-Jet.	Water	2	No	

[1] Ext.—Extensive; Ltd.—Limited; Few—Few, if any; None—None; Suit.—Suitable but few if any.
[2] Ult.—Ultra-Low, −130 to −75 F; Low—Low, −75 to 0 F; Med.—Medium, 0 F to 30 F; High—High, 30 F to 50 F.
[3] Rec.—Reciprocating; Cen.—Centrifugal; Rot.—Rotary.
[4] Numbers refer to those described on page 195.
[5] When using the Halide torch on equipment charged with USA Group 2 refrigerants (see Table 4) make certain that good ventilation is provided. Do not use Halide torch for USA Group 3 refrigerants.
[6] Ethyl chloride as unity.

ene chloride, dichlorethylene and trichlorethylene, have an odor which is not readily detectable. It is a slightly sweet, ethereal odor similar to that of natural gas. Methyl chloride may be obtained with 1% acrolein in it to give it a pungent, irritating odor.

Carbon dioxide, the Freon group, and water when used as a refrigerant have practically no odor. Also, they are essentially harmless, so no irritating odor is needed to serve as a warning of leakage. The products of decomposition of the nonflammable or slightly flammable halogenated hydrocarbons by a flame, or in some cases by hot wire, have pungent, detectable odors that serve as warning agents against those toxic gases produced by decomposition.

Refrigerants and absorbents in an absorption system. A refrigerant for use in an absorption system should, in general, have the same desirable qualities as a refrigerant for a compression system, and in addition it must be suitable for use with an absorbent so selected that in the combination:

(a) The vapor pressures of the two at the generator temperature shall be widely divergent, in order to obtain as nearly complete fractionation as possible in the condenser and rectifier.
(b) Temperature-pressure relations should conform to practical absorber and generator temperatures and pressures.
(c) The refrigerant must have high solubility in the absorbent at the absorber temperature and pressure (as governed by the condensing medium), and low solubility at the generator temperature and pressure.
(d) In most absorption equipment the solvent must remain liquid at absorber and generator temperatures and pressures.
(e) The refrigerant and the solvent must remain stable with one another within the generator-evaporator range of

temperatures. The solvent should have low specific heat, surface tension and viscosity, and must be neutral to the materials used in the equipment.

There are many known combinations of refrigerant and absorbent that could be used in an absorption system, but the only ones which have been commonly used are ammonia-water, methylene chloride-dimethyl ether of tetraethylene glycol, water-lithium chloride, water-lithium bromide, ammonia-calcium chloride, ammonia-strontium chloride, and sulfur dioxide-silica gel.

II. Reaction with Moisture

Water in a refrigerating system has two effects, either of which is sufficiently injurious to warrant the utmost care in preventing the entrance of moisture into a system, or in reducing to the lowest practical minimum any moisture that may have been introduced into the system.

1. Water unites in varying degrees with most of the refrigerants, which with, or in addition to, any air that may be in the system, in turn react on the oil and the various materials of which the equipment is made—metals, plastics, fibers, and various compounds in valve packing, gaskets, and electrical insulation. If the refrigerant holds considerable moisture in solution (as do ammonia, sulfur dioxide and carbon dioxide), this moisture is still able to cause **corrosion** to some of the materials with which it comes in contact. Therefore, because no trouble arises from freeze-ups at the expansion valve, it is not necessarily true that any moisture that is in the system is doing no harm. It is not uncommon to open a system and find damaging corrosion, although the system has operated with no expansion valve freeze-up trouble. Corrosion first makes itself known by pitted and damaged valve needles and seats; if not checked, it may extend to seal leakage, and damage to bearings, bushings, pins, shafts and other finely finished surfaces. Rust and scale washed from corroded surfaces may clog screens, filters and valve ports and restrict the flow of refrigerant.

Sludges and precipitates may form as a result of chemical reaction between the refrigerant and the oil, and in the presence of impurities, foreign substances or adulterants and their reactions are aggravated and amplified by moisture and/or air in the system. It is wise to exclude everything from the system other than the refrigerant and the oil of specifications approved by the manufacturer of the equipment, especially the manufacturer of the compressor.

2. After the refrigerant has absorbed all the moisture of which it is capable, any further moisture in the system exists as free water, which freezes into ice in those portions of the system below 32 F. This usually occurs first at the orifice in the expansion valve or float valve or in the capillary tube. **Formation of ice** causes restriction or stoppage of refrigerant flow and refrigeration ceases. The valve then warms up, the ice melts, and refrigeration is resumed. This may reoccur, causing "sticky" operation of the valve, or the water may pass on into the evaporator where it may freeze and remain as small droplets of ice. It may remain frozen in the evaporator for a considerable period until the evaporator is defrosted, when it returns to the water state and again circulates through the condensing unit to the expansion valve. It is, therefore, imperative that moisture be excluded or removed from the compression system.

For full information on moisture in the system and methods for its removal, see Chapter 18.

III. Leak Detection

The most frequent cause of service calls on refrigerating systems is "leaks in the system," usually caused by:

a. In portions of the system in which the pressure is **above atmospheric pressure,** the refrigerant will leak out of the system, eventually creating a shortage and necessitating replacement of the amount of leakage.

b. In other portions in which the pressure may be **below atmospheric** (on a vacuum), the leakage will be of air into the system. In most climates, air contains enough moisture to cause trouble in time inside the system. In either case the leak must be found and repaired and refrigerant added as required, or air and moisture removed.

The **methods** for **detecting leaks** differ according to the refrigerant used, the size and type of equipment being tested, and the facilities available at the test location.

1. In factories or well equipped shops, a favorite method is to put a pressure of 200 or 300 psi (depending upon the strength of the equipment) of **dried air** or some inert gas such as carbon dioxide or nitrogen, in the equipment and then submerge the equipment in a brightly lighted tank of water or kerosene. Location of a leak will be discovered by a stream of bubbles rising from it. The tank must be brilliantly lighted (preferably from beneath the water surface as well as from above) and the liquid clear; otherwise a fine stream of bubbles from a very small leak may escape detection.

2. Large equipment is frequently tested by putting **nitrogen** or **carbon dioxide** (never oxygen) in it to a definite pressure and letting it stand for at least 24 hr. Constancy of gage readings indicates absence of leaks. Allowance must be made for variations in pressure due to variations in ambient temperature.

3. Another method of testing is by **pumping a vacuum** on the equipment by means of an auxiliary vacuum pump (preferable to using the refrigerating compressor for this purpose), with the pump discharging through a tube, the outlet end of which is submerged in clear oil in a glass vessel. After discontinuance of bubbles, indicating a high degree of vacuum, the equipment is allowed to stand for an hour or two with the vacuum pump idle. The pump is then started. Issuance of bubbles indicates that air has leaked in during the idle period. In larger equipment, constancy of gage readings may be acceptable instead of the "no bubble" method.

4. Leaks may also be detected by putting a **refrigerant** or **air pressure** of at least 50 psi in the equipment, and then spreading a thick, ropey soap solution or heavy, clear oil over joints or other suspected places. Presence of bubbles indicates and locates the leaks.

5. Another method is to put the equipment under a pressure of 50 psi or more of one of the halogen refrigerants, and then test each joint or suspected place with the exploring tube of a **"halide torch"** burning either methyl alcohol or Prestolite gas. This is a very accurate method which is becoming increasingly popular.

If desired, the pressure of the refrigerant may be built up by adding with it carbon dioxide or nitrogen to a pressure high enough to accentuate small leaks. If this method is used with methyl chloride or other flammable halogen, due caution must be exercised to provide sufficient ventilation to prevent ignition of the refrigerant by the flame of the torch.

6. The **"smoke"** method is usable with ammonia and sulfur dioxide. If a swab dipped into "stronger" ammonia (about 25% solution of aqueous ammonia) is brought near a leak from which sulfur dioxide is emerging, a white smoke is easily visible, indicating the presence and relative location of the leak.

Conversely, ammonia equipment may be tested for leaks by burning a sulfur candle (thus producing sulfur dioxide) and exploring all suspected places, a leak evidencing itself by the "smoke." If the leak is large enough to pollute the nearby air considerably, its exact location may be difficult to determine by this method.

To detect or locate very small ammonia leaks, a piece of porous paper soaked in a phenolphthalein solution or Nessler solution, is applied to the suspected joint. Appearance of a red spot in the paper indicates the presence of a leak.

For practical purposes of maintenance cost and safety, a refrigerant should have some simple, positive method of locating leaks. The soap-and-water method can be used with any refrigerant but it is tedious, time-consuming and thus expensive. The "smoke" test method used with ammonia and sulfur dioxide is excellent.

The introduction of the halide torch has greatly simplified the location of leaks of the halogens. Its use makes it possible to detect and locate minute leaks of these refrigerants quickly and effectively.

IV. Action on Materials of which the Equipment is Constructed

The hydrocarbons attack natural **rubber,** causing it to become soft and to swell and to form blisters. For this reason the synthetic rubbers of the chloroprene or

neoprene types, which are less vulnerable, are used.

In hermetically sealed systems the refrigerant comes into direct contact with the motor winding. The hydrocarbons are very effective solvents for many of the **electrical insulating varnishes** or **binders** with which the insulating fabrics are impregnated. Much trouble has been experienced from the refrigerant washing out these varnishes and depositing them, sometimes with fragments of the fabrics, on valve seats and in valve mechanisms or small orifices. Insulations of bakelite or glass fibre have been successfully used in such applications.

Moisture and/or air in the system seem to aggravate any action that the anhydrous refrigerant alone might have on metals, fibrous materials, gaskets, oils or other substances which may be exposed to the refrigerant. Ammonia readily unites with water to form aqueous ammonia, a strong alkaline solution that attacks **copper, brass** or other cupreous alloys.

Inasmuch as in ordinary practice there is some moisture in the system, copper or copper alloys are not used in ammonia systems where the ammonia may come into contact with them.

Another well known instance of a refrigerant attacking materials of which the system is made is the case of sulfur dioxide that has been exposed to moisture. Sulfurous acid is formed, which strongly attacks the metals in the system, especially the **ferrous metals.** Depending upon the amount of moisture present, the resulting corrosion may range from mere discoloration of the metals up to "freezing" of pistons in cylinders, shafts in bearings or other damage, even to the extent of being entirely irreparable.

If free **aluminum, magnesium** or **zinc** or their alloys or die-castings containing them, come into contact with methyl chloride, they are strongly attacked. (There is no action of methyl chloride on activated alumina, the drying agent.) The resulting compounds react with the oil to form a heavy black gummy precipitate that stops up screens and oil passages and fouls valves, rings, bearings, and other working parts.

The action of methyl chloride on aluminum produces aluminum tri-methyl, a highly inflammable gas **that may ignite with explosive violence** when merely exposed to the air or abnormally high pressures. Aluminum or its alloys must not be used with methyl chloride, and if such a system is found, **much caution must be used** in opening it for repair.

Methyl chloride and the Freons attack magnesium, and in additon methyl chloride attacks zinc. (Zinc is sometimes used in dehydrators with sulfur dioxide as a "neutralizer" for sulfurous acid.) Magnesium and zinc are corroded by such action, but there are no known cases of formation of dangerous compounds. To some degree all of the halogenated hydrocarbons, which include methyl chloride, ethyl chloride, the Freons and methylene chloride, unite with water even in small quantities to form acids, particularly hydrochloric acid, which will readily attack any of the metals commonly used in refrigerating equipment, as well as some of the materials of which gaskets, electrical insulation, and other non-metallic materials may be composed.

Copper plating. In systems employing methyl chloride, some of the Freons or other halogenated hydrocarbon refrigerants, certain types of oils have elements in them that dissolve copper from tubing or other parts, as the oil is being circulated throughout the system. The copper thus picked up is deposited on hot surfaces such as bearings, shafts, bushings, pins, pistons, rings, cylinders, valves and valve seats. There the copper accumulates, in time causing seizure of frictional parts or leakage of valves.

Moisture in the system aggravates this action but is not its direct cause. The use of a more highly refined oil with a small content of unsaturated hydrocarbons, greater stability and low Sligh oxidation number, plus thoroughly drying the system, will practically eliminate instances of "copper plating."

All of the statements above emphasize the necessity of properly drying the system and keeping it dry, and of introducing into the system only clean, dry refrigerant and suitable oil.

The straight hydrocarbons such as ethane, propane, butane, etc., containing

carbon and hydrogen only, are less susceptible to formation of harmful compounds in the presence of moisture and air than the halogens that include fluorine or chlorine atoms. Therefore moisture in systems using the straight hydrocarbons may present a less serious problem, except that trouble will continue to result from freeze-ups at the expansion valve.

V. Safety Considerations (Table 4)

In choosing a refrigerant for a specific application, safety must be considered as a relative term. A refrigerant such as ammonia might not, because of its explosive and toxic characteristics, be considered safe for an automatic, small commercial or household application, yet is quite acceptable, from a safety viewpoint, for a large cold storage or ice-making plant in which the equipment is under constant surveillance by experienced operating personnel. The quantity of refrigerant used, the size of the room into which it might leak, whether or not flames are present, the type of occupancy and the odor of the refrigerant also have a direct bearing on the relative safety of the installation. A large multiple installation using methyl chloride would be unsafe in an institution, hotel or apartment house, while a small household refrigerator or freezer with only a pound or two of methyl chloride as the total charge, would be safe for all practical purposes.

Safety may be considered with respect to toxicity, inflammability and explosiveness.

1. Toxicity. In a strict sense, all of the refrigerants except air, which is rarely used, may cause suffocation by oxygen deficiency.

The term "toxic" is usually applied to those refrigerants which are actually injurious to human beings—those that may cause death or serious injury even though they are mixed in air in small enough proportions that sufficient oxygen can be obtained from the air-refrigerant vapor mixture.

The extent and permanence of the injury depend upon: (1) the nature of the refrigerant, i.e., its relative toxicity; (2) the percentage of the refrigerant in the air; (3) the duration of time that the person is subjected to the mixture. Obviously, the greater the percentage of concentration of refrigerant in the air and the longer the person is subjected to the vapor, the

Table 4. Relative Safety of Refrigerants

Refrigerant	A.S.A. Safety Code group classification	Nat'l Fire Underwriters group number	Toxicity Lethal or serious injury[2]					Inflammable or explosive limits of concentration in air
			Refrigerant in air			Products of decomposition by flame		
			Duration of exposure hours	Percentage by volume	Pounds per 1,000 cu ft	Duration of exposure min	Percentage by volume[4]	Percentage by volume
Methane	3[1]	+5						4.9–15.0
Ethylene	3[1]	+5						3.0–25.0
Nitrous oxide			8	.0025				Non-Flam.
Ethane	3	5	2	37.5–51.7				3.3–10.6
Carbon dioxide	1	5	½ to 1	29–30	33.2–34.3			Non-Flam.
Propane	3	5	2	37.5–51.7	42.4–58.5			2.3–7.3
Freon-22	1[1]	5A				16	1.0	Non-Flam.
Ammonia	2	2	½	.5–.6	.221–.256			16.0–25.0
Freon-12	1	6	2	28.5–30.4	89.6–95.7	20	1.0	Non-Flam.
Methyl chloride	2	4	2	2–2.5	2.62–3.28	30	2.4	8.1–17.2
Isobutane	3	+5						1.8–8.4
Sulfur dioxide	2	1	½	.7	1.165			Non-Flam.
Butane	3	5	2	37.5–51.7				1.6–6.5
Freon-114	1	6	2	20.1–21.5	90.5–96.8	15	1.0	Non-Flam.
Freon-21	1		½	10.2	27.1			Non-Flam.[2]
Ethyl chloride	2	4	1	4.0	6.72	18	2.0	3.7–12.0
Freon-11	1	5	2	10	35.7	5	1.0	Non-Flam.
Methyl formate	2	3	1	2–2.5	3.12–3.9			4.5–20.0
Methylene chloride	1	4A	½	5.1–5.3	11.25–11.7	20	1.0	Non-Flam.
Freon-113	1[1]	4	1	4.8–5.2	23.3–25.2	10	1.2	Non-Flam.
Dichlorethylene	2	4	1	2–2.5	5.04–6.3	5	2.1	5.6–11.4

[1] Unofficial.
[2] Very slightly flammable, but for practical purposes considered non-flammable.
[3] To guinea pigs.
[4] Initial concentration.

greater will be the injury, or, in fatal instances, the sooner death will occur. The National Board of Fire Underwriters has tested most of the commonly used refrigerants and classified them into groups according to their toxicity. Group 1 includes the most toxic, with toxicity decreasing as the group number increases, so that Group 6 includes those refrigerants which are harmless, except when a high concentration might dangerously exclude oxygen from the person exposed.

In Table 4, the refrigerants are classified as to toxicity according to the Underwriters' tests.

Mixtures of toxic vapors and air may attack the membranes of the lungs, nose, eyes or even the skin, and/or they may be absorbed into the blood and carried to other parts of the body to which they may be injurious.

Some refrigerants which in their normal state mixed with air, even in heavy percentages, are practically harmless, are decomposed by contact with a flame and form products that are highly toxic and even in low concentrations may cause fatal or serious injury in less than one-half hour. The halogens, including methyl chloride and the Freons, are in this category. A room into which these refrigerants are known or suspected of having leaked, if a flame or an electric heating element is present, should be well ventilated before entering.

2. Inflammability and explosiveness. Many of the refrigerants that have been in common and widespread use for many years will burn if mixed with just the right percentages of air, and yet fires or explosions from this cause have been extremely rare. This hazard is slight because: (1) the inflammable vapor-air mixture must be within a rather narrow limit of concentration of the vapor; (2) the refrigerant vapor is usually under sufficient pressure that it tends to blow out the flame; and (3) due to offensive odor or lack of refrigeration, the loss of refrigerant is usually detected before an inflammable concentration develops.

In Table 4, the various refrigerants that have been or are in fairly common use are classified as to inflammability. Some refrigerants, such as most of the Freons (those from which the hydrogen atom has been eliminated), will not burn nor will they support combustion. Others will burn, but only feebly and within close limits of concentration of air. Ammonia and methyl chloride fall in this second group. The third group includes those that are quite inflammable, such as methane, ethylene, ethane, propylene, propane, butane, and isobutane.

In general, those halogenated hydrocarbons, such as Freons-11, 12, 113 and 114, in which the hydrogen atoms have been entirely replaced by fluorine and/or chlorine atoms, are the least toxic and the least inflammable. These hydrocarbons in which the hydrogen atoms have not been entirely removed are moderately toxic and slightly flammable. The straight hydrocarbons are the most toxic and the most flammable (see bibliography, page 199).

The ASRE-sponsored ASA "American Standard Safety Code for Mechanical Refrigeration" is usually the basic model on which states and municipalities compile their codes. Despite this fact, there is much variance between the provisions of the various local codes. Since these refrigeration codes are units of local building or safety codes, they must be taken into consideration and followed when installing, repairing or operating refrigeration or air-conditioning equipment in any area under the jurisdiction of a local code.

VI. Cost

Other factors being equal, low cost of a refrigerant is quite desirable. However, the price per pound must be evaluated on the basis of: (1) the number of pounds that must be circulated per minute to provide the necessary refrigeration (dependent upon the "net refrigerating effect" per pound of refrigerant); (2) the density of the liquid. Since refrigerants are sold by the pound rather than by unit of volume, the cost advantage is with those having high net refrigerating effects and low liquid densities. Such refrigerants also permit smaller and less costly liquid receivers and piping.

VII. Type of Compressor

In late years, advancements in metallurgy have made new alloys available and

modern precision machining processes have allowed finer finishes, so that much greater rubbing speeds than heretofore are entirely practical. This has resulted in compressors driven directly by induction electric motors at full motor speed. Reduction of size, further aided by a multiplicity of cylinders, has permitted cost reduction and has stimulated the development and use of self-contained and portable fixtures. These advances in compressor design have affected the requirements of the refrigerant, putting increased emphasis on horsepower per ton, amount of refrigerant circulated, vapor density, viscosity and electrical resistance, and reducing the importance of compression ratio and evaporator pressure.

VIII. Methods of Handling

During storage, transport, transfer from larger to smaller cylinders, and installations, service and maintenance operations, refrigerants must be handled by experienced workmen who are familiar with their nature and with the precautions that must be taken.

Reference should be made to instructions issued by the refrigerant manufacturers on the handling of the refrigerant that they produce, and on the safe quantities of refrigerant that may be put into their particular containers.

The Interstate Commerce Commission regulates the type, size, strength, test pressures, design, marking and retest period of refrigerant containers for interstate shipment of the various refrigerants. ICC regulations must be complied with when shipping and handling refrigerants.

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9. BRINE PROPERTIES

1. Brine. Tabular material on brines begins with the two common salts, calcium chloride and sodium chloride. The properties of pure calcium chloride brine and sodium chloride brine are given in **Tables 1** and **6**. For commercial grades use the formulae indicated in the footnotes to these tables. **Tables 2** and **7**, which give for these salts the enthalpy for liquid and frozen brines, may be used for computa-

Table 1. Properties of Pure Calcium Chloride Brine

Pure $CaCl_2$ % by weight	Specific gravity 60 F	Beaumé density 60 F	Deg salometer 60/60 F	Specific heat 60 F Btu per lb F	Freezing point F	Weight per gallon*			Weight per cubic foot†		
						$CaCl_2$ lb/gal	Water lb/gal	Brine lb/gal	$CaCl_2$ lb/cu ft	Water lb/cu ft	Brine lb/cu ft
0	1.000	0.0	0.0	1.000	32.0	0.000	8.34	8.34	0.00	62.40	62.40
5	1.044	6.1	24.4	0.924	27.7	0.436	8.281	8.717	3.26	61.89	65.15
6	1.050	7.0	28.0	0.914	26.8	.526	8.234	8.760	3.93	61.59	65.52
7	1.060	8.2	32.8	0.898	25.9	.620	8.231	8.851	4.63	61.51	66.14
8	1.069	9.3	37.2	0.884	24.6	.714	8.212	8.926	5.34	61.36	66.70
9	1.078	10.4	41.6	0.869	23.5	.810	8.191	9.001	6.05	61.22	67.27
10	1.087	11.6	46.4	0.855	22.3	0.908	8.168	9.076	6.78	61.05	67.83
11	1.096	12.6	50.4	0.842	20.8	1.006	8.137	9.143	7.52	50.81	68.33
12	1.105	13.8	55.2	0.828	19.3	1.107	8.120	9.227	8.27	60.68	68.95
13	1.114	14.8	59.2	0.816	17.6	1.209	8.093	9.302	9.04	60.47	69.51
14	1.124	15.9	63.6	0.804	15.5	1.313	8.064	9.377	9.81	60.27	70.08
15	1.133	16.9	67.6	0.793	13.5	1.418	8.034	9.452	10.60	60.04	70.64
16	1.143	18.0	72.0	0.779	11.2	1.526	8.010	9.536	11.40	59.86	71.26
17	1.152	19.1	76.4	0.767	8.6	1.635	7.984	9.619	12.22	59.67	71.89
18	1.162	20.2	80.8	0.756	5.9	1.747	7.956	9.703	13.05	59.46	72.51
19	1.172	21.3	85.2	0.746	2.8	1.859	7.927	9.786	13.90	59.23	73.13
20	1.182	22.1	88.4	0.737	− 0.4	1.970	7.883	9.853	14.73	58.90	73.63
21	1.192	23.0	92.0	0.729	− 3.9	2.085	7.843	9.928	15.58	58.61	74.19
22	1.202	24.4	97.6	0.716	− 7.8	2.208	7.829	10.037	16.50	58.50	75.00
23	1.212	25.5	102.0	0.707	−11.9	2.328	7.792	10.120	17.40	58.23	75.63
24	1.223	26.4	105.6	0.697	−16.2	2.451	7.761	10.212	18.21	58.00	76.32
25	1.233	27.4	109.6	0.689	−21.0	2.574	7.721	10.295	19.24	57.70	76.94
26	1.244	28.3	113.2	0.682	−25.8	2.699	7.680	10.379	20.17	57.39	77.56
27	1.254	29.3	117.2	0.673	−31.2	2.827	7.644	10.471	21.13	57.12	78.25
28	1.265	30.4	121.6	0.665	−37.8	2.958	7.605	10.563	22.10	56.84	78.94
29	1.276	31.4	125.6	0.658	−49.4	3.090	7.565	10.655	23.09	56.53	79.62
29.6	1.290	32.6	—	0.655	−59.8	3.16	7.59	10.75	23.65	56.80	80.45
30	1.295	33.0	—	0.653	−50.8	3.22	7.58	10.80	24.06	56.70	80.76
32	1.317	34.9	—	0.640	−19.5	3.49	7.49	10.98	26.10	56.04	82.14
34	1.340	36.8	—	0.630	+ 4.3	3.77	7.40	11.17	28.22	55.35	83.57

Deg Beaumé heavy $= \frac{\text{deg } CaCl_2 \text{ sal.}}{4}$ * Wt of flake $CaCl_2 = \frac{\text{wt of pure } CaCl_2}{.785}$ † Wt of solid $CaCl_2 = \frac{\text{wt of pure } CaCl_2}{.74}$.

Table 2. Enthalpy ($CaCl_2$)

Enthalpy (Btu per lb from 32 F where it is 200 minus heat of solution)[1]

Sp gr		1.08	1.12	1.16	1.20	1.24	1.28
Lb $CaCl_2$ per 100 lb solution		9.2	13.5	17.6	21.5	25.1	28.7
Freezing point, deg F		23.2	16.5	7.0	−5.8	−21.5	−44.3
Lb solution per gal, 60 F		9.01	9.35	9.68	10.01	10.35	10.68
t, deg F							
−65	Solid	41.14	54.70	67.69	80.22	92.27	103.80
−60	Solid	43.07	56.59	69.54	82.01	94.02	105.52
−59.8	Solid	43.15	56.65	69.61	82.08	94.08	105.58
−59.8	Liquid and Solid	60.60	82.30	103.13	123.16	142.43	160.93
−55	Liquid and Solid	63.34	85.40	106.57	126.93	146.52	165.30
−50	Liquid and Solid	66.25	88.67	110.23	130.94	150.87	169.97
−45	Liquid and Solid	69.31	92.20	114.18	135.30	155.64	175.13
−40	Liquid and Solid	72.57	95.96	118.44	140.04	160.82	180.24
−35	Liquid and Solid	76.01	100.02	123.08	145.25	166.58	183.41
−30	Liquid and Solid	79.71	104.43	128.15	150.96	172.90	186.59
−25	Liquid and Solid	83.72	109.24	133.77	157.34	180.01	189.77
−20	Liquid and Solid	88.07	144.63	140.13	164.64	187.57	192.95
−15	Liquid and Solid	93.01	120.76	147.41	173.04	190.87	196.15
−10	Liquid and Solid	98.64	127.91	156.04	183.06	194.19	199.35
−5	Liquid and Solid	105.07	136.36	166.42	192.45	197.51	202.55
0	Liquid and Solid	112.60	146.29	178.67	195.95	200.84	205.77
5	Liquid and Solid	121.63	158.44	193.87	199.45	204.18	208.99
10	Liquid and Solid	133.07	174.13	198.38	202.97	207.53	212.21
15	Liquid and Solid	149.39	197.04	202.11	206.49	210.89	215.45
20	Liquid and Solid	176.58	201.63	205.86	210.03	214.27	218.70
25	Liquid	210.49	205.63	209.62	213.58	217.75	221.94
30	Liquid	205.79	209.65	213.38	217.14	221.04	225.20
35	Liquid	210.12	213.67	217.15	220.70	224.44	228.46
40	Liquid	214.44	217.70	220.94	224.28	227.85	231.74
45	Liquid	218.78	221.74	224.74	227.87	231.27	235.02
50	Liquid	223.11	225.79	228.54	231.47	234.69	238.31
55	Liquid	227.46	229.85	232.35	235.08	238.13	241.62
60	Liquid	231.82	233.91	236.17	238.70	241.58	244.93
65	Liquid	236.18	237.98	240.00	242.32	245.03	248.24
70	Liquid	240.55	242.07	243.83	245.95	248.50	251.57
75	Liquid	244.92	246.17	247.67	249.59	251.97	254.86

Table 3. Calcium Chloride Density
Specific gravity refers to water at 60 F

	Density (gr per cc)			
% Salt	−4 F	14 F	32 F	50 F
5			1.043	1.042
6			1.052	1.051
7			1.061	1.060
8			1.071	1.069
9			1.080	1.078
10			1.089	1.087
11			1.098	1.096
12			1.108	1.105
13			1.117	1.115
14			1.127	1.124
15		1.139	1.137	1.134
16		1.149	1.146	1.143
17		1.159	1.156	1.153
18		1.169	1.166	1.163
19		1.180	1.176	1.173
20		1.190	1.186	1.183
22	1.215	1.211	1.207	1.203
24	1.236	1.232	1.228	1.224

tion of heat loads with ordinary brine as well as for any problems dealing with frozen brines.[1] **Tables 3** and **8** show the density, commonly used as the measure of salt concentration. Viscosities are given in **Tables 4** and **9**. Chapter 6, on fluid flow, presents additional data on viscosity.

Tables 5 and **10** give data on the vapor pressures existing over salt solutions, in equilibrium with them. The temperature corresponding to the vapor pressure given, can be considered the boiling point of the solution at that pressure. A common example of the use of Tables 5 and 10 is to determine the amount of water to be taken up by a salt in coming into equilibrium with the vapor pressure of the moisture existing in the atmosphere. These tables also show what temperature would result from the establishment of a stated pressure above a solution, a principle used in some refrigerating machines to establish temperatures below the ordinary.

An elementary method of producing a lowering of temperature is by the mixing of salts. The effect of several mixtures is indicated in **Table 11.**

2. Freezing of brines. A solution of any salt in water has a certain concentration at which the freezing point is lowest. A solution of this concentration is called a **eutectic mixture,** or **cryohydrate,** and the temperature at which it freezes is the eutectic temperature. A solution at any other concentration will start to freeze at a higher temperature, and as the temperature is lowered, will freeze out pure ice or salt, the remaining liquid approaching the eutectic concentration, until the final freezing takes place at the eutectic point.

Referring to **Fig. 1,** if a solution of concentration x, less than the eutectic, at a temperature above 32 F, is cooled, it will not solidify when 32 is reached (point A), but will continue to cool as a liquid until point B is reached. At this point ice crystals of pure water will begin to form, accompanied by the removal of their latent heat. The removal of this water from the solution increases the concentration of the residual solution. As the temperature is lowered these crystals continue to form, and the mixture of ice crystals and brine solution forms a slush. When point C is reached, there is a mixture of ice crystals, C_2, and brine solution of concentration C_1, in the proportions of l_1 parts of brine to l_2 parts of ice crystals in (l_1+l_2) parts of mixture. When the process has continued to

Table 4. Calcium Chloride Viscosity

	Viscosity (in centipoises)[8]				
%	−13 F	−4 F	14 F	32 F	50 F
1.6				1.804	1.319
3.4				1.854	1.358
5.0				1.922	1.407
6.8				2.001	1.461
8.7				2.104	1.515
10.5				2.211	1.608
12.2				2.334	1.702
14.1				2.491	1.809
16.0			4.217	2.677	1.947
17.8			4.521	2.888	2.113
19.8			4.894	3.119	2.310
21.8		8.522	5.335	3.413	2.560
23.7	11.499	9.326	5.850	3.761	2.844
25.7	12.901	10.513	6.585	4.217	3.197
27.6	14.955	12.013	7.571	4.854	3.678
29.7		14.269	8.860	5.688	4.335
34.0				8.164	6.119
37.2				10.861	7.870

Table 5. Calcium Chloride Vapor Pressure

Vapor pressure above solutions, mm Hg[6]

Sp gr at 60 F	Temperature, deg F													
	42.8	39.2	35.6	32	28.4	24.8	21.2	17.6	14	10.4	6.8	3.2	−.4	−4.0
1.00	7.01	6.10	5.29	4.58	3.88	3.28	2.76	2.32	1.95	1.63	1.36	1.13	0.94	0.77
1.01	6.97	6.07	5.26	4.56										
1.02	6.93	6.03	5.23	4.53										
1.03	6.90	6.00	5.20	4.50										
1.04	6.86	5.96	5.17	4.48	3.86									
1.05	6.81	5.92	5.13	4.45	3.83									
1.06	6.76	5.88	5.10	4.42	3.81									
1.07	6.71	5.84	5.06	4.38	3.78	3.26								
1.08	6.66	5.79	5.02	4.35	3.75	3.23								
1.09	6.60	5.74	4.98	4.31	3.72	3.21	2.76							
1.10	6.56	5.70	4.95	4.28	3.69	3.18	2.74							
1.11	6.49	5.65	4.90	4.24	3.66	3.15	2.71	2.32						
1.12	6.41	5.58	4.84	4.19	3.61	3.11	2.68	2.29						
1.13	6.36	5.53	4.80	4.15	3.58	3.09	2.65	2.27	1.94					
1.14	6.28	5.46	4.74	4.10	3.54	3.05	2.62	2.25	1.92					
1.15	6.20	5.40	4.68	4.05	3.49	3.01	2.59	2.22	1.90	1.62				
1.16	6.13	5.33	4.62	4.00	3.45	2.97	2.56	2.19	1.87	1.60	1.35			
1.17	6.03	5.25	4.55	3.94	3.40	2.93	2.52	2.16	1.84	1.57	1.33			
1.18	5.93	5.16	4.47	3.87	3.34	2.88	2.48	2.12	1.81	1.54	1.31	1.11		
1.19	5.83	5.07	4.40	3.81	3.29	2.83	2.44	2.09	1.78	1.52	1.29	1.09	0.92	
1.20	5.71	4.97	4.31	3.73	3.22	2.77	2.39	2.04	1.75	1.49	1.26	1.07	0.91	0.76
1.21	5.59	4.87	4.22	3.65	3.15	2.72	2.34	2.00	1.71	1.46	1.24	1.05	0.89	0.75
1.22	5.47	4.76	4.12	3.57	3.08	2.65	2.28	1.95	1.67	1.42	1.21	1.03	0.87	0.73
1.23	5.30	4.61	4.00	3.46	2.99	2.57	2.21	1.90	1.62	1.38	1.17	1.00	0.85	0.71
1.24	5.15	4.48	3.88	3.36	2.90	2.50	2.15	1.84	1.57	1.34	1.14	0.97	0.82	0.69
1.25	4.98	4.33	3.75	3.25	2.80	2.42	2.08	1.78	1.52	1.29	1.10	0.93	0.79	0.66
1.26	4.77	4.14	3.59	3.11	2.68	2.31	1.99	1.70	1.46	1.24	1.05	0.89	0.75	0.63

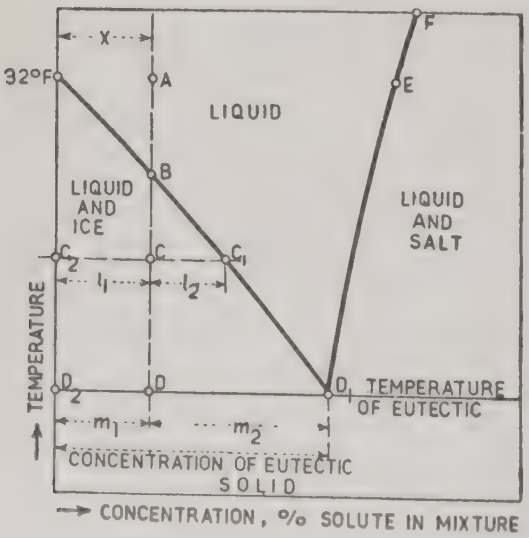

Fig. 1. Brine Mixture

point D, there is a mixture of m_1 parts of eutectic brine solution D_1, and m_2 parts of ice, D_2, all at the eutectic temperature. As more heat is removed, the m_1 parts of eutectic brine freeze at uniform temperature, until all of the latent heat is removed. The frozen eutectic mixture is a mechanical mixture of salt and frozen water, not a solution, and consequently the latent heat must be corrected for the heat of solution. If this is positive it decreases the effective latent heat, if negative it increases it.

If the initial solution is of greater concentration than the eutectic, salt freezes out as the temperature is lowered, instead of water, and the concentration is decreased until at the eutectic temperature the eutectic concentration is reached. In

brines used as a refrigerating fluid, salt sometimes freezes out due to the use of too high concentrations.

The upper right branch of the curve in Fig. 1 gives the solubility—point *E* at 32 F, point *F* at some higher temperature.

On melting a eutectic brine, the water must redissolve the salt, which may require agitation, circulation or other mixing aids.

The total heat of $CaCl_2$ solutions at various temperatures and concentrations is shown in Table 2. The −59.8 horizontal line represents the eutectic temperature, and the lower horizontal line in each column represents the temperature at which ice crystals start to freeze out. In Table 7 similar data are given for NaCl. **Table 12** gives the eutectic concentrations and temperatures for a number of two-component systems. The data are adapted from the International Critical Tables (which contain data on many additional systems, including some three-component systems).

Table 13 shows the concentration and freezing point of some aqueous solutions (non-eutectic). Other non-eutectic solutions can be found in the International Critical Tables.

Figs. 2 and **3** show the variations in specific heat of calcium chloride and sodium chloride brine solutions with changes in concentration and temperature.

The **applications** of brine in refrigeration are mainly in the industrial machinery field where the calcium chloride solution is the standard secondary refrigerating medium. Corrosion is the principal problem met in use of calcium chloride brines, es-

Table 6. Properties of Pure Sodium Chloride (NaCl) Brine

Pure NaCl % by wt	Specific gravity 59 F 39 F	Beaumé density 60 F	Salometer 59 F 39 F	Specific heat 59 F Btu/lb deg F	Freezing point deg F	Weight per gallon*			Weight per cubic foot*		
						NaCl lb/gal	Water lb/gal	Brine lb/gal	NaCl lb/cu ft	Water lb/cu ft	Brine lb/cu ft
0	1.000	0.0	0.0	1.000	32.0	0.000	8.34	8.34	0.000	62.40	62.4
5	1.035	5.1	18.2	0.938	27.0	0.432	8.22	8.65	3.230	61.37	64.6
6	1.043	6.1	22.5	0.927	25.5	0.523	8.19	8.71	3.906	61.19	65.1
7	1.050	7.0	26.0	0.917	24.0	0.613	8.15	8.76	4.585	60.91	65.5
8	1.057	8.0	29.6	0.907	23.2	0.706	8.11	8.82	5.280	60.72	66.0
9	1.065	9.0	33.5	0.897	21.8	0.800	8.09	8.89	5.985	60.51	66.5
10	1.072	10.1	37.2	0.888	20.4	0.895	8.05	8.95	6.690	60.21	66.9
11	1.080	10.8	41.1	0.879	18.5	0.992	8.03	9.02	7.414	59.99	67.4
12	1.087	11.8	44.8	0.870	17.2	1.090	7.99	9.08	8.136	59.66	67.8
13	1.095	12.7	48.7	0.862	15.5	1.188	7.95	9.14	8.879	59.32	68.3
14	1.103	13.6	52.6	0.854	13.9	1.291	7.93	9.22	9.632	59.17	68.8
15	1.111	14.5	56.8	0.847	12.0	1.392	7.89	9.28	10.395	58.90	69.3
16	1.118	15.4	60.0	0.840	10.2	1.493	7.84	9.33	11.168	58.63	69.8
17	1.126	16.3	64.0	0.833	8.2	1.598	7.80	9.40	11.951	58.36	70.3
18	1.134	17.2	68.0	0.826	6.1	1.705	7.76	9.47	12.744	58.06	70.8
19	1.142	18.1	71.7	0.819	4.0	1.813	7.73	9.54	13.547	57.75	71.3
20	1.150	19.0	75.2	0.813	+ 1.8	1.920	7.68	9.60	14.360	57.44	71.8
21	1.158	19.9	79.1	0.807	− 0.8	2.031	7.64	9.67	15.183	57.12	72.3
22	1.166	20.8	82.8	0.802	− 3.0	2.143	7.60	9.74	16.016	56.78	72.8
23	1.175	21.7	86.8	0.796	− 6.0	2.256	7.55	9.81	16.854	56.45	73.3
24	1.183	22.5	90.2	0.791	+ 3.8	2.371	7.51	9.88	17.712	56.09	73.8
25	1.191	23.4	94.0	0.786	+16.1	2.488	7.46	9.95	18.575	55.72	74.3
25.2	1.200				+32.0						

* Wt of commercial NaCl required $= \frac{\text{wt of pure NaCl required}}{\text{per cent purity}}$.

Table 7. Enthalpy (NaCl)

Enthalpy in Btu per lb from 32 F, where it is 200 minus heat of solution[1]

Sp gr		1.04	1.06	1.08	1.10	1.12	1.14	1.16	1.18
Lb NaCl per 100 lb solution		5.5	8.2	10.9	13.5	16.1	18.6	21.1	23.5
Freezing point, deg F		26.0	12.7	19.0	14.9	10.4	5.4	−0.3	−3.6
Lb NaCl per gal solution, 60 F		0.48	0.73	0.98	1.24	1.50	1.77	2.04	2.31
Lb solution per gal, 60 F		8.67	8.84	9.00	9.17	9.34	9.50	9.67	9.84
Deg F									
−6.03	Solid	46.46	50.34	54.35	58.29	62.17	65.96	69.98	72.93
−6.03		70.26	86.12	101.77	117.16	132.29	147.11	161.65	174.35
−4		72.42	88.88	105.12	121.09	136.79	152.17	167.26	176.03
−2		74.65	91.74	108.60	125.19	141.48	157.45	173.13	177.59
0		77.05	94.86	112.43	129.71	146.70	163.34	178.98	179.16
2		79.66	98.28	116.67	134.74	152.49	169.90	180.56	180.73
4		82.50	102.06	121.35	140.33	158.98	177.25	182.15	182.29
6		85.61	106.22	126.57	146.57	166.23	183.47	183.74	183.86
8		89.09	110.95	132.50	153.73	174.56	185.09	185.33	185.43
10		93.10	116.45	139.48	162.16	184.41	186.71	186.92	187.00
12		97.72	122.90	147.70	172.13	188.00	188.33	188.51	188.58
14	Liquid and Solid	103.14	130.47	157.44	183.96	189.66	189.96	190.11	190.15
16		109.60	139.62	169.24	190.85	191.31	191.58	191.71	191.72
18		117.62	151.09	184.13	192.54	192.57	193.21	193.30	193.30
20		127.97	166.04	193.64	194.24	194.63	194.83	194.90	194.88
22		141.96	186.43	195.39	195.94	196.29	196.46	196.50	196.45
24		162.72	196.38	197.14	197.63	197.94	198.09	198.10	198.03
26		196.19	198.18	198.88	199.33	199.61	199.72	199.70	199.61
28		199.00	199.98	200.63	201.03	201.27	201.34	201.30	201.19
30		200.85	201.77	202.38	202.74	202.94	202.98	202.91	202.77
32		202.71	203.57	204.12	204.44	204.60	204.61	204.51	204.35
34		204.57	205.37	205.87	206.14	206.26	206.24	206.11	205.93
36		206.43	207.17	207.62	207.85	207.93	207.88	207.72	207.51
38		208.29	208.97	209.37	209.56	209.60	209.51	209.33	209.10
40	Liquid	210.15	210.78	211.12	211.26	211.27	211.15	210.94	210.68
42		212.01	212.57	212.87	212.97	212.94	212.79	212.54	212.27
44		213.87	214.38	214.62	214.68	214.61	214.43	214.16	213.86
46		215.73	216.18	216.38	216.39	216.29	216.07	215.77	215.44
48		217.59	217.99	218.14	218.11	217.96	217.71	217.38	217.03
50		219.46	219.80	219.89	219.82	219.63	219.34	218.99	218.62

pecially in ice-making tanks where galvanzed iron cans are immersed.

Calcium chloride is also used in machinery of intermediate sizes to provide holdover capacity in refrigeration storage tanks or plates. Corrosion treatment in this class of work is not readily applied by the user, but manufacturers may supply the treatment or service.

Ordinary salt is used where contact with calcium chloride could not be tolerated, as for example in the brine fog method of freezing fish and other foods. A common application now is in air cooling in unit coolers, as spray, preventing formation of frost on coils. For most refrigerating work the lower freezing point of calcium makes its use more convenient.

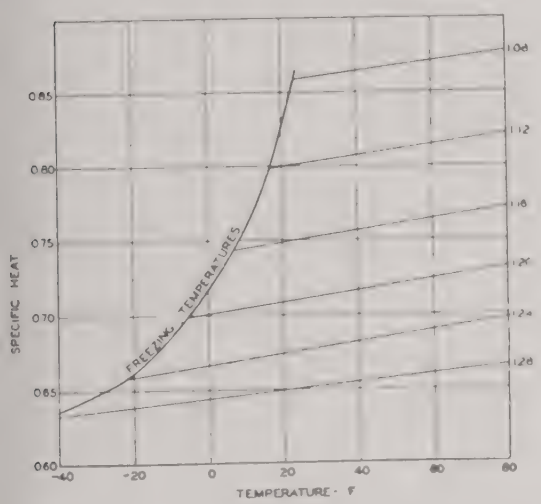

Fig. 2. Specific Heat of Calcium Chloride

Eutectic brine in metal containers is often used in place of dry ice for ice-cream cabinets and truck refrigeration. The eutectic brine, in metal containers, is frozen in cold rooms or by means of refrigeration coils built in the eutectic container.

Propylene glycol solution finds some use as a brine, principally in the brewing and ice cream industries. It is colorless, prac-

Table 8. Sodium Chloride Density

Specific gravity refers to water at 60 F

Sp gr or	Densities (gr per cc)			
%	14 F	32 F	50 F	68 F
5		1.0382	1.0366	1.0341
6		1.0459	1.0440	1.0413
7		1.0536	1.0515	1.0486
8		1.0613	1.0590	1.0559
9		1.0691	1.0665	1.0633
10		1.0769	1.0741	1.0707
11		1.0849	1.0817	1.0782
12		1.0925	1.0897	1.0857
13		1.1004	1.0971	1.0933
14		1.1083	1.1048	1.1009
15	1.1195	1.1163	1.1126	1.1086
16	1.1277	1.1243	1.1205	1.1163
17	1.1359	1.1323	1.1284	1.1241
18	1.1442	1.1404	1.1363	1.1319
19	1.1535	1.1486	1.1444	1.1398
20	1.1608	1.1568	1.1524	1.1478
21	1.1692	1.1651	1.1606	1.1559
22	1.1777	1.1734	1.1688	1.1640
23	1.1862	1.1818	1.1771	1.1721
24	1.1948	1.1902	1.1854	1.1804

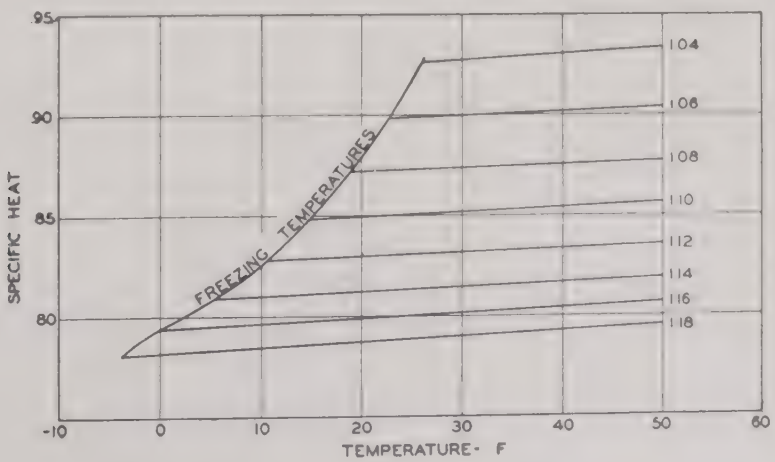

Fig. 3. Specific Heat of Sodium Chloride

Table 9. Sodium Chloride Viscosity

	Viscosities (centipoises)[5]				
%	5 F	14 F	23 F	32 F	55 F
4.9				1.824	1.368
5.9				1.849	1.388
7.0				1.878	1.412
8.0			2.297	1.912	1.437
9.1			2.349	1.952	1.466
10.2			2.403	1.991	1.496
11.3			2.464	2.040	1.530
12.3			2.525	2.091	1.564
13.4		3.212	2.499	2.153	1.606
14.5		3.320	2.682	2.224	1.649
15.6		3.437	2.768	2.308	1.701
16.7		3.575	2.874	2.391	1.755
17.9	4.629	3.741	3.011	2.490	1.813
19.0	4.830	3.908	3.148	2.592	1.872
20.1	5.056	4.009	3.295	2.702	1.941
21.2	5.291	4.310	3.454	2.820	2.004
22.3	5.556	4.521	3.624	2.947	2.098
23.5	5.865	4.791	3.817	3.091	2.199
24.7		5.105	4.026	3.246	2.316
25.9				3.433	2.463

tically odorless, non-corrosive chemically and inhibits bacterial growth. **Table 14** gives the properties of pure propylene glycol. In solution the specific gravity varies little, being 1.020 at 20% glycol and 1.055 at 60% glycol, both at 23 F.

For information on the application of these brines in respect to brine coolers, refer to Chap. 25.

References

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4. *Ibid.*, values selected from tables on p. 1066.
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7. Stakelbeck and Plank, *Zeitschrift f.d.g. Kalte, Ind.*, June and July 1929.
8. Smithsonian Tables, 6th ed.
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11. *Brewer's Digest*, Oct. 1938.

Table 10. Sodium Chloride Vapor Pressure[6]

Vapor pressure above solutions in mm or mercury[6]

Sp gr at 60 F	Temperature, deg F													
	42.8	39.2	35.6	32.0	28.4	24.8	21.2	17.6	14.0	10.4	6.8	3.2	−0.4	−4.0
1.00	7.01	6.10	5.29	4.58	3.88	3.28	2.76	2.32	1.95	1.63	1.36	1.13	0.94	0.77
1.01	6.94	6.04	5.24	4.54										
1.02	6.88	5.99	5.20	4.50										
1.03	6.82	5.94	5.15	4.46	3.84									
1.04	6.76	5.89	5.10	4.42	3.81									
1.05	6.69	5.82	5.05	4.37	3.77	3.25								
1.06	6.62	5.76	5.00	4.33	3.74	3.22								
1.07	6.56	5.70	4.95	4.28	3.70	3.18								
1.08	6.48	5.64	4.89	4.23	3.65	3.15	2.71							
1.09	6.41	5.58	4.83	4.19	3.61	3.11	2.68	2.29						
1.10	6.34	5.52	4.78	4.14	3.57	3.08	2.65	2.27	1.94					
1.11	6.26	5.45	4.72	4.09	3.54	3.04	2.61	2.24	1.91					
1.12	6.17	5.37	4.66	4.03	3.48	3.00	2.58	2.21	1.89	1.61				
1.13	6.09	5.30	4.60	3.98	3.48	2.96	2.54	2.18	1.86	1.59	1.35			
1.14	6.00	5.22	4.53	3.92	3.38	2.91	2.51	2.15	1.83	1.56	1.33	1.13		
1.15	5.91	5.14	4.46	3.86	3.33	2.78	2.47	2.11	1.81	1.54	1.31	1.11		
1.16	5.82	5.06	4.39	3.80	3.28	2.82	2.43	2.08	1.77	1.52	1.29	1.09	0.92	
1.17	5.17	4.97	4.30	3.73	3.22	2.77	2.38	2.04	1.74	1.49	1.26	1.07	0.90	0.76

Table 11. Endothermal Mixtures[9]

Mixtures		Temperature drop, deg F
Ammonium nitrate	1	From +40° to + 4°
Water	1	
Ammonium chloride	5	From +50° to +10°
Potassium nitrate	5	
Water	16	
Ammonium chloride	5	From +50° to + 4°
Potassium nitrate	5	
Sodium sulfate	8	
Water	16	
Sodium nitrate	3	From +50° to − 3°
Nitric acid, diluted	2	
Ammonium nitrate	1	From +50° to − 7°
Sodium carbonate	1	
Water	1	
Sodium phosphate	9	From +50° to −12°
Nitric acid, diluted	4	
Sodium sulfate	5	From +50° to + 3°
Sulfuric acid, diluted	4	
Sodium sulfate	6	From +50° to −10°
Ammonium chloride	4	
Potassium nitrate	2	
Nitric acid, diluted	4	
Sodium sulfate	6	From +50° to −40°
Ammonium nitrate	5	
Nitric acid, diluted	4	
Snow or pounded ice	2	From +50° to − 5°
Sodium chloride	1	
Snow or pounded ice	5	From +50° to −12°
Sodium chloride	2	
Ammonium chloride	1	
Snow or pounded ice	24	From +50° to −18°
Sodium chloride	10	
Ammonium chloride	5	
Potassium nitrate	5	
Snow or pounded ice	12	From +50° to −25°
Sodium chloride	5	
Ammonium nitrate	5	
Snow	3	From +32° to −23°
Sulfuric acid, diluted	2	
Snow	8	From +32° to −27°
Hydrochloric acid	5	
Snow	7	From +32° to −30°
Nitric acid, diluted	4	
Snow	4	From +32° to −40°
Calcium chloride	5	
Snow	2	From +32° to −50°
Calcium chloride, crystallized	3	
Snow	3	From +32° to −51°
Potash	4	

Table 12. Eutectic Points of Aqueous Solutions[5]

Substance	Eutectic concentration: wt of solute ÷ wt of solution	Eutectic temperature, deg F
Ammonium nitrate (NH_4NO_2)	.428	+ 2.0
Ammonium chloride (NH_4Cl)	.191	+ 4.3
Calcium chloride ($CaCl_2$)	.324	− 59.8
Sodium chloride (NaCl)	.233	− 6.0
Sodium sulfate (Na_2SO_4)	.049	+ 30.0
Sodium nitrate ($NaNO_3$)	.397	+ 0.6
Sodium carbonate (Na_2CO_3)	.080	+ 28.2
Potassium chloride (KCl)	.197	+ 12.8
Potassium sulfate (K_2SO_4)	.065	+ 28.3
Potassium nitrate (KNO_3)	.112	+ 26.6
Potassium carbonate (KCO_3)	.355	− 34.7
Methyl alcohol (CH_4O)	.695	−164
Ethyl alcohol (C_2H_6O)	.763	− 94
Glycerine ($C_3H_5O_3$)	.669	− 60.4

Table 13. See page 210.

Table 14. Properties of Pure Propylene Glycol

Molecular weight	76.06
Boiling point, deg F	369.1
Specific gravity at 68 F	1.0381
Weight per gal at 68 F, lb	8.64
Surface tension at 77 F, dynes per cm	40.1
Viscosity at 77 F, poises	0.406
Specific heat at 68 F, Btu per lb F	0.590
Latent heat of evaporation at boiling point, Btu per lb	303.5
Flash Point (A.S.T.M. Open Cup), deg F	210.2
Coefficient of expansion at 68 F (per deg F)	0.000386

Table 13. Freezing Points of Aqueous Solutions

Alcohol		Glycerine		Ethylene Glycol		Propylene Glycol	
% by wt	deg F	% by wt	deg F	% by vol	deg F	% by vol	deg F
5	28.0	10	29.1	15	22.4	5	29.0
10	23.6	20	23.4	20	16.2	10	26.0
15	19.7	30	14.9	25	10.0	15	22.5
20	13.2	40	4.3	30	3.5	20	19.0
25	5.5	50	− 9.4	35	− 4.0	25	14.5
30	− 2.5	60	−30.5	40	−12.5	30	9.0
35	−13.2	70	−38.0	45	−22.0	35	2.5
40	−21.0	80	− 5.5	50	−32.5	40	− 5.5
45	−27.5	90	+29.1			45	−15.0
50	−34.0	100	+62.6			50	−25.5
55	−40.5					55	−39.5
						59	−57.0
						Above 60% fails to crystallize at −99.4 F	

10. PSYCHROMETRIC TABLES

TABLES presented here include data on the properties of dry air, saturated ice, saturated water, saturated steam, and mixtures of dry air and saturated steam in the range of temperatures from −160 F to 200 F.

Although properties of dry air and of the mixture of dry air and saturated steam are given, directly, only for a total pressure of 29.921 in. Hg abs, sufficient information is given so that these properties may be found for any other total pressure. **Table 3** shows for the range in question all of the items in the following development. **Tables 1** and **2** are arranged for general use with certain of the developmental items omitted.

There follows a summary of data used in preparation of the tables:

Assumed composition of dry air:

Constituent	*Fraction by volume*	*Fraction of weight*
Nitrogen	0.7803	0.7547
Oxygen	0.2099	0.2319
Monatomic gases (principally argon)	0.0094	0.0130
Carbon dioxide	0.0003	0.0004
Hydrogen	0.0001	0.0000

Based upon this analysis, the equivalent molecular weight of dry air is 28.966 lb per mol.

Equation of state for dry air is the Beattie-Bridgeman form with all virial coefficients beyond the second ignored.

$$v_a = v_1 - v_2$$

$$v_1 = \frac{53.35(t+459.7)}{p}$$

$$v_2 = -0.0225 + \frac{13.06}{t+459.7} + \frac{3.87(10)^5}{(t+459.7)^3}.$$

Specific heat at constant pressure of dry air. The values of the molal specific heat at zero pressure of nitrogen and oxygen as calculated from quantum theory and spectroscopic data by Johnston and Davis and Johnston and Walker were used to derive an empirical equation for the specific heat of dry air at zero pressure as a function of temperature. (For the monatomic gases present in dry air, a constant value of the specific heat at zero pressure of 0.124 Btu/lb F was used.)

For the range in temperature covered by the tables, the specific heat at zero pressure of dry air is:

$$c_p^{(0)} = 0.2397 + 4.08(10)^{-6}t + 9.24(10)^{-9}t^2$$

where t is in degrees F.

The enthalpy of dry air was calculated from the following equation:

$$h_a = h_1 - h_2,$$

where

$$h_1 = \int_o c_p^{(0)} dt - h_{ao},$$

$$h_2 = \frac{B_a p_a}{778.18},$$

$$B_a = \frac{d\left(\frac{v_2}{T}\right)}{d\left(\frac{1}{T}\right)} = -.0225 + \frac{26.12}{(t+459.7)} + \frac{15.48(10)^5}{(t+459.7)^3}$$

The specific enthalpy of dry air was assigned, arbitrarily, a zero value at $t = 0$ F and at a dry air pressure of 29.921 in. Hg abs, so $h_{ao} = -0.1366$ Btu/lb.

Properties of saturated ice, saturated water, and saturated steam. The thermodynamic properties of ice, water, and steam were taken from the tables prepared by Goff and Gratch presented in *Heating, Piping and Air Conditioning*, February, 1946, and also in the 1946 A.S.H.V.E. Guide.

Properties of mixtures of dry air and saturated steam. In preparing these tables, the Gibbs-Dalton Law was assumed to

apply without any correction for interaction effects.

Properties of dry air at a pressure of P in. Hg abs:

$$v_a = \frac{29.921 v_1}{P} - v_2$$

$$h_a = h_1 - \frac{P h_2}{29.921}$$

Example: Find the specific volume and specific enthalpy of dry air at a pressure of 28.0 in. Hg abs and a temperature of 70 F.

$$v_a = \frac{29.921(13.354)}{28.0} - 0.005$$

$$= 14.265 \text{ cu ft per lb}$$

$$h_a = 16.93 - \frac{28(0.10)}{29.921} = 16.84 \text{ Btu/lb}$$

Properties of mixtures of dry air and saturated steam at a total pressure of P in. Hg abs:

$$v_{s1} = \frac{29.921 v_1}{P - p_s} - v_2,$$

$$W_{s1} = \frac{7{,}000 v_{s1}}{v_g};$$

$$h_{s1} = h_1 - \frac{(P - p_s) h_2}{29.921} + \frac{W_{s1} h_g}{7{,}000}$$

Example: Find the volume per lb of dry air, the specific humidity, and the enthalpy per lb of dry air of a mixture of dry air and saturated steam at a total pressure of 28.0 in. Hg abs and a temperature of 70 F.

Volume:

$$v_{s1} = \frac{29.921(13.354)}{(28.0 - 0.7392)} - 0.005$$

$$= 14.652 \text{ cu ft}$$

Specific humidity:

$$W_{s1} = \frac{7{,}000(14.652)}{868} = 118.2 \text{ gr}$$

Enthalpy:

$$h_{s1} = 16.93 - \frac{(28.0 - 0.7392)(0.10)}{29.921} + \frac{(118.2)(1091.8)}{7{,}000}$$

$$= 35.27 \text{ Btu}$$

Note that Table 3 was used in finding values of v_2 and h_2 in these examples. For temperatures not given in Table 3, Table 1 can be used, although it will be necessary to determine the approximate magnitudes of v_2 and h_2 from Table 3. In most cases, however, the corrective terms v_2 and h_2 can be disregarded and values taken from Table 1 can have pressure corrections applied directly. This is allowable as in most cases, particularly with enthalpy, the interest is in enthalpy change.

Table 1. Thermodynamic Properties of Air, Water and Steam

Temperature, F	Properties of Water and Steam				Properties of Dry Air at a Pressure of 29.921 in. Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Saturation pressure of water & steam, in. Hg	Enthalpy of sat water, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, ft³	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_f	h_g	v_g	v_a	h_a	v_s	h_s	W_s
32	0.1803	0.0	1075.2	3305	12.389	7.69	12.46	11.75	26.40
33	0.1878	1.0	1075.6	3180	12.414	7.93	12.49	12.16	27.49
34	0.1955	2.0	1076.0	3062	12.439	8.17	12.52	12.57	28.63
35	0.2034	3.0	1076.5	2948	12.464	8.41	12.55	13.00	29.80
36	0.2117	4.0	1076.9	2839	12.490	8.65	12.58	13.42	31.02
37	0.2202	5.0	1077.4	2734	12.515	8.89	12.61	13.86	32.28
38	0.2290	6.0	1077.8	2634	12.540	9.13	12.64	14.30	33.58
39	0.2382	7.0	1078.2	2538	12.565	9.37	12.67	14.76	34.94
40	0.2477	8.0	1078.7	2445	12.591	9.61	12.70	15.21	36.34
41	0.2575	9.0	1079.1	2357	12.616	9.85	12.73	15.68	37.80
42	0.2676	10.1	1079.5	2272	12.641	10.09	12.76	16.16	39.30
43	0.2781	11.1	1080.0	2190	12.667	10.34	12.79	16.64	40.86
44	0.2890	12.1	1080.4	2112	12.692	10.58	12.82	17.13	42.47
45	0.3002	13.1	1080.9	2037	12.717	10.82	12.85	17.63	44.14
46	0.3119	14.1	1081.3	1965	12.742	11.06	12.88	18.13	45.86
47	0.3239	15.1	1081.7	1896	12.768	11.30	12.91	18.66	47.65
48	0.3363	16.1	1082.2	1829	12.793	11.54	12.94	19.19	49.51
49	0.3491	17.1	1082.6	1766	12.818	11.78	12.97	19.73	51.42
50	0.3624	18.1	1083.1	1704	12.844	12.02	13.00	20.28	53.40
51	0.3761	19.1	1083.5	1645	12.869	12.26	13.03	20.84	55.44
52	0.3903	20.1	1083.9	1589	12.894	12.50	13.06	21.41	57.56
53	0.4049	21.1	1084.4	1534	12.919	12.74	13.10	21.99	59.75
54	0.4200	22.1	1084.8	1482	12.945	12.98	13.13	22.59	62.01
55	0.4356	23.1	1085.2	1431	12.970	13.22	13.16	23.20	64.36
56	0.4518	24.1	1085.7	1383	12.995	13.46	13.19	23.82	66.78
57	0.4684	25.1	1086.1	1336	13.020	13.70	13.23	24.45	69.28
58	0.4856	26.1	1086.5	1292	13.046	13.94	13.26	25.10	71.86
59	0.5033	27.1	1087.0	1249	13.071	14.18	13.29	25.76	74.54
60	0.5216	28.1	1087.4	1207	13.096	14.42	13.33	26.43	77.29
61	0.5405	29.1	1087.9	1167	13.122	14.66	13.36	27.11	80.14
62	0.5599	30.1	1088.3	1129	13.147	14.90	13.40	27.82	83.09
63	0.5800	31.1	1088.7	1092	13.172	15.14	13.43	28.54	86.14
64	0.6007	32.1	1089.2	1056	13.197	15.38	13.47	29.27	89.27
65	0.6221	33.1	1089.6	1022	13.223	15.62	13.50	30.03	92.51
66	0.6441	34.1	1090.0	988.6	13.248	15.86	13.54	30.79	95.86
67	0.6668	35.1	1090.5	956.8	13.273	16.10	13.58	31.58	99.32
68	0.6902	36.1	1090.9	926.1	13.298	16.35	13.61	32.38	102.9
69	0.7143	37.1	1091.3	896.5	13.324	16.59	13.65	33.20	106.6
70	0.7392	38.1	1091.8	868.0	13.349	16.83	13.69	34.04	110.4
71	0.7648	39.1	1092.2	840.5	13.374	17.07	13.72	34.90	114.3
72	0.7911	40.1	1092.6	814.0	13.399	17.31	13.76	35.79	118.4
73	0.8183	41.1	1093.1	788.4	13.425	17.55	13.80	36.69	122.6
74	0.8463	42.1	1093.5	763.8	13.450	17.79	13.84	37.61	126.9
75	0.8751	43.1	1093.9	740.0	13.475	18.03	13.88	38.55	131.3
76	0.9047	44.1	1094.4	717.0	13.501	18.27	13.92	39.52	135.9
77	0.9352	45.1	1094.8	694.9	13.526	18.51	13.96	40.51	140.6
78	0.9667	46.1	1095.2	673.5	13.551	18.75	14.00	41.52	145.5
79	0.9990	47.1	1095.7	652.9	13.576	18.99	14.04	42.56	150.6
80	1.0323	48.1	1096.1	633.0	13.602	19.23	14.09	43.63	155.8
81	1.0665	49.1	1096.6	613.8	13.627	19.47	14.13	44.72	161.2
82	1.1017	50.1	1097.0	595.3	13.652	19.71	14.17	45.84	166.7
83	1.1380	51.1	1097.4	577.4	13.678	19.95	14.22	46.98	172.4
84	1.1752	52.1	1097.8	560.1	13.703	20.19	14.26	48.16	178.3
85	1.2136	53.1	1098.3	543.3	13.738	20.43	14.31	49.36	184.4
86	1.2530	54.0	1098.7	527.2	13.753	20.67	14.35	50.59	190.6
87	1.2935	55.0	1099.1	511.6	13.778	20.91	14.40	51.86	197.0
88	1.3351	56.0	1099.6	496.5	13.804	21.15	14.45	53.14	203.7
89	1.3779	57.0	1100.0	482.0	13.829	21.39	14.50	54.48	210.6
90	1.4219	58.0	1100.4	467.9	13.854	21.64	14.55	55.85	217.6

Table 1. Thermodynamic Properties of Air, Water and Steam (Continued)

Temperature, F	Properties of Water and Steam				Properties of Dry Air at a Pressure of 29.921 in. Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Saturation pressure of water & steam, in. Hg	Enthalpy of sat water, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, ft³	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_f	h_g	v_g	v_a	h_a	v_s	h_s	W_s
91	1.4671	59.0	1100.9	454.3	13.880	21.88	14.60	57.25	224.9
92	1.5136	60.0	1101.3	441.1	13.905	22.12	14.65	58.69	232.4
93	1.5613	61.0	1101.7	428.4	13.930	22.36	14.70	60.16	240.1
94	1.6103	62.0	1102.2	416.1	13.955	22.60	13.75	61.67	248.1
95	1.6607	63.0	1102.6	404.2	13.981	22.84	14.80	63.12	256.4
96	1.7124	64.0	1103.0	392.7	14.006	23.08	14.86	64.81	264.8
97	1.7655	65.0	1103.4	381.5	14.031	23.32	14.91	66.20	273.6
98	1.8200	66.0	1103.9	370.7	14.057	23.56	14.97	68.53	282.6
99	1.8759	67.0	1104.3	360.3	14.082	23.80	15.02	70.03	291.9
100	1.9334	68.0	1104.7	350.2	14.107	24.04	15.08	71.62	301.5
101	1.9923	69.0	1105.2	340.4	14.132	24.28	15.14	73.44	311.3
102	2.0529	70.0	1105.6	331.0	14.157	24.52	15.20	75.31	321.5
103	2.1149	71.0	1106.0	321.8	14.183	24.76	15.26	77.22	332.0
104	2.1786	72.0	1106.4	313.0	14.208	25.00	15.32	79.19	342.8
105	2.2440	73.0	1106.9	304.4	14.233	25.24	15.39	81.21	353.9
106	2.3110	74.0	1107.3	296.0	14.259	25.48	15.45	83.29	365.4
107	2.3798	75.0	1107.7	288.0	14.284	25.72	15.52	85.42	377.2
108	2.4503	76.0	1108.2	280.2	14.309	25.96	15.59	87.62	389.4
109	2.5226	77.0	1108.6	272.6	14.334	26.20	15.65	89.87	402.0
110	2.5968	78.0	1109.0	265.3	14.360	26.45	15.72	92.19	414.9
111	2.6728	79.0	1109.4	258.2	14.385	26.69	15.80	94.58	428.3
112	2.7507	80.0	1109.9	251.3	14.410	26.93	15.87	97.03	442.1
113	2.8306	81.0	1110.3	244.6	14.435	27.17	15.94	99.55	456.3
114	2.9125	82.0	1110.7	238.1	14.461	27.41	16.02	102.16	471.0
115	2.9963	83.0	1111.1	231.8	14.486	27.65	16.10	104.81	486.1
116	3.0823	84.0	1111.6	225.8	14.511	27.89	16.18	107.55	501.6
117	3.1703	85.0	1112.0	219.9	14.537	28.13	16.26	110.38	517.7
118	3.2606	86.0	1112.4	214.1	14.562	28.37	16.34	113.29	534.3
119	3.3530	87.0	1112.8	208.6	14.587	28.61	16.43	116.28	551.4
120	3.4477	88.0	1113.3	203.2	14.612	28.85	16.51	119.36	569.0
121	3.5446	89.0	1113.7	197.9	14.637	29.09	16.60	122.52	587.2
122	3.6439	90.0	1114.1	192.9	14.663	29.33	16.70	125.79	606.0
123	3.7455	91.0	1114.5	188.0	14.688	29.57	16.79	129.15	625.3
124	3.8496	92.0	1114.9	183.2	14.713	29.82	16.89	132.61	645.3
125	3.9561	93.0	1115.4	178.5	14.739	30.06	16.98	136.17	665.9
126	4.0651	94.0	1115.8	174.0	14.764	30.30	17.08	139.88	687.2
127	4.1768	95.0	1116.2	169.6	14.789	30.54	17.19	143.64	709.2
128	4.2910	96.0	1116.6	165.4	14.814	30.78	17.29	147.54	731.9
129	4.4078	97.0	1117.0	161.3	14.839	31.02	17.40	151.57	755.4
130	4.5274	98.0	1117.5	157.3	14.865	31.26	17.52	155.72	779.6
131	4.6498	99.0	1117.9	153.4	14.890	31.50	17.63	160.00	804.6
132	4.7750	100.0	1118.3	149.6	14.915	31.74	17.75	164.43	830.5
133	4.9030	101.0	1118.7	145.9	14.941	31.98	17.87	168.98	857.2
134	5.0340	102.0	1119.2	142.4	14.966	32.22	17.99	173.69	884.8
135	5.1679	103.0	1119.6	138.9	14.991	32.46	18.12	178.54	913.3
136	5.3049	104.0	1120.0	135.5	15.016	32.70	18.25	183.57	942.8
137	5.4450	105.0	1120.4	132.2	15.043	32.94	18.39	188.75	973.4
138	5.5881	106.0	1120.8	129.1	15.067	33.18	18.53	194.09	1005.0
139	5.7345	107.0	1121.2	126.0	15.092	33.43	18.67	199.64	1038
140	5.8842	108.0	1121.7	123.0	15.117	33.67	18.82	205.34	1071
141	6.0371	109.0	1122.1	120.0	15.143	33.91	18.97	211.27	1106
142	6.1934	110.0	1122.5	117.2	15.168	34.15	19.13	217.39	1143
143	6.3532	111.0	1122.9	114.4	15.193	34.39	19.29	223.70	1180
144	6.5164	112.0	1123.3	111.7	15.218	34.63	19.45	230.28	1219
145	6.6832	113.0	1123.7	109.1	15.244	34.87	19.62	236.94	1259
146	6.8536	114.0	1124.1	106.6	15.269	35.11	19.81	244.06	1301
147	7.0277	115.0	1124.6	104.1	15.294	35.35	19.99	251.34	1344
148	7.2056	116.0	1125.0	101.7	15.319	35.59	20.18	258.88	1389
149	7.3872	117.0	1125.4	99.32	15.345	35.83	20.37	266.71	1436
150	7.5727	118.0	1125.8	97.04	15.370	36.07	20.58	274.84	1485

Table 1. Thermodynamic Properties of Air, Water and Steam (Concluded)

Temperature, F	Properties of Water and Steam				Properties of Dry Air at a Pressure of 29.921 in. Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Saturation pressure of water & steam, in. Hg	Enthalpy of sat water, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, ft³	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_f	h_g	v_g	v_a	h_a	v_g	h_s	W_s
151	7.7622	119.0	1126.2	94.81	15.395	36.31	20.79	283.25	1535
152	7.9556	120.0	1126.6	92.65	15.420	36.56	21.01	292.00	1587
153	8.1532	121.0	1127.0	90.54	15.446	36.80	21.23	301.07	1641
154	8.3548	122.0	1127.4	88.49	15.471	37.04	21.46	310.53	1698
155	8.5607	123.0	1127.8	86.50	15.496	37.28	21.71	320.34	1757
156	8.7708	124.0	1128.3	84.55	15.521	37.52	21.96	330.57	1818
157	8.9853	125.0	1128.7	82.66	15.547	37.76	22.22	341.18	1882
158	9.2042	126.0	1129.1	80.81	15.572	38.00	22.49	352.24	1948
159	9.4276	127.0	1129.5	79.02	15.597	38.24	22.77	363.29	2018
160	9.6556	128.0	1129.9	77.27	15.622	38.48	23.07	375.81	2090
161	9.8882	129.0	1130.3	75.56	15.648	38.72	23.37	388.34	2165
162	10.126	130.0	1130.7	73.90	15.673	38.96	23.69	401.45	2244
163	10.368	131.0	1131.1	72.28	15.698	39.21	24.02	415.14	2326
164	10.615	132.0	1131.5	70.71	15.724	39.45	24.37	429.44	2412
165	10.867	133.0	1131.9	69.17	15.849	39.69	24.73	444.41	2503
166	11.124	134.0	1132.3	67.67	15.774	39.93	25.11	460.10	2597
167	11.386	135.0	1132.7	66.21	15.799	40.17	25.50	476.53	2696
168	11.653	136.0	1133.1	64.79	15.824	40.41	25.92	493.77	2800
169	11.925	137.0	1133.5	63.40	15.850	40.65	26.35	511.83	2910
170	12.203	138.0	1133.9	62.04	15.875	40.89	26.81	530.86	3024
171	12.487	139.0	1134.4	60.73	15.900	41.13	27.29	550.89	3145
172	12.775	140.0	1134.8	59.44	15.925	41.37	27.79	571.97	3273
173	13.080	141.0	1135.2	58.18	15.951	41.61	28.32	594.19	3407
174	13.370	142.0	1135.6	56.96	15.976	41.85	28.88	617.65	3549
175	13.676	143.9	1136.0	55.77	16.001	41.10	29.47	642.48	3699
176	13.987	144.0	1136.4	54.60	16.026	42.34	30.09	668.67	3858
177	14.305	145.0	1136.8	53.47	16.052	42.58	30.75	696.49	4026
178	14.629	146.0	1137.2	52.36	16.077	42.82	31.46	726.04	4205
179	14.959	147.0	1137.6	51.28	16.103	43.06	32.21	757.46	4396
180	15.295	148.0	1137.9	50.22	16.128	43.30	32.99	790.88	4598
181	15.637	149.0	1138.3	49.19	16.153	43.54	33.83	826.46	4815
182	15.986	150.0	1138.7	48.19	16.178	43.78	34.74	864.74	5046
183	16.341	151.0	1139.1	47.20	16.203	44.02	35.70	905.58	5294
184	16.703	152.0	1139.5	46.25	16.229	44.26	36.74	949.49	5560
185	17.071	153.0	1139.9	45.31	16.254	44.51	37.85	996.65	5847
186	17.446	154.0	1140.3	44.40	16.279	44.75	39.04	1047.7	6156
187	17.829	155.0	1140.7	43.51	16.304	44.99	40.34	1102.8	6491
188	18.218	156.0	1141.1	42.64	16.329	45.23	41.75	1162.6	6854
189	18.614	157.0	1141.5	41.79	16.354	45.47	42.28	1227.8	7250
190	19.017	158.0	1141.9	40.96	16.380	45.71	44.94	1298.9	7682
191	19.428	159.1	1142.3	40.14	16.405	45.95	46.78	1376.9	8156
192	19.846	160.1	1142.7	39.35	16.430	46.19	48.79	1463.0	8679
193	20.271	161.1	1143.1	38.58	16.456	46.43	51.02	1558.2	9257
194	20.704	162.1	1143.5	37.82	15.481	46.68	53.50	1664.0	9901
195	21.145	163.1	1143.8	37.09	16.506	46.92	56.27	1782.6	10622
196	21 594	164.1	1144.2	36.36	16.531	47.16	59.40	1916.2	11434
197	22.050	165.1	1144.6	35.66	16.556	47.40	62.93	2067.6	12354
198	22.515	166.1	1145.0	34.97	16.582	47.64	66.99	2240.9	13409
199	22.987	167.1	1145.4	34.30	16.607	47.88	71.66	2440.9	14624
200	23.468	168.1	1145.8	33.64	16.632	48.12	77.14	2675.6	16052

Table 2. Thermodynamic Properties of Air, Ice and Steam

Temperature, F t	Properties of Ice and Steam: Saturation pressure of ice and steam, in. Hg p_s	Enthalpy of sat ice, Btu/lb h_i	Enthalpy of sat steam, Btu/lb h_g	Specific volume of sat steam, cu ft/lb v_g	Properties of Dry Air at a Pressure of 29.921 in. Hg abs: True specific volume, cu ft/lb v_a	Enthalpy, Btu/lb v_s	Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs: Volume of mixture per lb of dry air, cu ft h_a	Enthalpy of mixture per lb of dry air, Btu h_s	Specific humidity, grains per lb of dry air W_s
32	0.1803	−143.4	1075.2	3305	12.389	7.69	12.46	11.75	26.40
31	0.1723	−143.9	1074.7	3453	12.363	7.45	12.43	11.32	25.21
30	0.1645	−144.4	1074.3	3608	12.338	7.21	12.41	10.90	24.07
29	0.1571	−144.9	1073.8	3771	12.313	6.97	12.38	10.49	22.98
28	0.1500	−145.4	1073.4	3943	12.287	6.73	12.35	10.09	21.92
27	0.1431	−145.9	1073.0	4122	12.262	6.49	12.32	9.70	20.92
26	0.1366	−146.4	1072.5	4311	12.237	6.25	12.29	9.31	19.96
25	0.1303	−146.9	1072.1	4509	12.211	6.01	12.27	8.92	19.04
24	0.1243	−147.4	1071.7	4717	12.186	5.77	12.24	8.55	18.16
23	0.1186	−147.9	1071.2	4936	12.161	5.53	12.21	8.18	17.32
22	0.1130	−148.4	1070.8	5166	12.136	5.29	12.18	7.81	16.51
21	0.1078	−148.9	1070.3	5408	12.110	5.05	12.15	7.45	15.73
20	0.1027	−149.4	1069.9	5662	12.085	4.81	12.13	7.10	14.99
19	$9.789\ (10)^{-2}$	−149.8	1069.5	5929	12.060	4.56	12.10	6.75	14.28
18	$9.326\ (10)^{-2}$	−150.3	1069.0	6210	12.035	4.32	12.07	6.40	13.61
17	$8.884\ (10)^{-2}$	−150.8	1068.6	6505	12.009	4.08	12.05	6.06	12.96
16	$8.461\ (10)^{-2}$	−151.3	1068.1	6817	11.984	3.84	12.02	5.73	12.34
15	$8.056\ (10)^{-2}$	−151.8	1067.7	7144	11.959	3.60	11.99	5.40	11.75
14	$7.669\ (10)^{-2}$	−152.3	1067.3	7489	11.933	3.36	11.96	5.07	11.18
13	$7.300\ (10)^{-2}$	−152.8	1066.8	7851	11.918	3.12	11.94	4.75	10.64
12	$6.946\ (10)^{-2}$	−153.3	1066.4	8234	11.883	2.88	11.91	4.43	10.13
11	$6.608\ (10)^{-2}$	−153.7	1065.9	8636	11.857	2.64	11.88	4.11	9.633
10	$6.286\ (10)^{-2}$	−154.2	1065.5	9060	11.832	2.40	11.86	3.80	9.161
9	$5.977\ (10)^{-2}$	−154.7	1065.1	9507	11.807	2.16	11.83	3.49	8.711
8	$5.683\ (10)^{-2}$	−155.2	1064.6	9979	11.782	1.92	11.80	3.18	8.280
7	$5.402\ (10)^{-2}$	−155.7	1064.2	$1.048\ (10)^{4}$	11.756	1.68	11.78	2.88	7.867
6	$5.134\ (10)^{-2}$	−156.1	1063.7	$1.100\ (10)^{4}$	11.731	1.44	11.75	2.58	7.478
5	$4.878\ (10)^{-2}$	−156.6	1063.3	$1.155\ (10)^{4}$	11.706	1.20	11.72	2.28	7.106
4	$4.633\ (10)^{-2}$	−157.1	1062.8	$1.214\ (10)^{4}$	11.680	0.96	11.70	1.98	6.745
3	$4.400\ (10)^{-2}$	−157.6	1062.4	$1.275\ (10)^{4}$	11.655	0.72	11.56	1.68	6.348
2	$4.178\ (10)^{-2}$	−158.0	1062.0	$1.340\ (10)^{4}$	11.630	0.48	11.65	1.40	6.084
1	$3.966\ (10)^{-2}$	−158.5	1061.5	$1.408\ (10)^{4}$	11.604	+ 0.24	11.62	1.12	5.777
0	$3.764\ (10)^{-2}$	−159.0	1061.1	$1.481\ (10)^{4}$	11.579	0.00	11.59	0.83	5.480
− 1	$3.572\ (10)^{-2}$	−159.4	1060.6	$1.577\ (10)^{4}$	11.554	− 0.24	11.57	0.55	5.201
− 2	$3.388\ (10)^{-2}$	−159.9	1060.2	$1.638\ (10)^{4}$	11.538	− 0.48	11.54	+ 0.27	4.932
− 3	$3.213\ (10)^{-2}$	−160.4	1059.8	$1.723\ (10)^{4}$	11.503	− 0.72	11.52	− 0.01	4.679
− 4	$3.047\ (10)^{-2}$	−160.9	1059.3	$1.814\ (10)^{4}$	11.478	− 0.96	11.49	− 0.29	4.434
− 5	$2.888\ (10)^{-2}$	−161.3	1058.9	$1.909\ (10)^{4}$	11.453	− 1.20	11.46	− 0.57	4.204
− 6	$2.737\ (10)^{-2}$	−161.8	1058.4	$2.010\ (10)^{4}$	11.427	− 1.44	11.44	− 0.84	3.983
− 7	$2.594\ (10)^{-2}$	−162.3	1058.0	$2.116\ (10)^{4}$	11.402	− 1.68	11.41	− 1.11	3.775
− 8	$2.457\ (10)^{-2}$	−162.7	1057.6	$2.229\ (10)^{4}$	11.377	− 1.92	11.39	− 1.38	3.576
− 9	$2.327\ (19)^{-2}$	−163.2	1057.1	$2.349\ (10)^{4}$	11.351	− 2.16	11.36	− 1.65	3.385
− 10	$2.203\ (10)^{-2}$	−163.6	1056.7	$2.475\ (10)^{4}$	11.326	− 2.40	11.33	− 1.92	3.206
− 11	$2.086\ (10)^{-2}$	−164.1	1056.2	$2.609\ (10)^{4}$	11.301	− 2.64	11.31	− 2.19	3.034
− 12	$1.974\ (10)^{-2}$	−164.6	1055.8	$2.750\ (10)^{4}$	11.276	− 2.88	11.28	− 2.45	2.872
− 13	$1.867\ (10)^{-2}$	−165.0	1055.4	$2.901\ (10)^{4}$	11.250	− 3.12	11.26	− 2.71	2.716
− 14	$1.766\ (10)^{-2}$	−165.5	1054.9	$3.060\ (10)^{4}$	11.225	− 3.37	11.23	− 2.98	2.569
− 15	$1.670\ (10)^{-2}$	−165.9	1054.5	$3.228\ (10)^{4}$	11.200	− 3.61	11.21	− 3.24	2.430
− 16	$1.579\ (10)^{-2}$	−166.4	1054.0	$3.407\ (10)^{4}$	11.174	− 3.85	11.18	− 3.50	2.297
− 17	$1.493\ (10)^{-2}$	−166.8	1053.6	$3.596\ (10)^{4}$	11.149	− 4.09	11.15	− 3.76	2.171
− 18	$1.410\ (10)^{-2}$	−167.3	1053.1	$3.797\ (10)^{4}$	11.123	− 4.33	11.13	− 4.02	2.052
− 19	$1.333\ (10)^{-2}$	−167.8	1052.7	$4.011\ (10)^{4}$	11.098	− 4.57	11.10	− 4.28	1.938
− 20	$1.259\ (10)^{-2}$	−168.2	1052.3	$4.237\ (10)^{4}$	11.073	− 4.81	11.08	− 4.53	1.830
− 21	$1.188\ (19)^{-2}$	−168.7	1051.8	$4.477\ (10)^{4}$	11.048	− 5.05	11.05	− 4.79	1.728
− 22	$1.122\ (10)^{-2}$	−169.1	1051.4	$4.732\ (10)^{4}$	11.022	− 5.29	11.03	− 5.04	1.631
− 23	$1.059\ (10)^{-2}$	−169.6	1050.9	$5.003\ (10)^{4}$	10.997	− 5.53	11.00	− 5.30	1.539
− 24	$9.987\ (10)^{-3}$	−170.0	1050.5	$5.290\ (10)^{4}$	10.972	− 5.77	10.98	− 5.55	1.452

Table 2. Thermodynamic Properties of Air, Ice and Steam (Continued)

nper-ure, F	Properties of Ice and Steam				Properties of Dry Air at a Pressure of 29.921 in. Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Saturation pressure of ice and steam, in. Hg	Enthalpy of sat ice, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, cu ft	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_i	h_g	v_g	v_a	h_a	v_s	h_s	W_s
25	$9.420\ (10)^{-3}$	−170.5	1050.1	$5.596\ (10)^{4}$	10.947	− 6.01	10.95	− 5.80	1.370
26	$8.882\ (10)^{-3}$	−170.9	1049.6	$5.921\ (10)^{4}$	10.921	− 6.25	10.92	− 6.05	1.292
27	$8.373\ (10)^{-3}$	−171.4	1049.2	$6.267\ (10)^{4}$	10.896	− 6.49	10.90	− 6.31	1.217
28	$7.891\ (10)^{-3}$	−171.8	1048.7	$6.634\ (10)^{4}$	10.871	− 6.73	10.87	− 6.56	1.147
29	$7.435\ (10)^{-3}$	−172.2	1048.3	$7.025\ (10)^{4}$	10.845	− 6.97	10.85	− 6.81	1.081
30	$7.003\ (10)^{-3}$	−172.7	1047.8	$7.441\ (10)^{4}$	10.820	− 7.21	10.82	− 7.06	1.018
31	$6.595\ (10)^{-3}$	−173.1	1047.4	$7.883\ (10)^{4}$	10.795	− 7.45	10.80	− 7.31	0.9587
32	$6.208\ (10)^{-3}$	−173.6	1047.0	$8.355\ (10)^{4}$	10.769	− 7.69	10.77	− 7.56	0.9024
33	$5.843\ (10)^{-3}$	−174.0	1046.5	$8.857\ (10)^{4}$	10.744	− 7.93	10.75	− 7.80	0.8493
34	$5.497\ (10)^{-3}$	−174.4	1046.1	$9.391\ (10)^{4}$	10.719	− 8.17	10.72	− 8.05	0.7991
35	$5.170\ (10)^{-3}$	−174.9	1045.6	$9.961\ (10)^{4}$	10.693	− 8.41	10.70	− 8.30	0.7516
36	$4.862\ (10)^{-3}$	−175.3	1045.2	$1.057\ (10)^{5}$	10.668	− 8.65	10.67	− 8.55	0.7066
37	$4.570\ (10)^{-3}$	−175.8	1044.7	$1.122\ (10)^{5}$	10.642	− 8.89	10.64	− 8.79	0.6641
38	$4.295\ (10)^{-3}$	−176.2	1044.3	$1.191\ (10)^{5}$	10.617	− 9.13	10.62	− 9.04	0.6241
39	$4.035\ (10)^{-3}$	−176.6	1043.9	$1.264\ (10)^{5}$	10.592	− 9.37	10.59	− 9.29	0.5866
40	$3.790\ (10)^{-3}$	−177.1	1043.4	$1.343\ (10)^{5}$	10.567	− 9.61	10.57	− 9.53	0.5508
41	$3.559\ (10)^{-3}$	−177.5	1043.0	$1.427\ (10)^{5}$	10.541	− 9.85	10.54	− 9.78	0.5172
42	$3.340\ (10)^{-3}$	−177.9	1042.5	$1.516\ (10)^{5}$	10.516	−10.10	10.52	−10.02	0.4856
43	$3.134\ (10)^{-3}$	−178.4	1042.1	$1.612\ (10)^{5}$	10.491	−10.34	10.49	−10.27	0.4556
44	$2.940\ (10)^{-3}$	−178.8	1041.6	$1.715\ (10)^{5}$	10.466	−10.58	10.47	−10.51	0.4272
45	$2.757\ (10)^{-3}$	−179.2	1041.2	$1.824\ (10)^{5}$	10.440	−10.82	10.44	−10.76	0.4007
46	$2.585\ (10)^{-3}$	−179.6	1040.8	$1.941\ (10)^{5}$	10.415	−11.06	10.42	−11.00	0.3756
47	$2.422\ (10)^{-3}$	−180.1	1040.3	$2.066\ (10)^{5}$	10.389	−11.30	10.39	−11.24	0.3520
48	$2.270\ (10)^{-3}$	−180.5	1039.9	$2.200\ (10)^{5}$	10.364	−11.54	10.37	−11.49	0.3298
49	$2.126\ (10)^{-3}$	−180.9	1039.4	$2.343\ (10)^{5}$	10.339	−11.77	10.34	−11.73	0.3089
50	$1.990\ (10)^{-3}$	−181.3	1039.0	$2.496\ (10)^{5}$	10.314	−12.01	10.31	−11.97	0.2893
51	$1.863\ (10)^{-3}$	−181.8	1038.6	$2.660\ (10)^{5}$	10.288	−12.26	10.29	−12.22	0.2708
52	$1.743\ (10)^{-3}$	−182.2	1038.1	$2.836\ (10)^{5}$	10.263	−12.50	10.26	−12.46	0.2533
53	$1.631\ (10)^{-3}$	−182.6	1037.7	$3.024\ (10)^{5}$	10.238	−12.74	10.24	−12.70	0.2370
54	$1.525\ (10)^{-3}$	−183.0	1037.2	$3.226\ (10)^{5}$	10.212	−12.98	10.21	−12.95	0.2216
55	$1.426\ (10)^{-3}$	−183.4	1036.8	$3.443\ (10)^{5}$	10.187	−13.22	10.19	−13.19	0.2071
56	$1.332\ (10)^{-3}$	−183.9	1036.3	$3.675\ (10)^{5}$	10.161	−13.46	10.16	−13.43	0.1936
57	$1.244\ (10)^{-3}$	−184.3	1035.9	$3.925\ (10)^{5}$	10.136	−13.70	10.14	−13.67	0.1808
58	$1.162\ (10)^{-3}$	−184.7	1035.5	$4.192\ (10)^{5}$	10.111	−13.94	10.11	−13.92	0.1688
59	$1.085\ (10)^{-3}$	−185.1	1035.0	$4.479\ (10)^{5}$	10.086	−14.18	10.09	−14.16	0.1576
60	$1.012\ (10)^{-3}$	−185.5	1034.6	$4.788\ (10)^{5}$	10.060	−14.42	10.06	−14.40	0.1471
61	$9.444\ (10)^{-4}$	−185.9	1034.1	$5.120\ (10)^{5}$	10.035	−14.66	10.03	−14.64	0.1372
62	$8.808\ (10)^{-4}$	−186.4	1033.7	$5.476\ (10)^{5}$	10.010	−14.90	10.01	−14.88	0.1280
63	$8.211\ (10)^{-4}$	−186.8	1033.2	$5.859\ (10)^{5}$	9.984	−15.14	9.984	−15.12	0.1193
64	$7.652\ (10)^{-4}$	−187.2	1032.8	$6.272\ (10)^{5}$	9.959	−15.38	9.959	−15.37	0.1112
65	$7.128\ (10)^{-4}$	−187.6	1032.4	$6.715\ (10)^{5}$	9.933	−15.62	9.934	−15.61	0.1036
66	$6.638\ (10)^{-4}$	−188.0	1031.9	$7.193\ (10)^{5}$	9.908	−15.86	9.908	−15.85	$9.642\ (10)^{-2}$
67	$6.180\ (10)^{-4}$	−188.4	1031.5	$7.707\ (10)^{5}$	9.883	−16.10	9.883	−16.09	$8.976\ (10)^{-2}$
68	$5.751\ (10)^{-4}$	−188.8	1031.0	$8.261\ (10)^{5}$	9.858	−16.34	9.858	−16.33	$8.353\ (10)^{-2}$
69	$5.349\ (10)^{-4}$	−189.2	1030.6	$8.858\ (10)^{5}$	9.832	−16.58	9.832	−16.57	$7.770\ (10)^{-2}$
70	$4.974\ (10)^{-4}$	−189.6	1030.2	$9.501\ (10)^{5}$	9.807	−16.82	9.807	−16.81	$7.225\ (10)^{-2}$
71	$4.624\ (10)^{-4}$	−190.0	1029.7	$1.020\ (10)^{6}$	9.781	−17.06	9.781	−17.06	$6.713\ (10)^{-2}$
72	$4.296\ (10)^{-4}$	−190.4	1029.3	$1.094\ (10)^{6}$	9.756	−17.31	9.756	−17.30	$6.242\ (10)^{-2}$
73	$3.991\ (10)^{-4}$	−190.8	1028.8	$1.175\ (10)^{6}$	9.731	−17.55	9.731	−17.54	$5.797\ (10)^{-2}$
74	$3.705\ (10)^{-4}$	−191.2	1028.4	$1.262\ (10)^{6}$	9.705	−17.79	9.705	−17.78	$5.383\ (10)^{-2}$
75	$3.439\ (10)^{-4}$	−191.6	1027.9	$1.356\ (10)^{6}$	9.680	−18.03	9.680	−18.02	$4.997\ (10)^{-2}$
76	$3.191\ (10)^{-4}$	−192.0	1027.5	$1.458\ (10)^{6}$	9.655	−18.27	9.655	−18.26	$4.635\ (10)^{-2}$
77	$2.959\ (10)^{-4}$	−192.4	1027.1	$1.568\ (10)^{6}$	9.629	−18.51	9.629	−18.50	$4.299\ (10)^{-2}$
78	$2.743\ (10)^{-4}$	−192.8	1026.6	$1.687\ (10)^{6}$	9.604	−18.75	9.604	−18.74	$3.985\ (10)^{-2}$
79	$2.542\ (10)^{-4}$	−193.2	1026.2	$1.816\ (10)^{6}$	9.579	−18.99	9.579	−18.99	$3.692\ (10)^{-2}$
80	$2.355\ (10)^{-4}$	−193.6	1025.7	$1.955\ (10)^{6}$	9.553	−19.23	9.553	−19.23	$3.421\ (10)^{-2}$
81	$2.181\ (10)^{-4}$	−194.0	1025.3	$2.106\ (10)^{6}$	9.528	−19.47	9.528	−19.47	$3.167\ (10)^{-2}$
82	$2.018\ (10)^{-4}$	−194.4	1024.8	$2.270\ (10)^{6}$	9.503	−19.71	9.502	−19.71	$2.930\ (10)^{-2}$
83	$1.867\ (10)^{-4}$	−194.8	1024.4	$2.446\ (10)^{6}$	9.477	−19.95	9.477	−19.95	$2.712\ (10)^{-2}$
84	$1.727\ (10)^{-4}$	−195.2	1024.0	$2.638\ (10)^{6}$	9.472	−20.19	9.452	−20.19	$2.508\ (10)^{-2}$

Table 2. Thermodynamic Properties of Air, Ice and Steam (Continued)

Temperature, F	Properties of Ice and Steam				Properties of Dry Air at a Pressure of 29.921 in.Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Saturation pressure of ice and steam, in. Hg	Enthalpy of sat ice, Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	True specific volume, cu ft/lb	Enthalpy, Btu/lb	Volume of mixture per lb of dry air, cu ft	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_i	h_g	v_g	v_a	h_a	v_s	h_s	W_s
− 85	1.596 (10)$^{-4}$	−195.6	1023.5	2.846 (10)^{6}	9.426	−20.43	9.426	−20.43	2.318 (10)$^{-2}$
− 86	1.475 (10)$^{-4}$	−195.9	1023.1	3.072 (10)^{6}	9.401	−20.67	9.401	−20.67	2.142 (10)$^{-2}$
− 87	1.363 (10)$^{-4}$	−196.3	1022.6	3.317 (10)^{6}	9.376	−20.91	9.376	−20.91	1.979 (10)$^{-2}$
− 88	1.258 (10)$^{-4}$	−196.7	1022.2	3.583 (10)^{6}	9.350	−21.15	9.350	−21.15	1.827 (10)$^{-2}$
− 89	1.161 (10)$^{-4}$	−197.1	1021.8	3.872 (10)^{6}	9.325	−21.40	9.325	−21.40	1.686 (10)$^{-2}$
− 90	1.071 (10)$^{-4}$	−197.5	1021.3	4.186 (10)^{6}	9.299	−21.64	9.300	−21.64	1.555 (10)$^{-2}$
− 91	9.876 (10)$^{-5}$	−197.9	1020.9	4.528 (10)^{6}	9.274	−21.88	9.274	−21.88	1.434 (10)$^{-2}$
− 92	9.102 (10)$^{-5}$	−198.2	1020.4	4.899 (10)^{6}	9.249	−22.12	9.249	−22.12	1.322 (10)$^{-2}$
− 93	8.385 (10)$^{-5}$	−198.6	1020.0	5.303 (10)^{6}	9.224	−22.36	9.224	−22.36	1.218 (10)$^{-2}$
− 94	7.722 (10)$^{-5}$	−199.0	1019.5	5.743 (10)^{6}	9.198	−22.60	9.198	−22.60	1.121 (10)$^{-2}$
− 95	7.108 (10)$^{-5}$	−199.4	1019.1	6.223 (10)^{6}	9.173	−22.84	9.173	−22.84	1.032 (10)$^{-2}$
− 96	6.539 (10)$^{-5}$	−199.8	1018.7	6.745 (10)^{6}	9.147	−23.08	9.147	−23.08	9.493 (10)$^{-3}$
− 97	6.014 (10)$^{-5}$	−200.2	1018.2	7.314 (10)^{6}	9.122	−23.32	9.122	−23.32	8.730 (10)$^{-3}$
− 98	5.528 (10)$^{-5}$	−200.5	1017.8	7.936 (10)^{6}	9.097	−23.56	9.097	−23.56	8.023 (10)$^{-3}$
− 99	5.079 (10)$^{-5}$	−200.9	1017.3	8.613 (10)^{6}	9.071	−23.80	9.071	−23.80	7.372 (10)$^{-3}$
−100	4.664 (10)$^{-5}$	−201.3	1016.9	9.352 (10)^{6}	9.046	−24.04	9.046	−24.04	6.771 (10)$^{-3}$
−101	4.281 (10)$^{-5}$	−201.6	1016.5	1.016 (10)^{7}	9.020	−24.28	9.020	−24.28	6.215 (10)$^{-3}$
−102	3.928 (10)$^{-5}$	−202.0	1016.1	1.104 (10)^{7}	8.995	−24.52	8.995	−24.52	5.703 (10)$^{-3}$
−103	3.602 (10)$^{-5}$	−202.4	1015.6	1.201 (10)^{7}	8.970	−24.77	8.970	−24.77	5.228 (10)$^{-3}$
−104	3.301 (10)$^{-5}$	−202.8	1015.1	1.307 (10)^{7}	8.944	−25.01	8.944	−25.01	4.790 (10)$^{-3}$
−105	3.025 (10)$^{-5}$	−203.1	1014.7	1.422 (10)^{7}	8.919	−25.25	8.919	−25.25	4.390 (10)$^{-3}$
−106	2.770 (10)$^{-5}$	−203.5	1014.2	1.549 (10)^{7}	8.894	−25.49	8.894	−25.49	4.019 (10)$^{-3}$
−107	2.535 (10)$^{-5}$	−203.9	1013.8	1.687 (10)^{7}	8.868	−25.73	8.868	−25.73	3.680 (10)$^{-3}$
−108	2.319 (10)$^{-5}$	−204.2	1013.4	1.839 (10)^{7}	8.843	−25.97	8.843	−25.97	3.366 (10)$^{-3}$
−109	2.120 (10)$^{-5}$	−204.6	1012.9	2.006 (10)^{7}	8.817	−26.21	8.817	−26.21	3.077 (10)$^{-3}$
−110	1.938 (10)$^{-5}$	−205.0	1012.5	2.189 (10)^{7}	8.792	−26.45	8.792	−26.45	2.812 (10)$^{-3}$
−111	1.770 (10)$^{-5}$	−205.3	1012.0	2.389 (10)^{7}	8.767	−26.69	8.766	−26.69	2.569 (10)$^{-3}$
−112	1.616 (10)$^{-5}$	−205.7	1011.6	2.610 (10)^{7}	8.741	−26.93	8.741	−26.93	2.344 (10)$^{-3}$
−113	1.476 (10)$^{-5}$	−206.1	1011.1	2.852 (10)^{7}	8.716	−27.17	8.716	−27.17	2.139 (10)$^{-3}$
−114	1.344 (10)$^{-5}$	−206.4	1010.7	3.118 (10)^{7}	8.690	−27.41	8.690	−27.41	1.951 (10)$^{-3}$
−115	1.226 (10)$^{-5}$	−206.8	1010.3	3.411 (10)^{7}	8.665	−27.66	8.665	−27.66	1.778 (10)$^{-3}$
−116	1.116 (10)$^{-5}$	−207.2	1009.8	3.733 (10)^{7}	8.640	−27.90	8.640	−27.90	1.620 (10)$^{-3}$
−117	1.017 (10)$^{-5}$	−207.5	1009.4	4.088 (10)^{7}	8.614	−28.14	8.614	−28.14	1.475 (10)$^{-3}$
−118	9.251 (10)$^{-6}$	−207.9	1008.9	4.480 (10)^{7}	8.589	−28.38	8.589	−28.38	1.342 (10)$^{-3}$
−119	8.414 (10)$^{-6}$	−208.2	1008.5	4.911 (10)^{7}	8.563	−28.62	8.563	−28.62	1.221 (10)$^{-3}$
−120	7.649 (10)$^{-6}$	−208.6	1008.0	5.386 (10)^{7}	8.538	−28.86	8.538	−28.86	1.110 (10)$^{-3}$
−121	6.949 (10)$^{-6}$	−208.9	1007.6	5.911 (10)^{7}	8.512	−29.10	8.512	−29.10	1.008 (10)$^{-3}$
−122	6.310 (10)$^{-6}$	−209.3	1007.2	6.491 (10)^{7}	8.487	−29.34	8.487	−29.34	9.152 (10)$^{-4}$
−123	5.726 (10)$^{-6}$	−209.6	1006.7	7.131 (10)^{7}	8.462	−29.58	8.462	−29.58	8.306 (10)$^{-4}$
−124	5.194 (10)$^{-6}$	−210.0	1006.3	7.839 (10)^{7}	8.436	−29.82	8.436	−29.82	7.533 (10)$^{-4}$
−125	4.708 (10)$^{-6}$	−210.3	1005.8	8.622 (10)^{7}	8.411	−30.06	8.411	−30.06	6.829 (10)$^{-4}$
−126	4.265 (10)$^{-6}$	−210.7	1005.4	9.489 (10)^{7}	8.386	−30.30	8.386	−30.30	6.186 (10)$^{-4}$
−127	3.862 (10)$^{-6}$	−211.0	1005.0	1.045 (10)^{8}	8.360	−30.55	8.360	−30.55	5.600 (10)$^{-4}$
−128	3.494 (10)$^{-6}$	−211.4	1004.5	1.151 (10)^{8}	8.335	−30.79	8.335	−30.79	5.069 (10)$^{-4}$
−129	3.160 (10)$^{-6}$	−211.7	1004.1	1.269 (10)^{8}	8.309	−31.03	8.309	−31.03	4.584 (10)$^{-4}$
−130	2.856 (10)$^{-6}$	−212.1	1003.6	1.400 (10)^{8}	8.284	−31.27	8.284	−31.27	4.142 (10)$^{-4}$
−131	2.579 (10)$^{-6}$	−212.4	1003.2	1.545 (10)^{8}	8.258	−31.51	8.258	−31.51	3.742 (10)$^{-4}$
−132	2.328 (10)$^{-6}$	−212.8	1002.8	1.707 (10)^{8}	8.233	−31.75	8.233	−31.75	3.376 (10)$^{-4}$
−133	2.101 (10)$^{-6}$	−213.1	1002.3	1.886 (10)^{8}	8.207	−31.99	8.207	−31.99	3.046 (10)$^{-4}$
−134	1.894 (10)$^{-6}$	−213.5	1001.9	2.086 (10)^{8}	8.182	−32.23	8.182	−32.23	2.746 (10)$^{-4}$
−135	1.706 (10)$^{-6}$	−213.8	1001.4	2.308 (10)^{8}	8.157	−32.47	8.157	−32.47	2.474 (10)$^{-4}$
−136	1.536 (10)$^{-6}$	−214.1	1001.0	2.555 (10)^{8}	8.131	−32.71	8.131	−32.71	2.228 (10)$^{-4}$
−137	1.382 (10)$^{-6}$	−214.5	1000.5	2.831 (10)^{8}	8.106	−32.96	8.106	−32.96	2.004 (10)$^{-4}$
−138	1.243 (10)$^{-6}$	−214.8	1000.1	3.138 (10)^{8}	8.080	−33.20	8.080	−33.20	1.802 (10)$^{-4}$
−139	1.117 (10)$^{-6}$	−215.2	999.7	3.481 (10)^{8}	8.055	−33.44	8.055	−33.44	1.620 (10)$^{-4}$
−140	1.003 (10)$^{-6}$	−215.5	999.2	3.864 (10)^{8}	8.029	−33.68	8.029	−33.68	1.455 (10)$^{-4}$
−141	9.005 (10)$^{-7}$	−215.8	998.8	4.292 (10)^{8}	8.004	−33.92	8.004	−33.92	1.305 (10)$^{-4}$
−142	8.076 (10)$^{-7}$	−216.2	998.3	4.770 (10)^{8}	7.978	−34.16	7.978	−34.16	1.171 (10)$^{-4}$
−143	7.240 (10)$^{-7}$	−216.5	997.9	5.305 (10)^{8}	7.953	−34.40	7.953	−34.40	1.049 (10)$^{-4}$
−144	6.483 (10)$^{-7}$	−216.8	997.4	5.905 (10)^{8}	7.928	−34.65	7.928	−34.65	9.397 (10)$^{-5}$

Table 2. Thermodynamic Properties of Air, Ice and Steam (Concluded)

emperature, F	Properties of Ice and Steam				Properties of Dry Air at a Pressure of 29.921 in.Hg abs		Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Saturation pressure of ice and steam, in. Hg	Enthalpy of sat ice, Btu lb	Enthalpy of sat steam, Btu lb	Specific volume of sat steam, cu ft lb	True specific volume, cu ft lb	Enthalpy, Btu lb	Volume of mixture per lb of dry air, cu ft	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
t	p_s	h_i	h_g	v_g	v_a	h_a	v_s	h_s	W_s
−145	5.803 (10)$^{-7}$	−217.2	997.0	6.577 (10)^{8}	7.902	−34.89	7.902	−34.89	8.410 (10)$^{-6}$
−146	5.190 (10)$^{-7}$	−217.5	996.6	7.330 (10)^{8}	7.876	−35.13	7.877	−35.13	7.522 (10)$^{-6}$
−147	4.639 (10)$^{-7}$	−217.8	996.1	8.174 (10)^{8}	7.851	−35.37	7.851	−35.37	6.724 (10)$^{-6}$
−148	4.143 (10)$^{-7}$	−218.2	995.7	9.123 (10)^{8}	7.826	−35.61	7.826	−35.61	6.005 (10)$^{-6}$
−149	3.698 (10)$^{-7}$	−218.5	995.2	10.19 (10)^{8}	7.800	−35.85	7.800	−35.85	5.358 (10)$^{-6}$
−150	3.298 (10)$^{-7}$	−218.8	994.8	11.39 (10)^{8}	7.775	−36.09	7.775	−36.09	4.778 (10)$^{-6}$
−151	2.939 (10)$^{-7}$	−219.1	994.4	12.74 (10)^{8}	7.749	−36.33	7.749	−36.33	4.258 (10)$^{-6}$
−152	2.618 (10)$^{-7}$	−219.5	993.9	14.26 (10)^{8}	7.724	−36.58	7.724	−36.58	3.792 (10)$^{-6}$
−153	2.329 (10)$^{-7}$	−219.8	993.5	15.97 (10)^{8}	7.698	−36.82	7.698	−36.82	3.374 (10)$^{-6}$
−154	2.072 (10)$^{-7}$	−220.1	993.0	17.90 (10)^{8}	7.673	−37.06	7.673	−37.06	3.001 (10)$^{-6}$
−155	1.840 (10)$^{-7}$	−220.4	992.6	20.08 (10)^{8}	7.647	−37.30	7.647	−37.30	2.666 (10)$^{-6}$
−156	1.634 (10)$^{-7}$	−220.8	992.1	22.54 (10)^{8}	7.622	−37.54	7.622	−37.54	2.367 (10)$^{-6}$
−157	1.450 (10)$^{-7}$	−221.1	991.7	25.32 (10)^{8}	7.596	−37.78	7.597	−37.78	2.100 (10)$^{-6}$
−158	1.285 (10)$^{-7}$	−221.4	991.3	28.47 (10)^{8}	7.571	−38.02	7.571	−38.02	1.861 (10)$^{-6}$
−159	1.138 (10)$^{-7}$	−221.7	990.8	32.03 (10)^{8}	7.546	−38.27	7.546	−38.27	1.649 (10)$^{-6}$
−160	1.008 (10)$^{-7}$	−222.0	990.4	36.07 (10)^{8}	7.520	−38.51	7.520	−38.51	1.459 (10)$^{-6}$

Table 3. Condensed Table of Thermodynamic Properties of Air, Water (Ice) and Steam (Giving Magnitudes of Pressure Corrections)*

Temperature, F	Properties of Dry Air at a Pressure of 29.921 in. Hg abs						Properties of Water (Ice) and Steam				Properties of Mixture of Dry Air and Sat Steam at a Total Pressure of 29.921 in. Hg abs		
	Sp vol, cu ft/lb (varies with pressure)	Sp vol, cu ft/lb correction (independent of pressure)	True specific volume, cu ft/lb	Enthalpy, Btu/lb at zero pressure	Enthalpy, Btu/lb correction (varies with pressure)	Enthalpy, Btu/lb	Saturation pressure of ice (water) and steam, in. Hg	Enthalpy of sat ice (water), Btu/lb	Enthalpy of sat steam, Btu/lb	Specific volume of sat steam, cu ft/lb	Volume of mixture per lb of dry air, cu ft/lb	Enthalpy of mixture per lb of dry air, Btu	Specific humidity, grains per lb of dry air
	v_1	v_2	v_a	h_1	h_2	h_a	p_s	h_i (or) h_f	h_g	v_g	v_s	h_s	W_s
Part I. Conditions over Ice													
−150	7.808	0.033	7.775	−35.78	0.31	−36.09	$3.298(10)^{-7}$	−218.8	994.8	$11.39(10)^{8}$	7.775	−36.09	$4.778(10)^{-5}$
−140	8.060	0.031	8.029	−33.39	0.29	−33.68	$1.003(10)^{-6}$	−215.5	999.2	$3.864(10)^{8}$	8.029	−33.68	$1.455(10)^{-4}$
−130	8.312	0.028	8.284	−31.00	0.27	−31.27	$2.856(10)^{-6}$	−212.1	1003.6	$1.400(10)^{8}$	8.284	−31.27	$4.142(10)^{-4}$
−120	8.564	0.026	8.538	−28.60	0.26	−28.86	$7.649(10)^{-6}$	−208.6	1008.0	$5.386(10)^{7}$	8.538	−28.86	$1.110(10)^{-3}$
−110	8.816	0.024	8.792	−26.21	0.24	−26.45	$1.938(10)^{-5}$	−205.0	1012.5	$2.189(10)^{7}$	8.792	−26.45	$2.812(10)^{-3}$
−100	9.068	0.022	9.046	−23.81	0.23	−24.04	$4.664(10)^{-5}$	−201.3	1016.9	$9.352(10)^{6}$	9.046	−24.04	$6.771(10)^{-3}$
− 90	9.320	0.021	9.299	−21.42	0.22	−21.64	$1.071(10)^{-4}$	−197.5	1021.3	$4.186(10)^{6}$	9.300	−21.64	$1.555(10)^{-2}$
− 80	9.572	0.019	9.553	−19.03	0.20	−19.23	$2.355(10)^{-4}$	−193.6	1025.7	$1.955(10)^{6}$	9.553	−19.23	$3.421(10)^{-2}$
− 70	9.824	0.017	9.807	−16.63	0.19	−16.82	$4.974(10)^{-4}$	−189.6	1030.2	$9.501(10)^{5}$	9.807	−16.81	$7.225(10)^{-2}$
− 60	10.076	0.016	10.060	−14.24	0.18	−14.42	$1.012(10)^{-3}$	−185.5	1034.6	$4.788(10)^{5}$	10.06	−14.40	0.1471
− 50	10.329	0.015	10.314	−11.84	0.17	−12.01	$1.990(10)^{-3}$	−181.3	1039.0	$2.496(10)^{5}$	10.31	−11.97	0.2893
− 40	10.581	0.014	10.567	− 9.45	0.16	− 9.61	$3.790(10)^{-3}$	−177.1	1043.4	$1.343(10)^{5}$	10.57	− 9.53	0.5508
− 30	10.833	0.013	10.820	− 7.05	0.16	− 7.21	$7.003(10)^{-3}$	−172.7	1047.8	$7.441(10)^{4}$	10.82	− 7.06	1.018
− 20	11.085	0.012	11.073	− 4.66	0.15	− 4.81	$1.259(10)^{-2}$	−168.2	1052.3	$4.237(10)^{4}$	11.08	− 4.53	1.830
− 10	11.337	0.011	11.326	− 2.26	0.14	− 2.40	$2.203(10)^{-2}$	−163.6	1056.7	$2.475(10)^{4}$	11.33	− 1.92	3.206
0	11.589	0.010	11.579	+ 0.14	0.14	0.00	$3.764(10)^{-2}$	−159.0	1061.1	$1.481(10)^{4}$	11.59	0.83	5.480
10	11.841	0.009	11.832	2.53	0.13	2.40	$6.286(10)^{-2}$	−154.2	1065.5	9060	11.86	3.80	9.161
20	12.093	0.008	12.085	4.93	0.12	4.81	0.1027	−149.4	1069.9	5662	12.13	7.10	14.99
30	12.345	0.007	12.338	7.33	0.12	7.21	0.1645	−144.4	1074.3	3608	12.41	10.90	24.07
32	12.396	0.007	12.389	7.81	0.12	7.69	0.1803	−143.4	1075.2	3305	12.46	11.75	26.40
Part II. Conditions over Water													
32	12.396	0.007	12.389	7.81	0.12	7.69	0.1803	0.0	1075.2	3305	12.46	11.75	26.40
40	12.598	0.007	12.591	9.73	0.12	9.61	0.2477	8.0	1078.7	2445	12.70	15.21	36.34
50	12.850	0.006	12.844	12.13	0.11	12.02	0.3624	18.1	1083.1	1704	13.00	20.28	53.40
60	13.101	0.005	13.096	14.53	0.11	14.42	0.5216	28.1	1087.4	1207	13.33	26.43	77.29
70	13.354	0.005	13.349	16.93	0.10	16.83	0.7392	38.1	1091.8	868.0	13.69	34.04	110.4
80	13.606	0.004	13.602	19.33	0.10	19.23	1.0323	48.1	1096.1	633.0	14.09	43.63	155.8
90	13.858	0.004	13.854	21.73	0.09	21.64	1.4219	58.0	1100.4	467.9	14.55	55.85	217.6
100	14.110	0.003	14.107	24.13	0.09	24.04	1.9334	68.0	1104.7	350.2	15.08	71.62	301.5
110	14.362	0.002	14.360	26.54	0.09	26.45	2.5968	78.0	1109.0	265.3	15.72	92.19	414.9
120	14.614	0.002	14.612	28.93	0.08	28.85	3.4477	88.0	1113.3	203.2	16.51	119.36	569.0
130	14.866	0.001	14.865	31.34	0.08	31.26	4.5274	98.0	1117.5	157.3	17.52	155.72	779.6
140	15.118	0.001	15.117	33.75	0.08	33.67	5.8842	108.0	1121.7	123.0	18.82	205.34	1071
150	15.371	0.001	15.370	36.14	0.07	36.07	7.5727	118.0	1125.8	97.04	20.58	274.84	1485
160	15.622	0.000	15.622	38.55	0.07	38.48	9.6556	128.0	1129.9	77.27	23.07	375.81	2090
170	15.875	0.000	15.875	40.96	0.07	40.89	12.203	138.0	1133.9	62.04	26.81	530.86	3024
180	16.127	−0.001	16.128	43.37	0.07	43.30	15.295	148.0	1137.9	50.22	32.99	790.88	4598
190	16.379	−0.001	16.380	45.77	0.06	45.71	19.017	158.0	1141.9	40.96	44.94	1298.9	7682
200	16.631	−0.001	16.632	48.18	0.06	48.12	23.468	168.1	1145.8	33.64	77.14	2675.6	16052

11. WEATHER AND WATER DATA

THE data included in this chapter are limited in their scope to the information necessary to permit general engineering work and the design of both summer and winter air conditioning systems. Water temperature data are applicable for use in general industrial design as well as in refrigeration equipment selection.

The tabular data in this chapter are divided into three main classifications—winter weather conditions, summer weather conditions, and water temperature data for summer.

Winter Weather Conditions

Winter weather conditions are based on certain temperatures occurring during the colder periods of the normal heating season. There is a wide variation in the duration of the heating season among various localities or sections of the country. The variation is from approximately 135 days in certain parts of Arizona, Florida, and Texas to 365 days, or all year round, in San Francisco.

The duration of the heating season, however, is not necessarily a criterion of heating requirements, since San Francisco with a year-round heating season has a mean temperature of 56 F and a resultant average degree-day figure of 2,500 annually, while Memphis, Tenn., with a heating season of about 195 days, has a mean temperature of about 50 F during this period, with a corresponding degree-day figure of more than 3,000.

The choice of a design temperature in any locality is a matter of judgment and experience. Several methods for establishing winter outside design temperatures have been proposed and used successfully, although there is not exact agreement among these methods.

The most important factors influencing the design of heating or winter air conditioning are outside design temperature and wind direction and velocity. The choice of the design temperature is probably most logically based on the frequency of its occurrence. Other interesting data are elevation, latitude, lowest temperature on record, and average annual minimum temperature. The latter temperature is obtained by averaging the lowest temperatures (one for each year) recorded for the period of years the local weather bureau has been in operation.

Table 1 lists the data outlined in the foregoing paragraph for representative cities in each state of the United States and some data on Canadian cities.

Elevations given in column A, Table 1, are based on information from the "Dictionary of Elevations" issued by the United States Geodetic Survey. The elevations shown are not necessarily at the weather stations for the same localities. In many instances, the stations are located some distance from the city and at higher elevations. Wherever possible, the elevations are those listed at public buildings in the city proper, such as municipal buildings or court houses. In a few cases, railroad elevations at local depots have been tabulated.

Latitude in degrees and minutes is given in column B, Table 1. Most of these data are from previously published sources. The Canadian latitudes have been approximated from atlas maps.

Column C, Table 1, shows the lowest temperature on record from the United States Department of Agriculture's "Yearbook of Agriculture," 1941 Edition.

The average annual minimum temperatures shown in column D, Table 1, are taken from official U. S. Weather Bureau records, in most cases representing an average of all data available since the establishment of the weather station.

The recommended design winter outside dry-bulb temperature is given in column E.

Table 1. Winter Weather Conditions

(All temperatures in deg F)

State	City	A	B	C	D	E	F	G
		Elevation	Latitude degrees and minutes	Lowest temp on record	Average annual min temp	Recommended design temp	Average winter wind direction and velocity	Design temperature frequency occurrence*
Ala.	Birmingham	694	33–32	−10	12	10	N— 8.0	20– 5
	Mobile	15	30–42	− 1	22	20	N— 10.2	17– 3
Ariz.	Flagstaff	6,907	35–12	−30	−15	−15	SW— 7.8	20– 5
	Phoenix	1,083	33–28	16	26	25	E— 5.3	15– 3
Ark.	Fort Smith	448	35–32	−15	6	5	E— 8.1	23– 7
	Little Rock	288	34–45	−12	10	5	NW— 8.3	8– 4
Calif.	Los Angeles	286	34– 3	28	37	35	NE— 6.3	15– 6
	San Francisco	6	37–47	27	37	35	N— 7.5	27– 7
Colo.	Denver	5,183	39–45	−29	−11	−10	S— 7.5	33– 9
	Grand Junction	4,573	39– 4	−21	− 2	−10	NW— 4.3	40– 9
Conn.	New Haven	11	41–18	−15	− 1	− 5	N— 9.5	14– 5
Del.	Dover	34	39– 8	−11	10	5	NW—14	—
	Wilmington	8	39–43	−15	6	0	NW—12	—
D. C.	Washington	5	38–54	−15	− 1	0	NW— 7.9	—
Fla.	Jacksonville	8	30–20	10	29	25	NE— 9.1	24– 5
	Tampa	15	27–57	19	30	35	NE— 8	34– 6
Ga.	Atlanta	1,050	33–39	− 8	12	10	NW—11.4	20– 3
	Savannah	21	32– 5	8	22	15	NW— 9.5	3– 2
Idaho	Lewiston	756	46–25	−23	1	− 5	E— 5.3	48– 9
	Pocatello	4,468	42–52	−22	−12	−10	SE— 9	52– 9
Ill.	Chicago	600	41–47	−23	− 8	−10	W— 12.5	18– 6
	Springfield	612	39–48	−24	− 7	−10	NW—12.1	15– 3
Ind.	Evansville	383	37–58	−16	1	0	S— 9.8	34–10
	Fort Wayne	788	41– 5	−24	−10	− 5	SW—10.5	54–13
	Indianapolis	708	39–46	−25	− 6	− 5	S— 11.5	42–12
Iowa	Dubuque	647	42–30	−32	−17	−15	NW— 7.1	46– 9
	Sioux City	1,107	42–30	−35	−20	−20	NW—11.6	23– 7

Table 1. Winter Weather Conditions (Continued)

State	City	A Elevation	B Latitude degrees and minutes	C Lowest temp on record	D Average annual min temp	E Recommended design temp	F Average winter wind direction and velocity	G Design temperature frequency occurrence*
Kan.	Concordia	1,375	39–35	−25	−13	−10	N— 7.7	24– 6
	Dodge City	2,522	37–45	−26	−10	−10	NW—10.3	16– 4
Ky.	Louisville	449	48–15	−20	− 5	0	SW— 9.9	—
La.	New Orleans	2	29–57	7	26	20	NE— 8.7	4– 2
	Shreveport	204	32–30	− 5	16	10	SE— 8.1	5– 2
Maine	Eastport	33	44–54	−23	−15	−10	NW—12.6	40– 8
	Portland	14	43–39	−21	− 6	−10	NW— 9.2	18– 5
Md.	Baltimore	20	39–17	− 7	8	0	SW— 8.1	4– 2
Mass.	Boston	10	42–21	−18	− 3	− 5	W— 11.2	16– 5
Mich.	Alpena	587	45– 4	−26	−12	−15	W— 11.0	16– 5
	Grand Rapids	638	42–58	−24	−10	−10	W— 9.0	9– 3
	Detroit	583	42–24	−24	−11	−10	SW—12.7	8– 3
	Marquette	652	46–34	−27	−13	−15	NW—10.8	21– 5
Minn.	Duluth	609	46–47	−41	−28	−30	SW—13.5	21– 5
	Minneapolis	812	44–59	−34	−23	−20	NW—11.3	46–11
Miss	Vicksburg	100	32–22	− 1	18	15	SE— 8.3	17– 6
Mo.	St. Joseph	957	39–49	−24	−12	−15	W— 9.3	11– 3
	St. Louis	460	38–38	−22	− 2	− 5	S— 11.7	16– 6
	Springfield	1,301	37–12	−29	− 5	− 5	SE— 10.9	23– 6
Mont.	Billings	3,117	45–43	−49	−30	−30	SW—11.9	27– 4
	Havre	2,480	48–34	−57	−36	−40	SW— 9.5	9– 5
Neb.	Lincoln	1,180	40–49	−29	−13	−15	S— 10.7	28– 9
	North Platte	2,805	41– 8	−35	−17	−20	W— 8.3	14– 3
Nev.	Reno	4,490	39–32	−19	−10	− 5	W— 6.1	17– 6
	Tonopah	6,033	38– 6	−15	− 2	0	SE— 10.0	18– 5
	Winnemucca	4,287	40–58	−28	−10	−15	NE— 8.2	36– 9
N. Hamp.	Concord	244	43–12	−32	−15	−15	NW— 6.6	28– 7
N. J.	Atlantic City	10	39–22	− 9	6	0	NW—15.6	6– 5

Table 1. Winter Weather Conditions (Continued)

State	City	A Elevation	B Latitude degrees and minutes	C Lowest temp on record	D Average annual min temp	E Recommended design temp	F Average winter wind direction and velocity	G Design temperature frequency occurrence*
N. Y.	Albany	16	42–39	−24	−11	−10	S— 8.4	34– 9
	Buffalo	580	42–53	−20	− 4	− 5	W— 17.2	21– 7
	New York	10	40–43	−14	− 3	− 5	NW—16.7	8– 5
N. Mex.	Santa Fe	6,950	35–41	−13	0	0	NE— 7.8	27– 4
N. C.	Raleigh	316	35–45	− 2	13	15	SW— 7.9	46–11
	Wilmington	6	34–14	5	18	20	W— 8.1	54– 8
N. D.	Bismarck	1,672	46–48	−45	−31	−30	NW— 9.0	38– 6
	Devils Lake	1,471	48– 7	−46	−33	−30	W— 10.4	67–14
Ohio	Cincinnati	490	39– 9	−17	− 5	− 5	SW— 8.3	27– 9
	Cleveland	582	41–30	−17	− 2	− 5	SW—15	19– 5
	Columbus	780	39–58	−20	− 3	− 5	SW—11.6	30–10
Okla.	Okla. City	1,197	35–26	−17	2	0	NW—11.8	19– 6
Ore.	Baker	3,435	44–46	−24	−17	−15	SE— 6.8	23– 5
	Portland	29	45–32	− 2	18	10	S— 7.4	3– 3
Penna.	Philadelphia	9	39–57	−11	6	0	NW—11	6– 3
	Pittsburgh	744	40–21	−20	− 2	− 5	W— 11.7	14– 7
	Scranton	737	41–24	−19	−10	− 5	SW— 7.9	20– 7
R. I.	Providence	11	41–50	−17	1	− 5	NW—12.6	13– 5
S. C.	Charleston	10	32–47	7	22	20	SW—10.6	16– 6
	Columbia	312	34– 0	− 2	19	15	NE— 8.1	16– 5
S. D.	Huron	1,282	44–21	−43	−26	−20	NW—10.5	62–14
	Rapid City	3,231	44– 4	−40	−21	−20	W— 7.9	52–11
Tenn.	Knoxville	890	35–58	−16	2	5	SW— 7.1	18– 4
	Memphis	270	35– 9	− 9	9	5	S— 9.4	10– 4
Tex.	El Paso	3,695	31–47	− 5	16	15	NW— 9.0	8– 3
	Ft. Worth	616	32–45	− 8	12	10	NW—10.3	12– 3
	San Antonio	654	29–27	4	21	20	NE— 8.3	18– 5
Utah	Modena	5,460	37–48	−27	−15	−15	W— 9.3	37– 9
	Salt Lake City	4,220	40–46	−20	2	− 5	SE— 6.7	15– 6
Vt.	Burlington	112	44–29	−30	−17	−15	S— 11.7	60–12

Table 1. Winter Weather Conditions (Concluded)

State	City	A Elevation	B Latitude degrees and minutes	C Lowest temp on record	D Average annual min temp	E Recommended design temp	F Average winter wind direction and velocity	G Design temperature frequency occurrence*
Va.	Lynchburg	686	37–25	— 7	8	5	NW— 8.1	25– 7
	Norfolk	12	36–51	2	15	15	N— 12.4	34– 8
	Richmond	20	37–32	— 3	10	10	SW— 8.0	—
Wash.	Seattle	10	47–36	3	20	15	SE— 10.1	11– 3
	Spokane	1,890	47–40	—30	— 5	—15	SW— 6.3	7– 2
W. Va.	Elkins	1,940	38–53	—28	— 8	—10	W— 5.8	15– 3
	Parkersburg	615	39–16	—27	— 1	— 5	SW— 7.3	17– 7
Wis.	Green Bay	589	44–31	—36	—18	—20	SW—10.6	21– 4
	La Crosse	673	43–49	—43	—21	—25	S— 5.6	16– 4
	Milwaukee	591	43– 2	—25	—12	—15	W— 12.0	16– 5
Wyo.	Lander	5,370	42–50	—40	—12	—20	SW— 3.9	85– 9
	Sheridan	3.724	44–48	—45	—26	—25	NW— 5.1	48– 8
Alberta	Edmonton	2,183	53–30	—57	—41.3	—40	SW— 7.5	—
B. C.	Vancouver	38	49–18	2	13.1	15	E— 4.5	—
	Victoria	56	48–30	— 2	19.4	15	N— 12.6	—
Man.	Winnipeg	760	49–55	—54	—37.7	—40	NW—10.1	—
N. B.	Fredericton	31	45–57	—35	—25	—25	NW— 9.1	—
N. S.	Yarmouth	31	43–50	—12	— 0.1	0	NW—14.3	—
Ont.	London	804	42–59	—27	—13.7	—10	W— 10.3	—
	Ottawa	214	45–25	—35	—24.1	—20	W— 8.4	—
	Port Arthur	616	48–32	—40	—29.5	—30	NW— 8	—
	Toronto	254	43–40	—26	—11.2	—10	SW—13.6	—
P. E. I.	Charlottetown	9	46–16	—27	—13.2	—15	NW— 9.8	—
Que.	Montreal	63	45–30	—29	—18.4	—15	SW—11.3	—
	Quebec	21	46–50	—34	—23.2	—20	SW—13.3	—
Sask.	Prince Albert	1,414	53–15	—70	—47.3	—45	W— 5.1	—
Yukon	Dawson		64	—68	—54.3	—50	S— 3.7	—

* 1st Fig.—Number of days in 23-yr. period when daily minimum temp was equal to or lower than recommended design temp.

2nd Fig.—Maximum number of days in one heating season, when daily minimum temp was equal to or lower than recommended design temp.

In most cases, this is correlated with the frequency of occurrence of the listed temperature as shown in Column G, which is explained later. It is interesting to note that there is some correlation between average annual minimum temperature and recommended design temperature. Some engineers prefer to use the former for design conditions.

Prevailing winter wind direction and average wind velocity in miles per hour are shown in column F, Table 1.

Column G gives design temperature frequency of occurrence. In this column, two figures are shown. The first is the number of days in a 23-year period when the daily minimum temperatures have been equal to or lower than the recommended design temperature shown in column E. The second figure is the maximum number of times during any one heating season that the daily minimum temperatures have been equal to or lower than the temperature in column E.

The design temperatures recommended show a frequency of occurrence in any one heating season of from two to fourteen times. The design temperature is attained more frequently in the upper central United States. In general, the recommended design temperatures occur about 15 to 40 times in a 23-year period and 5 to 10 times in any one heating season of that period. A comparison of this type permits local designers to adjust their design figures upward or downward, depending on frequency, local experience and type of construction.

Summer Weather Data

Tabular data of maximum recorded temperatures, recommended design dry-bulb and wet-bulb temperatures, prevailing summer wind direction and velocity in miles per hour are presented in **Table 2.**

The necessity for ventilation and control of both temperature and humidity requires knowledge of both dry and wet-bulb design temperatures. In winter it is not customary to use the lowest temperature on record for design purpose, and, similarly in summer, the maximum outside dry-bulb or wet-bulb temperatures are not commonly considered for design.

The question of what summer design temperatures should be used is less controversial than that of winter design conditions. It has been almost universally accepted that the summer design outside dry-bulb temperature for any locality should be a temperature which is exceeded only 2.5 to 5% of the time during normal daylight hours in the summer season. Accepted design wet-bulb temperatures are usually set at a figure which will be exceeded only 5% of the total hours during June to September inclusive. Strict application of these rules for establishing the design temperatures for any locality will result in temperatures expressed in degrees and tenths. It is customary for convenience to round out the figures for wet-bulb temperatures to the nearest degree. Dry-bulb temperatures are rounded out to some convenient figure close to and usually slightly higher than the result obtained by the 5% rule.

The maximum dry-bulb and maximum wet-bulb temperatures seldom occur simultaneously, and the use of off-peak conditions results in loads which are somewhat compensated for by this fact.

Table 2, column A, shows the highest dry bulb on record for representative cities in the United States. Recommended design outside dry-bulb and wet-bulb temperatures are listed in columns B and C, Table 2. Column D shows the prevailing summer wind direction and velocity in miles per hour.

The section of the table listing recommended design temperatures will be found to be consistent with the method previously outlined for establishing these conditions. In most cases the design dry bulb is rounded out to be within the 2.5 to 5% rule.

Summer Water Temperature Data

Water temperature data from various sources for key cities and states are shown in **Table 3.** Most of these data are from tables some of which have been previously published but all of which originated in reports published by the United States Department of Commerce.

Column A, Table 3, gives the expected temperature of water from mechanical draft cooling towers for each state. It is not anticipated that these data will be of

Table 2. Summer Weather Conditions

State	City	A Highest dry bulb on record	B Design dry bulb	C Design wet bulb	D Summer wind direction and velocity
Ala.	Birmingham	107	95	78	S— 5.2
	Mobile	103	95	80	SW— 8.6
Ariz.	Phoenix	118	105	76	W— 6.0
	Tucson	111	105	72	W— 5.0
Ark.	Little Rock	110	95	78	NW— 6.0
Cal.	Los Angeles	109	90	70	SW— 6.0
	San Francisco	101	90	65	SW—11.0
Colo.	Denver	105	95	64	S— 6.8
Conn.	Hartford	101	93	75	S— 6.9
	New Haven	101	95	75	S— 7.3
Del.	Dover	104	93	78	SW— 9.0
	Wilmington	107	95	78	SW— 9.7
D. C.	Washington	106	95	78	S— 6.2
Fla.	Jacksonville	104	95	79	SW— 8.7
	Miami	96	91	79	SE— 8.2
	Tampa	98	94	79	E— 7.0
Ga.	Atlanta	103	95	76	NW— 7.3
	Savannah	105	95	78	SW— 7.8
Idaho	Boise	121	95	65	NW— 5.8
Ill.	Chicago	105	95	75	NE—10.2
	Peoria	113	95	76	S— 8.2
Ind.	Fort Wayne	106	95	75	SW— 8.0
	Indianapolis	106	95	76	SW— 9.0
Iowa	Des Moines	110	95	77	SW— 6.6
	Sioux City	111	96	75	S— 10.1
Kan.	Topeka	114	102	76	S— 10.0
	Wichita	114	100	75	S— 11.0
Ky.	Louisville	107	95	76	SW— 8.0
La.	New Orleans	102	95	79	SW— 7.0
	Shreveport	110	100	78	S— 6.2
Me.	Portland	103	90	73	S— 7.3
Md.	Baltimore	107	95	78	SW— 6.9
Mass.	Boston	104	92	75	SW— 9.2
	Springfield	104	93	75	SW— 9.0
Mich.	Detroit	105	95	75	SW—10.3
	Grand Rapids	108	95	75	W— 7.7
Minn.	Duluth	106	93	73	NE— 10.2
	Minneapolis	108	95	75	SE— 8.4
Miss.	Vicksburg	104	95	78	SW— 6.2
Mo.	Kansas City	113	100	76	S— 9.5
	St. Louis	110	95	78	SW— 9.4
Mont.	Helena	103	95	67	SW— 7.3
Neb.	Omaha	114	95	77	S— 7.7
Nev.	Reno	106	95	65	W— 7.4
N. H.	Concord	102	87	72	NW— 5.4
N. J.	Atlantic City	104	92	77	SW—13.4
	Newark	105	95	75	SW—12.6
	Trenton	106	95	78	SW—10.0
N. Mex.	Santa Fe	97	90	65	SE— 6.5
N. Y.	Albany	104	92	75	S— 7.1
	Buffalo	97	93	75	SW—12.2
	New York	102	95	75	SW—12.9
N. C.	Asheville	97	90	75	SE— 5.6
	Charlotte	103	94	75	SW— 4.8
	Raleigh	103	94	78	SW— 5.9
N. D.	Bismarck	114	5	73	NW— 8.8
Ohio	Cincinnati	108	95	78	SW— 6.6
	Cleveland	100	95	75	S— 9.9
	Columbus	106	95	76	SW— 8.5
	Toledo	105	95	75	SW— 9.7
Okla.	Okla. City	113	101	76	S— 10.1
	Tulsa	115	101	77	S— 10.0
Ore.	Portland	105	90	65	NW— 6.6
Penna.	Philadelphia	106	95	78	SW— 9.7
	Pittsburgh	103	95	75	NW— 9.0
	Scranton	103	95	75	SW— 5.8
R. I.	Providence	100	93	75	NW—10.0
S. C.	Charleston	104	95	80	SW— 9.9
	Greenville	106	95	76	NE— 6.8
S. D.	Sioux Falls	110	95	75	S— 7.6

Table 2. Summer Weather Conditions (Concluded)

State	City	A Highest dry bulb on record	B Design dry bulb	C Design wet bulb	D Summer wind direction and velocity
Tenn.	Chattanooga	104	95	77	SW— 6.5
	Knoxville	104	95	78	SW— 5.9
	Memphis	106	95	78	SW— 7.5
	Nashville	106	95	78	W— 7.5
Tex.	Dallas	110	100	78	S— 9.4
	El Paso	106	100	69	E— 6.9
	Galveston	101	95	80	S— 9.7
	Houston	108	95	78	S— 7.7
	San Antonio	107	100	78	SE— 7.4
Utah	Salt Lake City	105	92	63	SE— 8.2
Vt.	Burlington	100	90	73	S— 8.9
Va.	Norfolk	105	95	78	S— 10.9
	Richmond	107	95	78	SW— 6.2
	Roanoke	105	95	76	
Wash.	Seattle	98	85	65	S— 7.9
	Spokane	108	90	65	SW— 6.5
W. Va.	Charleston	108	95	75	SW— 4.0
Wis.	Milwaukee	105	95	75	S— 10.4
Wyo.	Cheyenne	100	95	65	S— 9.2

Table 3. Maximum Summer Water Temperature Data

(June through September)

State	City	A Mech draft cooling tower	B Surface water	C Well water	D City main
Ala.	Birmingham	87	85	73	
	Mobile		82	—	84
				—	73
Ariz.	Phoenix	78		—	81
	Tucson			80	80
Ark.	Little Rock	86	88	88	88
Calif. (Southern)	Los Angeles	76	76	76	76
	Sacramento		80.6	—	80.6
	San Bernardino		67	67	65
	San Francisco		60	60	62
Colo.	Denver	74	70.9	70.9	70.9
Conn.	Bridgeport	79	66	—	66
	Hartford		70	—	70
	New Haven		70	—	76
Del.	Wilmington		79.0	—	79
D. C.	Washington	83	75.0	—	75
Fla.	Jacksonville	87		86.0	86
	Miami			—	75
	Tampa			—	87
Ga.	Atlanta	86		—	83.5
	Macon			—	80
Idaho		74			
Ill.	Chicago	80	69.4	—	69.4
	Peoria			56.0	56
	Springfield		80.6	—	80.6
Ind.	Evansville	82	85.0	—	85
	Ft. Wayne		79.0	—	79
	Indianapolis		82.0	—	82
Iowa	Cedar Rapids	80	82.5	—	82.5
	Des Moines		72.9	—	72.9
Kan.	Kansas City	82	93.0	—	93
	Topeka		84.0	84.0	84
Ky.	Louisville	82	82	—	82
La.	New Orleans	87	90.0	—	90.0
	Shreveport		91.0	—	91.0
Me.	Augusta	73		—	60
	Portland		66.0	—	66
Md.	Baltimore	82	70.0	—	70
Mass.	Boston	78	75.0	—	75
	Cambridge		74.0	—	74
	Springfield		—	—	55
Mich.	Detroit	76	75.0	—	75
	Grand Rapids		74.0	—	74
	Lansing		—	59.0	59
	Saginaw		78.0	—	78

Table 3. Maximum Summer Water Temperature Data (Concluded)

State	City	A Mech draft cooling tower	B Surface water	C Well water	D City main
Minn.	Duluth	76	70.6	—	70.6
	Minneapolis		80.2	—	80.2
Miss.	Jackson	87	82.0	—	82
Mo.	Jefferson City	82	80.0	—	80
	Springfield		76.0	76.0	76
	St. Louis		85.0	—	85
Mont.		73			
Neb.	Lincoln	80		59.0	59
	Omaha		80.9	—	80.9
N. H.	Concord	78	73.0	—	
	Manchester			—	76
N. J.	Atlantic City	81	72.5	72.5	72.5
	Newark		71.5	—	71.5
	Trenton			—	79
N. M.	Albuquerque	80	—	72.0	72.0
N. Y.	Albany	78	66.0	—	66
	Buffalo		73.0	—	73
	New York		70.0	—	70
	Rochester		68.9	—	68.9
N. C.	Asheville	85	76.0	—	76
	Charlotte		81.0	—	81
	Raleigh		83.4	—	83.4
N. D.	Fargo	76	85.1	—	
Ohio	Cincinnati	80	82.0	—	82
	Cleveland		73.5	—	73.5
	Columbus		76.0	—	76
	Toledo		87.0	—	87
Okla.	Okla. City	83	77.2	—	77.2
	Tulsa		81.8	—	81.8
Ore.	Portland	73	62.0	—	62
Penna.	Philadelphia	80	79.0	—	79
	Pittsburgh		80.6	—	80.6
	Scranton		70.9	—	71
R. I.	Providence	78	65.0	—	65
S. C.	Charleston	86	85.0	—	85
	Greenville		76.0	—	76
S. D.	Rapid City	76		—	55
	Sioux Falls		—	55	55
Tenn.	Chattanooga	85	79.0	—	79
	Knoxville		84.3	—	84.3
	Memphis				70
	Nashville		88.0	—	88
Tex.	Dallas	83	82.5	82.5	82.5
	El Paso		85.0	85.0	85
	Houston		84.0	—	84
	San Antonio		76.0	—	76
Utah	Salt Lake City	68	58.0	58.0	58
Vt.	Burlington	76	68.0	—	68
Va.	Lynchburg	84	73.0	—	73
	Norfolk		83.0	—	83
	Richmond		80.4	—	80.4
Wash.	Seattle	70	—	—	62
	Spokane		—	—	47
W. Va.	Charleston	81	83.0		83
Wis.	Madison	78	—	53.0	53.0
	Milwaukee		60.6	—	60.6
Wyo.	Cheyenne	73			

great value, since conditions may vary widely over any one state and the cooling tower design will have a decided influence. It is recommended that water temperatures from mechanical draft cooling towers be predicated on the basis of design wet-bulb temperatures and cooling tower efficiencies.

Surface water, well water, and city main temperatures for many cities are listed in columns B, C, and D, Table 3. In many instances where city main temperatures are shown, there is no indication as to whether the water is from wells or from the surface. Where the source has been shown as from both, the temperatures are listed in all three columns.

These water temperatures are based on the highest monthly average occurring in any of the four months, June to September inclusive, and represent only one or two years' record. Most of these data were obtained and compiled originally in the period from 1930 to 1935, but past experience has shown that they do not change to any great degree from year to year. For

application of self-contained air-conditioning units or commercial condensing units, cooled with city water, the temperatures listed in column D of Table 3 will be satisfactory for purposes of estimating.

In any locality, application engineers concerned with local conditions may check local weather bureau records for information on dry and wet-bulb temperatures. Water temperature data may be obtained from municipal engineers' offices, utility company or industrial plant records.

For other weather data not shown in the tables in this chapter, various sources are recommended. The United States Yearbook of Agriculture, 1941 Edition, contains information on maximum and minimum dry-bulb temperatures, rainfall statistics and dates of first and last killing frosts. Probably one of the most complete summaries of weather information useful in design of summer air conditioning equipment is "Summer Weather Data" published by the Marley Company, Kansas City, Kan. This book contains analytical data on a nationwide basis of dry-bulb temperatures, wet-bulb temperatures, dew points, wind and sunshine for the summer period of June to September inclusive, and much information on conditions at other times of the year. Material in this book has been drawn upon freely in the preparation of this chapter.

The frequency of occurrence of recommended design temperatures for winter is based upon a study, the results of which were published in 1939 by the Carnegie Institute of Technology—Bulletin No. 82, "An Analysis of Winter Temperatures for One Hundred and Twenty Cities," by Clark M. Humphreys.

Summer water temperature data are from several sources, including a table appearing in earlier editions of the *Refrigerating Data Book*, tables in "Heating and Ventilating Data Sheets" which were originally published by the Department of Commerce, and a few local sources.

12. PHYSICAL PROPERTIES OF MATERIALS, INCLUDING INSULATION

Table 1. Gases and Vapors

Name or description	Formula	Sp ht C_p		Density at 1 atm and 32F	Mol wt	Boiling point F	Thermal conductivity		Critical temp F	Pressure psia
		Btu/lb	Temp, F	lb/cu ft			F	K		
Air		.2377	−22 to +50	.0808	28.97	−317	−148	.0095	221	910
							32	.0140		
Air		.2375	32 to 400				212	.0183		
Alcohol	C_2H_5OH	.4534	110 to 400							
Alcohol	CH_3OH	.4580	110 to 400							
Ammonia	NH_3	.5202	73 to 212	.0482	17.03	− 28	− 76	.0095	270	1,638
							+ 32	.0128		
Argon	A	.1233	68 to 195	.1114	39.91	−302			−187	705
Acetylene	C_2H_2			.0732	26.02					
Benzene	C_6H_6	.2990	94 to 235	.206	78.11	176.18	32	.0052	551	700
Bromine	Br_2	.0555	181.4 to 442.4	.4460	159.83					
Butane	C_4H_{10}			.1623	58.08	31.1	32	.0078	307	528
Benzol				.233						
Butylene	C_4H_8			.148	56.06					
Carbon dioxide	CO_2	.1843	−20 to 45			−164.2	− 58	.0068		
							32	.0085		
Carbon dioxide	CO_2	.2025	59 to 212	.1235	44.0				88	1,072
Carbon monoxide	CO	.2425	74 to 210	.0781	28.0	−311.8	32	.0135	−218	514
Chlorine	Cl_2	.1125	61 to 700	.2011	70.91	− 22	32	.0043	291	1,118
Chloroform	$CHCl_3$	.1441	80 to 230		119.39	142.16	32	.0038		
Carbon disulfide	CS_2				76.1	115			523	1,116
Carbon tetrachloride	CCl_4				153.8	170			541	661
Dichlorodifluoromethane (F_{12})	CCl_2F_2	.143	0 to 100		120.92	− 21.64	32	.0048		
Dichloromethane	CH_2Cl_2				84.9	105			421	1,490
Ether	$C_4H_{10}O$	.4797	156.2 to 435.2		74.12	34	− 94	.0066	90	717
Ethane	C_2H_6			.0848	30.05	−126.94	− 29	.0086		
							32	.0106		
Ethyl chloride	C_2H_5Cl			.1793	64.5	−155.02	32	.0055	370	764
Ethylene	C_2H_4			.0783	28.03	−155	− 96	.0064	50	747
							32	.0101		
Ethyl ether	$C_4H_{10}O$	.4280	77 to 232				32	.0077		
Fluorine	F_2			.1022	38.00	−304.6				
Freon-12	CCl_2F_2	.143	1 to 100		120.9	− 21	207	.057	233	580
Helium	He			.011	4.00	−452			−450	33
Hydrochloric acid	HCl	.1940	55.4 to 212	.1024	36.47					
Hydrogen	H_2	3.3996	−2.2 to +118.4	.0056	2.016	−421.6	− 58	.083	−400	188
							32	.100		
Hydrogen sulfide	H_2S	.2451	68 to 402.8	.096	34.08	139.28	32	.0076	212	1,306
Hydrogen chloride	HCl				36.4	−121			124	1,198
Heptane	C_7H_{16}			.301	100.2	209.08	392	.0112		
Hexane	C_6H_{14}			.2553	86.17	155.66	32	.0072	454	433
Hexylene	C_6H_{12}				84.09					
Isobutane	C_4H_{10}			.1669	58.07	14	32	.0080	273	543
Isopentane	C_5H_{12}				72.09					
Methyl chloride	CH_3Cl			.1440	50.48	− 10.66	32	.0053	289	966
Methane	CH_4	.5929	66 to 390	.0448	16.03	−257.8			−116	672
Neon	Ne			.0544	20.40	−320.44			−380	389
Nitrogen		.2438	32 to 390				32	.0140		
Nitrous oxide	N_2O	.2126	80 to 220	.1235	44.02	−129.1	− 98	.0067		
							32	.0087	98	1,053
Nitrogen tetroxide	NO_2	1.625	80 to 150							
Naphthalene	$C_{10}H_8$				128.0					
Natural gas					19.46					
Oxygen	O_2	.2175	50 to 400	.0892	32.00	−297.4	− 58	.0119	−182	730
							32	.0142		
Propane	C_3H_8			.126	44.09	− 44.14	32	.0087	204	632
Pentane	C_5H_{12}			.211	72.15	96.8	32	.0074	387	485
Phenol	C_6H_5				94.0					
Propylene	C_3H_6				42.0	− 52			198	661
Refinery gas					1.20					
Sulfur dioxide	SO_2	.1544	16 to 202	.1828	64.06	+ 14	32	.0050	315	1,141
Water vapor	H_2O	.46	100		18.01	212	115	.0120	706	3,226
		.49	210				212	.0137		

Note: K=Btu/hr sq ft F/ft, however lower case k is k=Btu/hr sq ft F/in.

Table 2. Liquids

Name or description	Formula	Boiling point F	Enthalpy of vaporization Btu/lb	Sp ht		Viscosity		Freezing point F	Enthalpy of fusion Btu/lb	Sp gr or density (ρ)		Thermal conductivity, K		Vapor pressure	
				Btu/lb	Temp F	Centipoises	Temp F			Temp F	lb/cu ft ρ	Btu ft per hr sq ft F	Temp F	mm of Hg	Temp F
Acetaldehyde	C_2H_4O	69.44	244.8			.275	32	−190.3						400	42.8
Acetic acid	$C_2H_4O_2$	245.3	173.	.522	78.8–203	1.040	86	62.00	77.7			.099	68	200	176
Acetone	C_3H_6O	132.98	223.	.506	32			203	42.1	68	49.4 (ρ)	.102	77–86	185	68
Allyl alcohol	C_3H_6O	206.6	293.	.665	69.8–204.8	1.168	86	−200.2				.104	77–86		
Amyl alcohol	$C_5H_{12}O$	280.22	216.					−109.3	48.			.094	86	100	177.8
Ammonia	NH_3			1.099	32				94.5			.29	5–86		
Alcohol-ethyl	C_2H_6O	172.94	366.9	.548	0	1.2	68	−174.28	45.6			.789		44	68
Alcohol-methyl	CH_4O	148.46	472.	.601	64	.596	68	−142.6	39.6			.796		94	68
Aniline	$C_6H_5NH_2$	363.2	198.	.514	60	4.467	68	+ 20.77	48.6	32	64.5 (ρ)			10	156.2
Benzol	C_6H_6			.340	50	.567	86		55.						
Benzene	C_6H_6	176.18	167.						54.6	32	56.1 (ρ)	.092	86	75	68
Bromine	Br	137.84	86.4	.107	13–45	.911	86		29.15	32					
Butyl-alcohol	$C_4H_{10}O$	243.86	254.	.687	20–115			−129.64	54.			.097	86		
Butyric acid	$C_4H_8O_2$	326.3	205.	.515	20–100	1.304	86	+ 22.01	54.2					50	197.6
Calcium chloride brine	$CaCl_2$			.764	5					5	1.14				
				.787	68					68	1.14	30% .32	86		
				.695	+4					+4	1.20	15% .34	86		
				.725	68					68	1.20				
				.651	−4					−4	1.26				
				.676	68					68	1.26				
Carbolic acid	C_6H_6O	359.96		.561	57.2–78.8				52.2	59	60.2 (ρ)				
Carbon disulfide	CS_2	115.27	151.3	.240	68	.352	86	−169.24		32	80.6 (ρ)	.093	86	100	+23
Carbon tetrachloride	CCl_4	169.16	83.5	.201	68	.848	86	− 9.04	74.8			.107	32	90	68
Chloroform	$CHCl_3$	142.16	106.	.226	59	.519	86	− 82.3			1.50	.080	86	160	68
Copper sulfate	$CuSO_4$														
	+ 50 H_2O			.848	58										
	+100 H_2O			.951	57										
	+400 H_2O			.975	59										
Diphenylamine	$C_{12}H_{11}N$	575.6		.464	125				127.36					10	314.6
Decane	$C_{10}H_{22}$	345.2	108.2	.500	0–50	.77	72.14	− 22	86.4			.085	86	10	132.8
Dow Corning 500		211.1													
		305.6													
		377.6													
		446.0													
Ethyl ether	$C_4H_{10}O$	94.08	159.	.529	32	.223	86	−176.8	41.4		.736	.08	86	442	68
Ethyl acetate	$C_4H_8O_2$	170.78	183.5 (32 F)	.475	68			−118.84	51.1			.101	68	72.8	68
Ethyl alcohol	C_2H_6O	172.94	367.	.548	32				44.8			.105	68	50	72.86
Ethyl bromide	C_2H_5Br	101.12	108.	.215	59–68	.368	86	−182.2						386	68
Ethyl chloride	C_2H_5Cl	53.96	166.5	.367	32			−217.66						1,000	68
Ethyl iodide	C_2H_5I	161.78	81.3	.161	68	.540	86	−163.3				.064	104	108	68
Ethylene bromide	$C_2H_4Br_2$	269.06	83.	.173	68	1.475	86	50.108						10	72.86
Ethylene chloride	$C_2H_4Cl_2$	182.66	139.	.299	68	.736	86	− 31						100	85.46
Ethylene glycol	$C_2H_6O_2$	386.6	344.									.153	32	10	192.56
Formic acid	CH_2O_2	213.44	216.	.525	68–212	1.46	86	47.12	104.4					50	84.2
Glycerine	$C_3H_8O_3$	554		.575	59–120	830.	68.54	64.58	85.6					385	−184
Glycerol	$C_3H_8O_3$	555.08				207.	68	64.4	85.5			.164	68		

Gasoline		158–194		.5	32–212						.73				
Heptane	C_7H_{16}	209.12	137.1	.490	68	.375	86	−131.08	60.6			.081	86	50	80.96
Hexane	C_6H_{14}	755.6	142.5	.600	68	.296	86	−139	65.			.080	86	100	61.16
Hydrochloric acid	HCl													100	140.18
Isobutyl alcohol	$C_4H_{10}O$	224.42	248.												
Kerosene				.5	32–212						.78–.82	.086	68		
Linseed oil						33.1	86				.925				
Methyl acetate	$C_4H_8O_2$	134.78	176.5	.468	59			−144.49						200	74.84
Methyl iodide	CH_3I	108.14	82.6			.460	86	− 86.98						400	76.64
Naphthalene	$C_{10}H_8$	424.4	136.	.396	185			176.396	64.					10	176
Nitrobenzole				.350	58										
Nitric acid	HNO_3	186.8						− 43.6	17.15		91%—1.50				
Nitrobenzene	CH_5NO_2	411.62	142.2	.350	57.2			42.53	40.5			.095	86	10	185.9
Nonane	$CH_3(CH_2)_7CH_3$	302		.503	32–122	.62	72.14	− 64.66				.084	86	100	199.94
Oils															
Castor				.434		451.	86			59	60.5 (ρ)	.104	68		
Citron				.438	42										
Olive				.471	44	54.	86			59	.906	.097	68		
Sesame				.387											
Rapeseed						42.2	100.04			59	57. (ρ)				
Soybean						40.6	86			194	.919				
Sperm						42.	60.08			77	55. (ρ)				
Octane	$CH_3(CH_2)_6CH_3$	256.28	127.5	.587	68–253.4	.483	86	− 70.42				.083	86	10	66.2
Petroleum				.511	70–135						.87				
Pentane	C_5H_{12}	96.8				.220	86	−201.82		32	40.6 (ρ)			400	65.84
Propionic acid	$C_3H_6O_2$	285.98	177.8	.560	68–278.6	.960	86	− 5.44						10	104
Propyl alcohol	C_3H_8O	207.5	295.2	.57	68	1.779	86	−194.98			2.04			10	57.56
Potassium hydroxide	2 KOH														
	$KOH + 30\ H_2O$			.876	64										
	$KOH + 100\ H_2O$			.975	64										
Sea water				.980							1.004				
				.938							1.024				
				.903							1.046				
Sulfuric acid 100%	H_2SO_4	626	219.6	.344	68			50.882	43.2		87%—1.80	.21	86		
Sodium chloride brine	NaCl														
	$+ 10\ H_2O$			.791	64						25%—.33		86		
	$+200\ H_2O$			.978	64						13%—.34		86		
Sodium hydroxide	$NaOH + 50\ H_2O$			.942	64										
	$NaOH + 100\ H_2O$			.983	64										
Toluol	C_6H_6	231.8	154.8	.364	50										
				.534	185										
Toluene	C_7H_8	230.54	155.7	.440	53.6–210.2	.525	86	−139							
Turpentine	$C_{10}H_{16}$	320	123.4	.411	32	1.272	86				.864	.074	59	.4	68
Water	H_2O	212	969.7	1.000	60.8	.8007	86	32	143.05	39.2	1.00	.330	32	10	52.358
										39.2	62.4 (ρ)	.356	86	100	125.06
Xylene	$C_6H_4(CH_3)_2$	287.6	149.2	.411	86			− 16.78				.090	68		
Zinc sulfate	$ZNSO_4$														
	$+ 50\ H_2O$			.842	68 to 125										
	$+200\ H_2O$			.952	125										

Table 3. Solids

Name or description	Sp ht		Sp gr	Conductivity Btu/hr sq ft F/ft		Emissivity	
	Btu/lb F	Temp F		Temp F	*K*	Temp F	Ratio
Aluminum	.226	100	2.55–2.80	64	116.0	73	.040
Aluminum bronze			7.7				
Alundum	.186	212					
Asbestos	.25	.47–.58	2.1–2.8	32	.09	100–700	.93–.95
Asphalt	.3–.4						
Ashes	.20		.64–.72	32	.041		
Bakelite	.3–.4						
Brickwork	.2		1.85–2.00	70	.33–.92	70	.93
Brass, red	.08991	32	8.4–8.7	32	59.5	73	.043
Brass, yellow	.08831	32	8.4–8.7	32	49.4		.043
Bismuth tin	.040			64	37.6		
Bell metal	.086	59–208.4					
Bronze	.104		7.4–8.8				
Cadmium	.0548		8.65	64	53.7		
Carbon (gas retort)	.204					68	.927–.967
Cardboard					.1–.2		
Cellulose	.32						
Cement, Portland clinker	.186		1.5–2.4		.017		
Charcoal (wood)	.242		.28–.57	172	.051		
Chrome brick	.17						
Clay	.224		1.28				
Coal	.26–.37		.65–1.8				
Coal tars	.35	104					
Coal tar oils	.34	59–194					
Coke	.265	69.8–752	1.0–1.4	32	.106		
Concrete (stone)	.156	70–213	1.5–2.4		.5–.75		
Copper (cast rolled)			8.8–8.9	64	222.0	66	.030
Cryolite	.253	60.8–131					
Chalk	.215		1.8–2.8		.48	145–380	.32–.36
Cork (granulated rolled)	.485		.22–.26	24	.028		
Cotton (flax, hemp)			1.47–1.50	32	.033		
Cotton (wool)					.01		
Diamond	.147						
Earth (dry & packed)			1.5 loose	32	.035	145	.39–.43
				100	.039		
Felt				86	.022		
Fireclay brick	.198	212					
Fluorspar	.21	86					
Glass (crown)	.16–.2		2.4–2.7		.333–.5	72	.937
(flint)	.117		3.2–4.7			72	.937
(pyrex)	.20					72	.937
(silicate)	.188–.2	32–212				72	.937
(wool)	.157						
(common)			2.40–2.80			72	.937
Graphite (powder)	.165	78.8–168.8		104	.106		
Graphite	.20	68–212	2.4–2.7		1.0–2.32	145–380	.40–.56
Gypsum	.259	60.8–114.8	2.3–2.8	68	.25	70	.903
German silver	.0946	32–212	8.58				
Garnet	.1758	60.8–212					
Gold	.0308		19.25–19.2	64	169.0		.05
Ice	.350	−112	.88–.92	32	1.28 (water)	32	.95
Ice	.434	− 40		14	1.35		
Ice	.465	− 4		− 4	1.41		
Ice	.487	32		−22	1.471		
Ice				−40	1.538		
India rubber (para)	.481	−148					
Iron (gray cast)	.101		7.03–7.13	129	27.6		

Table 3. Solids (Concluded)

Name or description	Sp ht		Sp gr	Conductivity Btu/hr sq ft F/ft		Emissivity	
	Btu/lb F	Temp F		Temp F	K	Temp F	Ratio
Iron (cast pig)			7.2				
Iron (wrought)			7.6–7.9	64	34.9	70–680	.94
Lead	.030		11.34	64	20.1	75	.281
Limestone	.217	59–212	2.1–2.8		.3–.75	145–380	.36–.90
Litharge	.055						
Leather			.86–1.02		.092		
Linen					.05		
Marble	.21	64.4	2.4–2.8		1.2–1.7	72	.931
Manganese			7.42				
Magnesia	.234	212			.04		
Magnesite brick	.222	212			.9–2.5		
Monel metal	.127	68–2372	8.97				
Mica	.10	68			.44		
Nickel	.103		8.9	64	34.4	74	.045
Nickel steel	.109						
Paper	.324		.70–1.15		.075	66	.924
Paraffin	.6939	32–68	.87–.91	86	.145		
Platinum (cast)			21.5	64	40.2		
Porcelain	.22			329	.945	72	.924
Pyrites (copper)	.131	66.2–122					
Pyrites (iron)	.136	59–208.4					
Paint (lacquers, varnish)							
White lacquer						100–200	.80–.95
White enamel						73	.906
Black lacquer						76	.875
Black shellac						70	.821
Aluminum paint						212	.3–.5
Plaster (rough lime)					.25–.05	50–190	.91
Sawdust			.21	68	.042		
Rock salt	.219	55.4–113					
Rubber (goods)	.48		1.0–2.0	100	.92	76	.859
Salt peter			1.07				
Sand	.191		1.4–1.9	68	.188		
Silica	.316						
Steel (cold-drawn)	.12		7.83	64	26.2	75	.82
Stone	.2						
Silver (cast)			10.4–10.6	64	244.0	75	.82
Snow (fresh fallen)			.125				
Tin (cast)	.053		7.2–7.5	64	37.6	76	.043–.064
Tungsten	.034		19.22				
Tar (bituminous)			1.20				
Wood (oak)	.570		.65–.84		.085–.125	70	.895
most woods vary between	.45–.65						
Ash			.55–.71				
Fir	.65		.40	86	.094		
Elm			.56				
Hickory			.74–.80				
Mahogany			.56–.85				
Maple			.53–.68	86	.092		
Pine	.67		.43–.67	86	.065–.085		
Spruce			.45				
Walnut			.59				
Wool			1.32	86	.022		
Zinc (cast)			7.1				.10
Zinc (galvanized)						75	.22–.28

Note: The emissivity of a surface varies with its temperature, its degree of roughness, and, if a metal, its degree of oxidation. Therefore it must be remembered, emissivity for surfaces whose degree of roughness and oxidation is not specified is only an approximate value, and a wide variation is possible.

Table 4. Insulating Materials

Conductivity, k, is in Btu per hr sq ft deg F per inch of thickness. Conductance, c, is in Btu per hr sq ft deg F for the thickness specified.

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance k or c
BLANKET & BATT INSULATIONS	Asphalt-glass wool pad	17.7	90	.287
		17.7	60	.273
		17.7	30	.256
		17.7	0	.241
		17.7	−30	.226
		17.7	−60	.211
	Chemically treated wood fibers held between layers of strong paper	3.62	70	.25
	Aerated wood fiber mat	1.7	90	.272
		1.7	60	.252
		1.7	30	.236
		1.7	0	.219
		1.7	−30	.203
		1.7	−60	.187
	Eelgrass between strong paper	4.60	90	.26
		3.40	90	.25
	Flax fibers between strong paper	4.90	90	.28
	Chemically treated mohair between Kraft paper & asbestos paper	5.76	71	.26
	Chemically treated hog hair between Kraft paper & asbestos paper	7.70	71	.28
	Hair felt between layers of paper	11.00	75	.25
	Kapok between burlap or paper	1.00	90	.24
	Stitched & creod expanding fibrous blanket	1.50	70	.27
	Paper & asbestos fiber with emulsified asphalt & binder	4.2	94	.28
	Cotton insulating batt	.875	72	.24
	Felted cattle hair	13.00	90	.26
		11.00	90	.26
	Hair felt $\frac{3}{4}$ in. thick	10.9	90	.265
		10.9	60	.250
		10.9	30	.236
		10.9	0	.223
		10.9	−30	.210
		10.9	−60	.199
	Felted hair & asbestos	7.8	90	.28
	Ground paper between two layers, each $\frac{3}{8}$ in. thick made up of two layers of Kraft paper (sample $\frac{3}{4}$ in. thick)	12.1	75	.40
	Granulated mineral wool	6	85	.26
	Mineral wool felt	5.9	58.6	.224
		5.9	106.0	.252
		8.5	59.3	.216
		8.5	107.2	.238
	Kold board	16.2	32.0	.283
		16.2	89.9	.318
	Wood fiber blanket	3.5	90	.266
		3.5	60	.250
		3.5	30	.236
		3.5	0	.223
		3.5	−30	.211
		3.5	−60	.198
LOOSE FILL TYPE	Made from ceiba fibers	1.90	75	.23
	Cotton fibers	1.60	75	.24
	Short staple linters fireproofed	6.25	90	.25

Table 4. Insulating Materials (Continued)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance k or c
LOOSE FILL TYPE	Cotton	4.50	90	.24
		2.65	90	.246
		2.65	60	.253
		2.65	30	.221
		2.65	0	.208
		2.65	−30	.195
		2.65	−60	.183
	Cotton fibers Short staple linters fireproofed	2.45	90	.24
		1.60	90	.26
	Cotton	1.53	90	.247
		1.53	60	.230
		1.53	30	.216
		1.53	0	.202
		1.53	−30	.188
		1.53	−60	.176
		0.95	90	.266
		0.95	60	.245
		0.95	30	.227
		0.95	0	.208
		0.95	−30	.191
		0.95	−60	.173
		.65	90	.30
	Excelsior	1.84	90	.343
		1.84	60	.313
		1.84	30	.286
		1.84	0	.261
		1.84	−30	.237
		1.84	−60	.213
	Fibrous material made from dolomite & silica	1.5	75	.27
	Fibrous material made from slag	9.40	103	.27
	Redwood bark	3.0	90	.31
		4.0	90	.282
		4.0	60	.263
		4.0	30	.247
		4.0	0	.229
		4.0	−30	.213
		4.0	−60	.198
	Redwood bark	5.0	75	.26
	Glass wool fiber .0003 to .0060 in. in diameter	1.5	75	.27
	Glass wool	1.65	90	.303
		1.65	60	.272
		1.65	30	.247
		1.65	0	.225
		1.65	−30	.203
		1.65	−60	.183
	Granular insulation made from combined silicate of lime & alumina	4.2	72	.24
	Expanded vermiculite			.48
	Expanded vermiculite (particle size—3×14)	6.2		.32
	Regranulated cork $\frac{3}{16}$-in. particles	8.10	90	.31
	Hand applied granular mineral wool 2 in. to 6 in. thick, horizontal position, no covering	6.05 to 7.13		.30 to .33
	Mineral wool	3.5	90	0.287
		3.5	60	0.271
		3.5	30	0.238
		3.5	0	0.216

Table 4. Insulating Materials (Continued)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance k or c
LOOSE FILL TYPE		3.5	−30	0.198
		3.5	−60	0.186
	4-in. machine blown granular mineral wool, horizontal position, no covering	5.74		.30
	Rockwool	10.0	90	.27
	Nodulated mineral wool 1½ in. thick	10.0	90	.289
		10.0	60	.272
		10.0	30	.256
		10.0	0	.242
		10.0	−30	.227
		10.0	−60	.213
	Silica aerogel	7.6	90	.169
		7.6	60	.160
		7.6	30	.152
		7.6	0	.145
		7.6	−30	.138
		7.6	−60	.133
	Expanded vermiculite, fill	8.2	90	.476
		8.2	60	.457
		8.2	30	.438
		8.2	0	.420
		8.2	−30	.401
		8.2	−60	.383
INSULATING BOARD	Made from sugar cane fiber	13.5	70	.33
	Made from corn stalks	15.0	71	.33
	Made from exploded wood fiber	17.9	78	.32
	Made from hard wood fiber	15.2	70	.32
	Made from wood fiber	15.9	72	.33
		15.00	70	.33
			52	.33
		8.5	72	.29
		15.20		.33
		16.90	90	.34
	Made from licorice root	16.1	81	.34
	½-in. insulating board without special finish	16.5	90	.33
		to		to
	Eleven samples	21.8		.40
	1-in. insulating board	15.2		.34
	Corkboard (average) 1.12 in. thick	7.07	−14	.233
	Corkboard (average) 1.0 in. thick	7.07	0	.27
		7.07	−108	.201
	Corkboard (average) 1.04 in. thick	6.70	−21	.242
		6.70	−115	.204
	Glass fiber board 1.00 in. thick	11.0	−16	.213
		11.0	−111	.168
	Cellular glass block	10.3	90	0.422
		10.3	60	0.404
		10.3	30	0.387
		10.3	0	0.371
		10.3	−30	0.353
		10.3	−60	0.337
	Expanded glass block 1.07 in. thick	10.6	−22	.409
		10.6	−110	.381
	Vegetable fiber board 1.02 in. thick	14.4	−24	.284
		14.4	−110	.248
	Expanded rubber board 1.00 in. thick	4.9	−22	.206
		4.9	−100	.173

Table 4. Insulating Materials (Continued)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance k	or c
INSULATING BOARD	Shredded redwood bark 1.00 in. thick	4.0	−25	.208	
		4.0	−119	.172	
	Mineral wool, board	15.7	90	.322	
		15.7	60	.308	
		15.7	30	.294	
		15.7	0	.281	
		15.7	−30	.268	
		15.7	−60	.254	
	Mineral board .99 in. thick wool	14.3	−21	.208	
		14.3	−112	.189	
	Vermiculite board	18.9	90	.927	
		18.9	60	.908	
		18.9	30	.889	
		18.9	0	.870	
		18.9	−30	.852	
		18.9	−60	.833	
	Wood fiber board $\frac{3}{4}$ in. thick, 24.1% moisture	15.7	90		.424
		15.7	60		.408
		15.7	30		.392
		15.7	0		.376
		15.7	−30		.358
		15.7	−60		.343
	Wood fiber board $\frac{3}{4}$ in. thick, 3.1% moisture	15.7	90		.356
		15.7	60		.343
		15.7	30		.331
		15.7	0		.319
		15.7	−30		.306
		15.7	−60		.294
	Wood fiber board $\frac{3}{4}$ in. thick, 0.0% moisture	15.7	90		.348
		15.7	60		.335
		15.7	30		.323
		15.7	0		.310
		15.7	−30		.298
		15.7	−60		.285
SLAB INSULATIONS	Corkboard (no added binder)	14.0	90	.34	
		10.6	90	.30	
		7.0	90	.27	
		5.4	90	.25	
	Corkboard	8.7	—	.29	
	Corkboard (oven dry)	12.2	90	.309	
		12.2	60	.300	
		12.2	30	.292	
		12.2	0	.282	
		12.2	−30	.273	
		12.2	−60	.264	
		8.0	90	.276	
		8.0	60	.267	
		8.0	30	.258	
		8.0	0	.249	
		8.0	−30	.240	
		8.0	−60	.231	
		6.5	90	.277	
		6.5	60	.266	
		6.5	30	.255	
		6.5	0	.244	
		6.5	−30	.233	
		6.5	−60	.222	

Table 4. Insulating Materials (Continued)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance k or c
SLAB INSULATIONS	Corkboard asphaltic binder	14.5	90	.32
	Chemically treated hog hair with film of asphalt	10.0	75	.28
	Sugar cane fiber insulating blocks encased in asphalt membrane	13.8	70	.30
	Made from 85% magnesia and 15% asbestos	19.3	86	.51
	Made from shredded wood & cement	24.2	72	.46
		29.8		.77
BUILDING BOARDS (Non-insulating)	Compressed cement & asbestos sheets	123	86	2.70
	Corrugated asbestos board	20.4	110	0.48
	Lignin-impregnated cellulose fiber board $1\frac{1}{4}$ in. thick	85.5	90	1.84
		85.5	60	1.81
		85.5	30	1.77
		85.5	0	1.74
		85.5	−30	1.71
		85.5	−60	1.67
	Pressed asbestos mill board	60.5	86	.84
	Gypsum board—gypsum between paper	62.8	70	1.41
	$\frac{3}{8}$-in. gypsum board			3.73
	$\frac{1}{2}$-in. gypsum board			2.82
	$\frac{1}{2}$-in. gypsum board	53.5	90	2.60
CLAY TILE, HOLLOW	2-in. tile, $\frac{1}{2}$-in. plaster both sides	120.0	110	1.00
	4-in. tile, $\frac{1}{2}$-in. plaster both sides	127.0	100	.60
	6-in. tile, $\frac{1}{2}$-in. plaster both sides	124.3	105	.47
	8-in. tile, average of 8 types			.52
	12-in. clay tile wall			
	8×5×12			
	4×5×12			.26
CONCRETE	Sand & gravel aggregate, various ages and mixes			11.35 to 16.36
	Sand & gravel aggregate	142	75	12.6
	Limestone aggregate	132	75	10.8
	Cinder aggregate	97	75	4.9
	Steam treated limestone slag aggregate	74.6	75	2.27
	Pumice aggregate	65.0	75	2.42
	Expanded burned clay aggregate	59.9	75	2.28
	Burned clay aggregate	67.1	75	2.86
	Blast furnace slag aggregate	76.0	70	1.6
	Expanded vermiculite aggregate	20	90	.68
		26.7	90	.76
		35	90	.86
		50	90	1.10
	Concrete plank	76	75	2.5
	Cellular concrete	40.0	75	1.06
		50	75	1.44
		60	75	1.80
		70	75	2.18
CONCRETE BLOCKS 8-in. concrete blocks 8×8×16 3 oval	8-in. three oval core sand & gravel aggregate	126.4	40	.90
	8-in. three oval core crushed limestone	134.3	40	.86
	8-in. three oval core burned clay aggregate	67.7	40	.50
	8-in. three oval core cinder aggregate	86.2	40	.58
	8-in. three oval core expanded blast fu nace aggregate		40	.40

Table 4. Insulating Materials (Continued)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance k	or c
12-in. concrete blocks 8×12×16 3 oval	12-in. three oval core sand & gravel aggregate	124.9	40		.78
	12-in. three oval core cinder aggregate	86.2	40		.53
	12-in. three oval core burned clay	76.7	40		.47
RAME CONSTRUCTION COMBINATIONS	1-in. fir sheathing and building paper		30		.86
	1-in. fir sheathing building paper and yellow pine siding		20		.50
	1-in. fir sheathing building paper and stucco		20		.82
	Pine lap siding and building paper, siding 4 in. wide		16		.85
	Yellow pine lap siding				1.28
YPSUM	3-in. solid gypsum partitioned tile			2.41	
	3-in. three cell gypsum partitioned tile				.74
	4-in. three cell gypsum partitioned tile				.60
	87½%, gypsum, 12½% wood chips	51.2	74	1.66	
	Gypsum plaster			3.30	
IASONRY MATERIALS BRICK	Damp or wet			5.0	
	Common yellow clay brick			4.8	
	One tier yellow common clay brick, one-tier face brick, approx. 8 in. thick				.77
LASTERING MATERIALS	Gypsum plaster ⅜ in. thick		73		8.8
	Cement plaster			12.00	
	Wood lath & plaster, total thickness ¾ in.		70		2.50
	Gypsum plaster and expanded vermiculite, 4 to 1 mix	39.9	75	.85	
	Insulating plaster 0.9 in. thick applied to ⅜-in. gypsum board	54.0	75		1.07
OOFING	Asbestos shingles	65.0	75		6.0
	Asphalt composition or prepared	70	75		6.5
	Asphalt shingles	70	75		6.5
	Built up roofing, bitumen or felt, gravel or slag surfaced			1.33	
	Slate			10.0	
	Wood shingles			1.28	
VOODS	Balsa	20.0	90	.58	
		8.8	90	.38	
		7.3	90	.33	
	California redwood, 0% moisture	28.0	75	.70	
	Cypress	28.7	86	.67	
	Douglas fir, 0% moisture	34	75	.67	
	Eastern hemlock, 0% moisture	30	75	.76	
	Fir plywood, ¼ in. thick	33.2	90		.813
		33.2	60		.791
		33.2	30		.768
		33.2	0		.745
		33.2	−30		.723
		33.2	−60		.702
	Fir plywood, four ¼ in. thick, 17.3% moisture	31.9	90		.896
		31.9	60		.873
		31.9	30		.847
		31.9	0		.823

Table 4. Insulating Materials (Concluded)

Material	Description	Density lb/cu ft	Mean temp F	Conductivity or conductance	
				k or c	
WOODS		31.9	−30		.800
		31.9	−60		.776
	Fir plywood, four $\frac{1}{4}$ in. thick, 2.5% moisture	31.9	90		.735
		31.9	60		.714
		31.9	30		.693
		31.9	0		.672
		31.9	−30		.650
		31.9	−60		.629
	Fir plywood, four $\frac{1}{4}$ in. thick, 0% moisture	31.9	90		.707
		31.9	60		.683
		31.9	30		.659
		31.9	0		.636
		31.9	−30		.612
		31.9	−60		.589
	Long leaf yellow pine, 0% moisture	40	75	.86	
	Mahogany	34.3	86	.90	
	Hard maple, 0% moisture	46.0	75	1.05	
	Maple	44.3	86	1.10	
	Maple crossgrain	40.0	75	1.20	
	Norway pine, 0% moisture	32.0	75	.74	
	Pine $1\frac{1}{4}$ in. thick	24.1	90		.707
		24.1	60		.689
		14.1	30		.670
		24.1	0		.651
		24.1	−30		.633
		24.1	−60		.614
	Red cypress, 0% moisture	32.0	75	.77	
	Short leaf yellow pine, 0% moisture	36.0	75	.91	
	Soft elm, 0% moisture	34.0	75	.88	
	Soft maple, 0% moisture	42.0	75	.95	
	Sugar pine, 0% moisture	28.0	75	.64	
	Virginia pine	34.3	86	.96	
	West coast hemlock, 0% moisture	30.0	75	.79	
	White pine	31.2	86	.78	
	Yellow pine			1.00	
	Sawdust various	12.0	90	.41	
	Shavings from planer	8.8	90	.41	
	Shavings from maple, beech & birch	13.2	90	.36	
	Air space-bounded by ordinary materials 1 in. thick		90	1.18	
			60	1.07	
			30	.98	
			0	.89	
			−30	.82	
			−60	.74	
	Air space—bounded warm side by aluminum paint 1 in. thick		90	.64	
			60	.62	
			30	.59	
			0	.57	
			−30	.55	
			−60	.53	
	Air space—bounded warm side by aluminum foil 1 in. thick		90	.47	
			60	.46	
			30	.46	
			0	.45	
			−30	.45	
			−60	.45	

Table 5. Food Products Data

Name	Freezing point F	% water	Sp ht, Btu/lb F		Enthalpy of fusion, Btu/lb	Latent heat of respiration, Btu/24 hr per ton	
			Above freezing	Below freezing		F	
VEGETABLES							
Artichokes	27.5	92	.90	.49	134	40	10,140
Asparagus	30	94	.91	.49	135	40	11.500
Beans, string	30	83	.87	.47	119	32	1,694
						40	2,706
Beans, Lima	30	69	.75		79		
Beans, dried		12.5	.30	.25	18		
Beets	27	90	.90	.48	129	32	1,166
						40	1,820
Broccoli	29	93	.90	.48	135		
Brussel sprouts	31	94.5	.91	.49	136		
Cabbage	31	91.5	.93	.47	132		
Cantaloupe	30.5	89	.91	.47	128	40	506
						60	3,938
Carrots	29.5	88	.86	.45	126	32	314
						40	1,408
Cauliflower	30	92.5	.90	.46	133		
Celery	30	94.5	.91	.46	136	32	704
						40	2,606
Corn (green)	29	75.5	.86	.38	108	32	1,694
						40	2,706
Corn (dried)		10.5	.29	.24	15		
Cucumbers	30.5	95.5	.93	.48	137		
Eggplant	30.5	92	.91	.45	132		
Endive	30.9	89	.90	.46	136		
Lettuce	31.2	89	.90	.46	136	32	638
						40	7,392
Mushrooms		88	.90		126		
Onions	30	89	.91	.51	130	32	880
						50	1,870
Parsnips	29	83	.86	.44	119		
Peas (green)	30	75	.85	.42	107		
Peas (dried)		9.5	.28	.23	136		
Peppers (sweet)	30	85	.90	.46		40	1,518
Potatoes (white)	29	78.5	.86	.47	113	40	1,430
						50	1,320
Potatoes (sweet)	28.5	78	.86	.42	102	40	1,100
Pumpkin	30	88	.90	.43	127		
Radishes		90	.90	.48	132		
Rhubarb	28	94	.90	.46	134		
Sauerkraut	26	89	.92	.52	128		
Spinach	30	90	.92	.51	129		
Squash	29	88	.88	.43	128		
Tomatoes (green)	30.5	94	.92	.46	132	60	2,574
Tomatoes (ripening)	30.5	94.5	.92	.46	132	40	132
Turnips	30	89.5	.90	.45	128	32	66
						40	572
Vegetables (mixed)	30	90	.90	.45	130		
MEATS & FISH							
Bacon		20	.50	.30	29	40	5,000
						28	2,400
Beef (dried)		5–15	.22–.34	.19–.26	7–22	45	280
						36	186
Beef (fresh)	27	68	.75	.40	98	35	10,000
						30	3,400
Brined meats			.75			40	1,980
						31	1,540

Table 5. Food Products Data (Continued)

Name	Freezing point F	% water	Sp ht, Btu/lb F		Enthalpy of fusion, Btu/lb	Latent heat of respiration, Btu/24 hr per ton	
			Above freezing	Below freezing		F	
Cut meats	29	65	.72	.40	95	34	11,200
Fish (frozen)	28	70	.76	.41	101	5	200
						15	2,040
Fish (iced)		70	.76	.41	101	30	840
						34	1,140
Hams & loins	27	60	.68	.38	86.5	28	3,500
						34	6,800
Lamb	29	58	.67	.30	83.5	34	6,720
						28	2,500
Livers	29	65.5	.72	.40	93.3	28	7,200
						20	1,506
Oysters (shell)	27	80.4	.83	.44	116	35	8,300
						32	1,020
Oysters (tub)	27	87	.90	.46	125	35	4,500
						32	440
Pork (fresh)	28	60	.68	.38	86.5	34	6,800
Pork (smoked)		57	.60	.32		55	2,500
Poultry (fresh)	27	74	.79	.37	106	28	6,880
Poultry (frozen)	27	74	.79	.37	106	15	2,480
							480
Sausage (casings)			.60				
Sausage (drying)	26	65.5	.89	.56	93		
Ssuage (franks)	29	60	.86	.56	86	45	8,600
Sausage (fresh)	26	65	.89	.56	93	35	8,600
Sausage (smoked)	25	60	.86	.56	86	45	6,440
Veal	29	63	.71	.39	91	34	7,010
						28	2,500
MISCELLANEOUS							
Beer (wood kegs)	28	.92	1.0				
Beer (steel kegs)	28	.92	1.0				
Butter	30-0	15	.64	.34	15	45	3,960
						35	3,640
Caviar (tub)	20					40	3,820
Cheese (curing)							
Cheese (American)	17	55	.64	.36	79	40	4,680
Cheese (Camembert)	18	60	.70	.40	86	40	4,920
Cheese (limburger)	19	60	.70	.40	86	40	4,920
Cheese (Roquefort)	3	55	.65	.32	79	45	4,000
Cheese (Swiss)	15	55	.64	.36	79	40	4,660
Chocolate (coating)	95–85	.5	.30	.55	40		
Cream (40%)	28	55	.85	.40	90		
Eggs (crated)	27	73	.76	.40	100	40	7,010
Eggs (frozen)	27			.41	100		
Flowers (cut)	32						480/Floor
Furs—Woolens				.40			
Honey		18	.35	.26	26	40	1,420
Hops						35	1,500
Ice cream	27-0	10	.78	.45	96		
Lard			.52			45	3,960
Malt						50	1,500
Maple sugar		5	.24	.21	7	45	1,420
Maple syrup		36	.49	.31	52	45	1,420
Milk	31	87.5	.93	.49	124		
Nuts (dried)		3–10	.21–.29	.19–.24	4.3–14	35	1,000
Oleomargarine			.48			45	3,960
Tobacco & cigars	25						
Vaccine serum							
Candy			.93				
Flour		13.5	.38	.28			

Table 5. Food Products Data (Concluded)

Name	Freezing point F	% water	Sp ht, Btu/lb F		Enthalpy of fusion, Btu/lb	Latent heat of respiration, Btu/24 hr per ton	
			Above freezing	Below freezing		F	
FRUITS							
Apples	28.5	85	.90	.49	122	32	770
						40	1,435
Apricots	29	85	.92	.50	122		
Avocadoes	31	94	.91	.49	136		
Bananas	26 30	75	.90		108	68	8,360
Berries (general)	26–29	84	.90	.49	120		
Cherries	24–28	84	.92	.46	121		
Cranberries	27	88	.91	.47	122		
Dates (dry-cured)	−4	18	.35		26		
Dates (dry non-cured)	−4	18	.35		26		
Dates (fresh)	28	80	.88	.48	116		
Figs (fresh)	28	80	.88	.48	116		
Fruit (dried)		30	.47	.32	43		
Grapes	28	77	.90	.61	112	35	830
Grapefruit	28.5	88	.91	.49	128	32	460
						40	1,070
Lemons	28	88	.91	.49	126	40	810
						60	2,970
Limes	29	88	.91	.49	126	40	810
						60	2,970
Mangoes	32	93	.90	.46	134		
Melons (general)	28-30	85	.91	.46	115		
Oranges	28	87	.90	.47	125	32	795
						40	1,400
Peaches	29.5	88	.92	.48	128	32	1,110
						40	1,735
Pears	28.5	84	.91	.49	122	32	770
Persimmons	28.5	66	.80	.46	96		
Pineapples	28	88	.90	.50	128		
Plums	28.5	84	.91	.49	120		
Pomegranates	28	77	.87	.48	112		
Prunes (fresh)	28	80	.88	.48	116		
Quinces	28	85	.90	.49	122		
Strawberries	30	90.5	.92	.48	131	32	3,265
						40	5,865

Table 6. Permeability of Various Materials to Water Vapor

(Units—Grains per sq ft per day per mm of Hg)

Item No.	Material	Thickness, in.	Investigator and permeability: Babbitt (dry method)	Miscellaneous	J-M Research Laboratories: Humid method	J-M Research Laboratories: Dry method
	PAPER AND FELTS					
1	Kraft paper, 1 sheet	0.00394	74.3			
2	Kraft paper, 2 sheets in series		47.2			
3	Kraft paper, 3 sheets in series		35.4			
4	Kraft paper, 4 sheets in series		28.1			
5	Kraft paper, 5 sheets in series		26.4			
6	Uncoated kraft (about 30-lb.)				130	
7	Rosin sized sheathing paper (30–40 lb.)				91	
8	Asphalt saturated rag felts (15–20 lb.)		0.4 –30		4.0 –30	
9	Asphalt saturated and coated kraft papers		0.45	0.17– 2.8T	0.0 – 3.0	
10	Asphalt saturated asbestos felt	0.03–0.04			1.0 – 5.0	
11	Asphalt saturated and coated asbestos felt, 20-lb.	0.03–0.04	0.14		1.0 – 2.0	
12	Tarred slaters felt	0.03–0.04	4–10	5.2 –25.0T	40–45	
13	35-lb. specification cold application asbestos felt	0.052			0.0	
14	No. 50 asbestos waterproofing felt	0.051			0.9	
15	asph. sat. rfg. felt (30-lb.) 1 sheet	0.062			5.1	1.7
6	asph. sat. rfg. felt (30-lb.) 2 sheets in series				2.5	1.1
17	asph. sat. rfg. felt (30-lb.) 3 sheets in series				1.4	0.87
18	Duplex papers		0.3 – 1.5	0.57– 2.5T	0.2 – 6.0	
19	Waxed papers		0.1 – 4.5			
20	Heavy roofing sheets		0.03– 1.4	0.13– 0.17T	0.03– 1.0	
	MISCELLANEOUS MATERIALS					
21	Wood	0.5	1.74– 2.02			
22	Wood	1.0	1.08– 1.25			
23	Insulating board	0.5	36.4			
24	Insulating board	1.0	20.6			
25	Insulating board, Plain	25/32		26.0–35.0T	30.0	
26	Insulating board, Sealed or Coated	25/32		3.1 – 4.3T	0.20–10.0	
27	Rock Cork	0.523			5.6	2.8
28	Rock Cork	0.984			4.2	2.3
29	Rock Cork	2.001			2.7	1.9
30	Pure corkboard (insulating)	1.0		6.2T	9.0	
31	Plaster on metal lath	0.5		15.2R		
32	Plaster on metal lath	0.75		12.8R		
33	Plaster on wood lath	0.5		6.4R, 3.6–3.9T		
34	Plaster ½-in. on ⅜-in. gypsum board			10.7R		
35	Plaster ½-in. on ½-in. wood fiber board			15.5R		
36	Stucco	1.0			2.0	
37	Marinite	0.625			7.5	
38	Transite	0.25			6.0	
39	Flexboard	0.125			15.5	5.5
40	1½-in. Concrete (1:2:4 Mix)				2.7	
41	Concrete	1.5		15.7B		
42	Brick wall section—with mortar	4.0		7.5B		
43	Tile wall section—with mortar	4.0		1.2B		
	COMBINATIONS					
44	⅛-in. Zeroseal on ½-in. J-M Rock Cork				0.16– 0.21	
45	⅛-in. Zerokote on ½-in. J-M Rock Cork				2.4	
46	¼-in. Portland cement mortar (1:3 Mix) on ½-in. J-M Rock Cork				17.6	
47	½-in. Rock Cork dipped in asphalt, then applied to 15-lb. roofing felt nailed to wood sheathing				0.51– 0.75	
48	½-in. Rock Cork dipped in hot asphalt, then applied to 1½-in. concrete—concrete primer				0.24	
49	½-in. Rock Cork dipped in hot asphalt, then applied to Vaporseal paper nailed to wood sheathing				0.00	

Note: Under Miscellaneous: B denotes Barre—temperature gradient method; R denotes Rowley—temperature gradient method; T denotes Teesdale—humid method.

13. MATHEMATICAL TABLES

Table 1. Natural Trigonometric Functions

Degrees	Sines							Cosines
	0′	10′	20′	30′	40′	50′	60′	
0	0.00000	0.00291	0.00582	0.00873	0.01164	0.01454	0.01745	89
1	0.01745	0.02036	0.02327	0.02618	0.02908	0.03199	0.03490	88
2	0.03490	0.03781	0.04071	0.04362	0.04653	0.04943	0.05234	87
3	0.05234	0.05524	0.05814	0.06105	0.06395	0.06685	0.06976	86
4	0.06976	0.07266	0.07556	0.07846	0.08136	0.08426	0.08716	85
5	0.08716	0.09005	0.09295	0.09585	0.09874	0.10164	0.10453	84
6	0.10453	0.10742	0.11031	0.11320	0.11609	0.11898	0.12187	83
7	0.12187	0.12476	0.12764	0.13053	0.13341	0.13629	0.13917	82
8	0.13917	0.14205	0.14493	0.14781	0.15069	0.15356	0.15643	81
9	0.15643	0.15931	0.16218	0.16505	0.16792	0.17078	0.17365	80
10	0.17365	0.17651	0.17937	0.18224	0.18509	0.18795	0.19081	79
11	0.19081	0.19366	0.19652	0.19937	0.20222	0.20507	0.20791	78
12	0.20791	0.21076	0.21360	0.21644	0.21928	0.22212	0.22495	77
13	0.22495	0.22778	0.23062	0.23345	0.23627	0.23910	0.24192	76
14	0.24192	0.24474	0.24756	0.25038	0.25320	0.25601	0.25882	75
15	0.25882	0.26163	0.26443	0.26724	0.27004	0.27284	0.27564	74
16	0.27564	0.27843	0.28123	0.28402	0.28680	0.28959	0.29237	73
17	0.29237	0.29515	0.29793	0.30071	0.30348	0.30625	0.30902	72
18	0.30902	0.31178	0.31454	0.31730	0.32006	0.32282	0.32557	71
19	0.32557	0.32832	0.33106	0.33381	0.33655	0.33929	0.34202	70
20	0.34202	0.34475	0.34748	0.35021	0.35293	0.35565	0.35837	69
21	0.35837	0.36108	0.36379	0.36650	0.36921	0.37191	0.37461	68
22	0.37461	0.37730	0.37999	0.38268	0.38537	0.38805	0.39073	67
23	0.39073	0.39341	0.39608	0.39875	0.40142	0.40408	0.40674	66
24	0.40674	0.40939	0.41204	0.41469	0.41734	0.41998	0.42262	65
25	0.42262	0.42525	0.42788	0.43051	0.43313	0.43575	0.43837	64
26	0.43837	0.44098	0.44359	0.44620	0.44880	0.45140	0.45399	63
27	0.45399	0.45658	0.45917	0.46175	0.46433	0.46690	0.46947	62
28	0.46947	0.47204	0.47460	0.47716	0.47971	0.48226	0.48481	61
29	0.48481	0.48735	0.48989	0.49242	0.49495	0.49748	0.50000	60
30	0.50000	0.50252	0.50503	0.50754	0.51004	0.51254	0.51504	59
31	0.51504	0.51753	0.52002	0.52250	0.52498	0.52745	0.52992	58
32	0.52992	0.53238	0.53484	0.53730	0.53975	0.54220	0.54464	57
33	0.54464	0.54708	0.54951	0.55194	0.55436	0.55678	0.55919	56
34	0.55919	0.56160	0.56401	0.56641	0.56880	0.57119	0.57358	55
35	0.57358	0.57596	0.57833	0.58070	0.58307	0.58543	0.58779	54
36	0.58779	0.59014	0.59248	0.59482	0.59716	0.59949	0.60182	53
37	0.60182	0.60414	0.60645	0.60876	0.61107	0.61337	0.61566	52
38	0.61566	0.61795	0.62024	0.62251	0.62479	0.62706	0.62932	51
39	0.62932	0.63158	0.63383	0.63608	0.63832	0.64056	0.64279	50
40	0.64279	0.64501	0.64723	0.64945	0.65166	0.65386	0.65606	49
41	0.65606	0.65825	0.66044	0.66262	0.66480	0.66697	0.66913	48
42	0.66913	0.67129	0.67344	0.67559	0.67773	0.67987	0.68200	47
43	0.68200	0.68412	0.68624	0.68835	0.69046	0.69256	0.69466	46
44	0.69466	0.69675	0.69883	0.70091	0.70298	0.70505	0.70711	45
	60′	50′	40′	30′	20′	10′	0′	
Sines	Cosines							Degrees

Table 1. Natural Trigonometric Functions (Continued)

Degrees	Cosines 0′	10′	20′	30′	40′	50′	60′	Sines
0	1.00000	1.00000	0.99998	0.99996	0.99993	0.99989	0.99985	89
1	0.99985	0.99979	0.99973	0.99966	0.99958	0.99949	0.99939	88
2	0.99939	0.99929	0.99917	0.99905	0.99892	0.99878	0.99863	87
3	0.99863	0.99847	0.99831	0.99813	0.99795	0.99776	0.99756	86
4	0.99756	0.99736	0.99714	0.99692	0.99668	0.99644	0.99619	85
5	0.99619	0.99594	0.99567	0.99540	0.99511	0.99482	0.99452	84
6	0.99452	0.99421	0.99390	0.99357	0.99324	0.99290	0.99255	83
7	0.99255	0.99219	0.99182	0.99144	0.99106	0.99067	0.99027	82
8	0.99027	0.98986	0.98944	0.98902	0.98858	0.98814	0.98769	81
9	0.98769	0.98723	0.98676	0.98629	0.98580	0.98531	0.98481	80
10	0.98481	0.98430	0.98378	0.98325	0.98272	0.98218	0.98163	79
11	0.98163	0.98107	0.98050	0.97992	0.97934	0.97875	0.97815	78
12	0.97815	0.97754	0.97692	0.97630	0.97566	0.97502	0.97437	77
13	0.97437	0.97371	0.97304	0.97237	0.97169	0.97100	0.97030	76
14	0.97030	0.96959	0.96887	0.96815	0.96742	0.96667	0.96593	75
15	0.96593	0.96517	0.96440	0.96363	0.96285	0.96206	0.96126	74
16	0.96126	0.96048	0.95964	0.95882	0.95799	0.95715	0.95630	73
17	0.95630	0.95545	0.95459	0.95372	0.95284	0.95195	0.95106	72
18	0.95106	0.95015	0.94924	0.94832	0.94740	0.94646	0.94552	71
19	0.94552	0.94457	0.94361	0.94264	0.94167	0.94068	0.93969	70
20	0.93969	0.93869	0.93769	0.93667	0.93565	0.93462	0.93358	69
21	0.93358	0.93253	0.93148	0.93042	0.92935	0.92827	0.92718	68
22	0.92718	0.92609	0.92499	0.92388	0.92276	0.92164	0.92050	67
23	0.92050	0.91936	0.91822	0.91706	0.91590	0.91472	0.91355	66
24	0.91355	0.91236	0.91116	0.90996	0.90875	0.90753	0.90631	65
25	0.90631	0.90507	0.90383	0.90259	0.90133	0.90007	0.89879	64
26	0.89879	0.89752	0.89623	0.89493	0.89363	0.89232	0.89101	63
27	0.89101	0.88968	0.88835	0.88701	0.88566	0.88431	0.88295	62
28	0.88295	0.88158	0.88020	0.87882	0.87743	0.87603	0.87462	61
29	0.87462	0.87321	0.87178	0.87036	0.86892	0.86748	0.86603	60
30	0.86603	0.86457	0.86310	0.86163	0.86015	0.85866	0.85717	59
31	0.85717	0.85567	0.85416	0.85264	0.85112	0.84959	0.84805	58
32	0.84805	0.84650	0.84495	0.84339	0.84182	0.84025	0.83867	57
33	0.83867	0.83708	0.83549	0.83389	0.83228	0.83066	0.82904	56
34	0.82904	0.82741	0.82577	0.82413	0.82248	0.82082	0.81915	55
35	0.81915	0.81748	0.81580	0.81412	0.81242	0.81072	0.80902	54
36	0.80902	0.80730	0.80558	0.80386	0.80212	0.80038	0.79864	53
37	0.79864	0.79688	0.79512	0.79335	0.79158	0.78980	0.78801	52
38	0.78801	0.78622	0.78442	0.78261	0.78079	0.77897	0.77715	51
39	0.77715	0.77531	0.77347	0.77162	0.76977	0.76791	0.76604	50
40	0.76604	0.76417	0.76229	0.76041	0.75851	0.75661	0.75471	49
41	0.75471	0.75280	0.75088	0.74896	0.74703	0.74509	0.74314	48
42	0.74314	0.74120	0.73924	0.73728	0.73531	0.73333	0.73135	47
43	0.73135	0.72937	0.72737	0.72537	0.72337	0.72136	0.71934	46
44	0.71934	0.71732	0.71529	0.71325	0.71121	0.70916	0.70711	45
Cosines	60′	50′	40′	30′	20′	10′	0′	Degrees
	Sines							

Table 1. Natural Trigonometric Functions (Concluded)

Degrees	Tangents							Co-tangents
	0′	10′	20′	30′	40′	50′	60′	
0	0.00000	0.00291	0.00582	0.00873	0.01164	0.01455	0.01746	89
1	0.01746	0.02036	0.02328	0.02619	0.02910	0.03201	0.03492	88
2	0.03492	0.03783	0.04075	0.04366	0.04658	0.04949	0.05241	87
3	0.05241	0.05533	0.05824	0.06116	0.06408	0.06700	0.06993	86
4	0.06993	0.07285	0.07578	0.07870	0.08163	0.08456	0.08749	85
5	0.08749	0.09042	0.09335	0.09629	0.09923	0.10216	0.10510	84
6	0.10510	0.10805	0.11099	0.11394	0.11688	0.11983	0.12278	83
7	0.12278	0.12574	0.12869	0.13165	0.13461	0.13758	0.14054	82
8	0.14054	0.14351	0.14648	0.14945	0.15243	0.15540	0.15838	81
9	0.15838	0.16137	0.16435	0.16734	0.17033	0.17333	0.17633	80
10	0.17633	0.17933	0.18233	0.18534	0.18835	0.19136	0.19438	79
11	0.19438	0.19740	0.20042	0.20345	0.20648	0.20952	0.21256	78
12	0.21256	0.21560	0.21864	0.22169	0.22475	0.22781	0.23087	77
13	0.23087	0.23393	0.23700	0.24008	0.24316	0.24624	0.24933	76
14	0.24933	0.25242	0.25552	0.25862	0.26172	0.26483	0.26795	75
15	0.26795	0.27107	0.27419	0.27732	0.28046	0.28360	0.28675	74
16	0.28675	0.28990	0.29305	0.29621	0.29938	0.30255	0.30573	73
17	0.30573	0.30891	0.31210	0.31530	0.31850	0.32171	0.32492	72
18	0.32492	0.32814	0.33136	0.33460	0.33783	0.34108	0.34433	71
19	0.34433	0.34758	0.35085	0.35412	0.35740	0.36068	0.36397	70
20	0.36397	0.36727	0.37057	0.37388	0.37720	0.38053	0.38386	69
21	0.38386	0.38721	0.39055	0.39391	0.39727	0.40065	0.40403	68
22	0.40403	0.40741	0.41081	0.41421	0.41763	0.42105	0.42447	67
23	0.42447	0.42791	0.43136	0.43481	0.43828	0.44175	0.44523	66
24	0.44523	0.44872	0.45222	0.45573	0.45924	0.46277	0.46631	65
25	0.46631	0.46985	0.47341	0.47698	0.48055	0.48414	0.48773	64
26	0.48773	0.49134	0.49495	0.49858	0.50222	0.50587	0.50953	63
27	0.50953	0.51320	0.51688	0.52057	0.52427	0.52798	0.53171	62
28	0.53171	0.53545	0.53920	0.54296	0.54674	0.55051	0.55431	61
29	0.55431	0.55812	0.56194	0.56577	0.56962	0.57348	0.57735	60
30	0.57735	0.58124	0.58513	0.58905	0.59297	0.59691	0.60086	59
31	0.60086	0.60483	0.60881	0.61280	0.61681	0.62083	0.62487	58
32	0.62487	0.62892	0.63299	0.63707	0.64117	0.64528	0.64941	57
33	0.64941	0.65355	0.65771	0.66189	0.66608	0.67028	0.67451	56
34	0.67451	0.67875	0.68301	0.68728	0.69157	0.69588	0.70021	55
35	0.70021	0.70455	0.70891	0.71329	0.71769	0.72211	0.72654	54
36	0.72654	0.73100	0.73547	0.73996	0.74447	0.74900	0.75355	53
37	0.75355	0.75812	0.76272	0.76733	0.77196	0.77661	0.78129	52
38	0.78129	0.78598	0.79070	0.79544	0.80020	0.80498	0.80978	51
39	0.80978	0.81461	0.81946	0.82434	0.82923	0.83415	0.83910	50
40	0.83910	0.84407	0.84906	0.85408	0.85912	0.86419	0.86929	49
41	0.86929	0.87441	0.87955	0.88473	0.88992	0.89515	0.90040	48
42	0.90040	0.90569	0.91099	0.91633	0.92170	0.92709	0.93252	47
43	0.93252	0.93797	0.94345	0.94896	0.95451	0.96008	0.96569	46
44	0.96569	0.97133	0.97700	0.98270	0.98843	0.99420	1.00000	45
	60′	50′	40′	30′	20′	10′	0′	
Tangents	Cotangents							Degrees

Table 2. Common Logarithms, Base of 10

N	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5	6	7	8	9
10	0000	0043	0086	0128	0170	0212	0253	0294	0334	0374	*4	8	12	17	21	25	29	33	37
11	0414	0453	0492	0531	0569	0607	0645	0682	0719	0755	4	8	11	15	19	23	26	30	34
12	0792	0828	0864	0899	0934	0969	1004	1038	1072	1106	3	7	10	14	17	21	24	28	31
13	1139	1173	1206	1239	1271	1303	1335	1367	1399	1430	3	6	10	13	16	19	23	26	29
14	1461	1492	1523	1553	1584	1614	1644	1673	1703	1732	3	6	9	12	15	18	21	24	27
15	1761	1790	1818	1847	1875	1903	1931	1959	1987	2014	*3	6	8	11	14	17	20	22	25
16	2041	2068	2095	2122	2148	2175	2201	2227	2253	2279	3	5	8	11	13	16	18	21	24
17	2304	2330	2355	2380	2405	2430	2455	2480	2504	2529	2	5	7	10	12	15	17	20	22
18	2553	2577	2601	2625	2648	2672	2695	2718	2742	2765	2	5	7	9	12	14	16	19	21
19	2788	2810	2833	2856	2878	2900	2923	2945	2967	2989	2	4	7	9	11	13	16	18	20
20	3010	3032	3054	3075	3096	3118	3139	3160	3181	3201	2	4	6	8	11	13	15	17	19
21	3222	3243	3263	3284	3304	3324	3345	3365	3385	3404	2	4	6	8	10	12	14	16	18
22	3424	3444	3464	3483	3502	3522	3541	3560	3579	3598	2	4	6	8	10	12	14	15	17
23	3617	3636	3655	3674	3692	3711	3729	3747	3766	3784	2	4	6	7	9	11	13	15	17
24	3802	3820	3838	3856	3874	3892	3909	3927	3945	3962	2	4	5	7	9	11	12	14	16
25	3979	3997	4014	4031	4048	4065	4082	4099	4116	4133	2	3	5	7	9	10	12	14	15
26	4150	4166	4183	4200	4216	4232	4249	4265	4281	4298	2	3	5	7	8	10	11	13	15
27	4314	4330	4346	4362	4378	4393	4409	4425	4440	4456	2	3	5	6	8	9	11	13	14
28	4472	4487	4502	4518	4533	4548	4564	4579	4594	4609	2	3	5	6	8	9	11	12	14
29	4624	4639	4654	4669	4683	4698	4713	4728	4742	4757	1	3	4	6	7	9	10	12	13
30	4771	4786	4800	4814	4829	4843	4857	4871	4886	4900	1	3	4	6	7	9	10	11	13
31	4914	4928	4942	4955	4969	4983	4997	5011	5024	5038	1	3	4	6	7	8	10	11	12
32	5051	5065	5079	5092	5105	5119	5132	5145	5159	5172	1	3	4	5	7	8	9	11	12
33	5185	5198	5211	5224	5237	5250	5263	5276	5289	5302	1	3	4	5	6	8	9	10	12
34	5315	5328	5340	5353	5366	5378	5391	5403	5416	5428	1	3	4	5	6	8	9	10	11
35	5441	5453	5465	5478	5490	5502	5514	5527	5539	5551	1	2	4	5	6	7	9	10	11
36	5563	5575	5587	5599	5611	5623	5635	5647	5658	5670	1	2	4	5	6	7	8	10	11
37	5682	5694	5705	5717	5729	5740	5752	5763	5775	5786	1	2	3	5	6	7	8	9	11
38	5798	5809	5821	5832	5843	5855	5866	5877	5888	5899	1	2	3	5	6	7	8	9	10
39	5911	5922	5933	5944	5955	5966	5977	5988	5999	6010	1	2	3	4	5	7	8	9	10
40	6021	6031	6042	6053	6064	6075	6085	6096	6107	6117	1	2	3	4	5	6	8	9	10
41	6128	6138	6149	6160	6170	6180	6191	6201	6212	6222	1	2	3	4	5	6	7	8	9
42	6232	6243	6253	6263	6274	6284	6294	6304	6314	6325	1	2	3	4	5	6	7	8	9
43	6335	6345	6355	6365	6375	6385	6395	6405	6415	6425	1	2	3	4	5	6	7	8	9
44	6435	6444	6454	6464	6474	6484	6493	6503	6513	6522	1	2	3	4	5	6	7	8	9
45	6532	6542	6551	6561	6571	6580	6590	6599	6609	6618	1	2	3	4	5	6	7	8	9
46	6628	6637	6646	6656	6665	6675	6684	6693	6702	6712	1	2	3	4	5	6	7	7	8
47	6721	6730	6739	6749	6758	6767	6776	6785	6794	6803	1	2	3	4	5	5	6	7	8
48	6812	6821	6830	6839	6848	6857	6866	6875	6884	6893	1	2	3	4	4	5	6	7	8
49	6902	6911	6920	6928	6937	6946	6955	6964	6972	6981	1	2	3	4	4	5	6	7	8
50	6990	6998	7007	7016	7024	7033	7042	7050	7059	7067	1	2	3	3	4	5	6	7	8
51	7076	7084	7093	7101	7110	7118	7126	7135	7143	7152	1	2	3	3	4	5	6	7	8
52	7160	7168	7177	7185	7193	7202	7210	7218	7226	7235	1	2	2	3	4	5	6	7	7
53	7243	7251	7259	7267	7275	7284	7292	7300	7308	7316	1	2	2	3	4	5	6	6	7
54	7324	7332	7340	7348	7356	7364	7372	7380	7388	7396	1	2	2	3	4	5	6	6	7
N	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9

* Interpolation in this section of the table is inaccurate.

Table 2. Common Logarithms (Concluded)

N	0	1	2	3	4	5	6	7	8	9	Proportional Parts 1	2	3	4	5	6	7	8	9
55	7404	7412	7419	7427	7435	7443	7451	7459	7466	7474	1	2	2	3	4	5	5	6	7
56	7482	7490	7497	7505	7513	7520	7528	7536	7543	7551	1	2	2	3	4	5	5	6	7
57	7559	7566	7574	7582	7589	7597	7604	7612	7619	7627	1	2	2	3	4	5	5	6	7
58	7634	7642	7649	7657	7664	7672	7679	7686	7694	7701	1	1	2	3	4	4	5	6	7
59	7709	7716	7723	7731	7738	7745	7752	7760	7767	7774	1	1	2	3	4	4	5	6	7
60	7782	7789	7796	7803	7810	7818	7825	7832	7839	7846	1	1	2	3	4	4	5	6	6
61	7853	7860	7868	7875	7882	7889	7896	7903	7910	7917	1	1	2	3	4	4	5	6	6
62	7924	7931	7938	7945	7952	7959	7966	7973	7980	7987	1	1	2	3	3	4	5	6	6
63	7993	8000	8007	8014	8021	8028	8035	8041	8048	8055	1	1	2	3	3	4	5	5	6
64	8062	8069	8075	8082	8089	8096	8102	8109	8116	8122	1	1	2	3	3	4	5	5	6
65	8129	8136	8142	8149	8156	8162	8169	8176	8182	8189	1	1	2	3	3	4	5	5	6
66	8195	8202	8209	8215	8222	8228	8235	8241	8248	8254	1	1	2	3	3	4	5	5	6
67	8261	8267	8274	8280	8287	8293	8299	8306	8312	8319	1	1	2	3	3	4	5	5	6
68	8325	8331	8338	8344	8351	8357	8363	8370	8376	8382	1	1	2	3	3	4	4	5	6
69	8388	8395	8401	8407	8414	8420	8426	8432	8439	8445	1	1	2	2	3	4	4	5	6
70	8451	8457	8463	8470	8476	8482	8488	8494	8500	8506	1	1	2	2	3	4	4	5	6
71	8513	8519	8525	8531	8537	8543	8549	8555	8561	8567	1	1	2	2	3	4	4	5	5
72	8573	8579	8585	8591	8597	8603	8609	8615	8621	8627	1	1	2	2	3	4	4	5	5
73	8633	8639	8645	8651	8657	8663	8669	8675	8681	8686	1	1	2	2	3	4	4	5	5
74	8692	8698	8704	8710	8716	8722	8727	8733	8739	8745	1	1	2	2	3	4	4	5	5
75	8751	8756	8762	8768	8774	8779	8785	8791	8797	8802	1	1	2	2	3	3	4	5	5
76	8808	8814	8820	8825	8831	8837	8842	8848	8854	8859	1	1	2	2	3	3	4	5	5
77	8865	8871	8876	8882	8887	8893	8899	8904	8910	8915	1	1	2	2	3	3	4	4	5
78	8921	8927	8932	8938	8943	8949	8954	8960	8965	8971	1	1	2	2	3	3	4	4	5
79	8976	8982	8987	8993	8998	9004	9009	9015	9020	9025	1	1	2	2	3	3	4	4	5
80	9031	9036	9042	9047	9053	9058	9063	9069	9074	9079	1	1	2	2	3	3	4	4	5
81	9085	9090	9096	9101	9106	9112	9117	9122	9128	9133	1	1	2	2	3	3	4	4	5
82	9138	9143	9149	9154	9159	9165	9170	9175	9180	9186	1	1	2	2	3	3	4	4	5
83	9191	9196	9201	9206	9212	9217	9222	9227	9232	9238	1	1	2	2	3	3	4	4	5
84	9243	9248	9253	9258	9263	9269	9274	9279	9284	9289	1	1	2	2	3	3	4	4	5
85	9294	9299	9304	9309	9315	9320	9325	9330	9335	9340	1	1	2	2	3	3	4	4	5
86	9345	9350	9355	9360	9365	9370	9375	9380	9385	9390	1	1	2	2	3	3	4	4	5
87	9395	9400	9405	9410	9415	9420	9425	9430	9435	9440	0	1	1	2	2	3	3	4	4
88	9445	9450	9455	9460	9465	9469	9474	9479	9484	9489	0	1	1	2	2	3	3	4	4
89	9494	9499	9504	9509	9513	9518	9523	9528	9533	9538	0	1	1	2	2	3	3	4	4
90	9542	9547	9552	9557	9562	9566	9571	9576	9581	9586	0	1	1	2	2	3	3	4	4
91	9590	9595	9600	9605	9609	9614	9619	9624	9628	9633	0	1	1	2	2	3	3	4	4
92	9638	9643	9647	9652	9657	9661	9666	9671	9675	9680	0	1	1	2	2	3	3	4	4
93	9685	9689	9694	9699	9703	9708	9713	9717	9722	9727	0	1	1	2	2	3	3	4	4
94	9731	9736	9741	9745	9750	9754	9759	9763	9768	9773	0	1	1	2	2	3	3	4	4
95	9777	9782	9786	9791	9795	9800	9805	9809	9814	9818	0	1	1	2	2	3	3	4	4
96	9823	9827	9832	9836	9841	9845	9850	9854	9859	9863	0	1	1	2	2	3	3	4	4
97	9868	9872	9877	9881	9886	9890	9894	9899	9903	9908	0	1	1	2	2	3	3	4	4
98	9912	9917	9921	9926	9930	9934	9939	9943	9948	9952	0	1	1	2	2	3	3	4	4
99	9956	9961	9965	9969	9974	9978	9983	9987	9991	9996	0	1	1	2	2	3	3	3	4
N	0	1	2	3	4	5	6	7	8	9	1	2	3	4	5	6	7	8	9

Table 3. Decimal of an Inch and of a Foot

Fractions of inch or foot		Inch equivalents to foot fractions	Fractions of inch or foot		Inch equivalents to foot fractions	Fractions of inch or foot		Inch equivalents to foot fractions	Fractions of inch or foot		Inch equivalents to foot fractions
	.0052	1/16		.2552	3 1/16		.5052	6 1/16		.7552	9 1/16
	.0104	1/8		.2604	3 1/8		.5104	6 1/8		.7604	9 1/8
1/64	.015625	3/16	17/64	.265625	3 3/16	33/64	.515625	6 3/16	49/64	.765625	9 3/16
	.0208	1/4		.2708	3 1/4		.5208	6 1/4		.7708	9 1/4
	.0260	5/16		.2760	3 5/16		.5260	6 5/16		.7760	9 5/16
1/32	.03125	3/8	9/32	.28125	3 3/8	17/32	.53125	6 3/8	25/32	.78125	9 3/8
	.0365	7/16		.2865	3 7/16		.5365	6 7/16		.7865	9 7/16
	.0417	1/2		.2917	3 1/2		.5417	6 1/2		.7917	9 1/2
3/64	.046875	9/16	19/64	.296875	3 9/16	35/64	.546875	6 9/16	51/64	.796875	9 9/16
	.0521	5/8		.3021	3 5/8		.5521	6 5/8		.8021	9 5/8
	.0573	11/16		.3073	3 11/16		.5573	6 11/16		.8073	9 11/16
1/16	.0625	3/4	5/16	.3125	3 3/4	9/16	.5625	6 3/4	13/16	.8125	9 3/4
	.0677	13/16		.3177	3 13/16		.5677	6 13/16		.8177	9 13/16
	.0729	7/8		.3229	3 7/8		.5729	6 7/8		.8229	9 7/8
5/64	.078125	15/16	21/64	.328125	3 15/16	37/64	.578125	6 15/16	53/64	.828125	9 15/16
	.0833	1		.3333	4		.5833	7		.8333	10
	.0885	1 1/16		.3385	4 1/16		.5885	7 1/16		.8385	10 1/16
3/32	.09375	1 1/8	11/32	.34375	4 1/8	19/32	.59375	7 1/8	27/32	.84375	10 1/8
	.0990	1 3/16		.3490	4 3/16		.5990	7 3/16		.8490	10 3/16
	.1042	1 1/4		.3542	4 1/4		.6042	7 1/4		.8542	10 1/4
7/64	.109375	1 5/16	23/64	.359375	4 5/16	39/64	.609375	7 5/16	55/64	.859375	10 5/16
	.1146	1 3/8		.3646	4 3/8		.6146	7 3/8		.8646	10 3/8
	.1198	1 7/16		.3698	4 7/16		.6198	7 7/16		.8698	10 7/16
1/8	.1250	1 1/2	3/8	.3750	4 1/2	5/8	.6250	7 1/2	7/8	.8750	10 1/2
	.1302	1 9/16		.3802	4 9/16		.6302	7 9/16		.8802	10 9/16
	.1354	1 5/8		.3854	4 5/8		.6354	7 5/8		.8854	10 5/8
9/64	.140625	1 11/16	25/64	.390625	4 11/16	41/64	.640625	7 11/16	57/64	.890625	10 11/16
	.1458	1 3/4		.3958	4 3/4		.6458	7 3/4		.8958	10 3/4
	.1510	1 13/16		.4010	4 13/16		.6510	7 13/16		.9010	10 13/16
5/32	.15625	1 7/8	13/32	.40625	4 7/8	21/32	.65625	7 7/8	29/32	.90625	10 7/8
	.1615	1 15/16		.4115	4 15/16		.6615	7 15/16		.9115	10 15/16
	.1667	2		.4167	5		.6667	8		.9167	11
11/64	.171875	2 1/16	27/64	.421875	5 1/16	43/64	.671875	8 1/16	59/64	.921875	11 1/16
	.1771	2 1/8		.4271	5 1/8		.6771	8 1/8		.9271	11 1/8
	.1823	2 3/16		.4323	5 3/16		.6823	8 3/16		.9323	11 3/16
3/16	.1875	2 1/4	7/16	.4375	5 1/4	11/16	.6875	8 1/4	15/16	.9375	11 1/4
	.1927	2 5/16		.4427	5 5/16		.6927	8 5/16		.9427	11 5/16
	.1979	2 3/8		.4479	5 3/8		.6979	8 3/8		.9479	11 3/8
13/64	.203125	2 7/16	29/64	.453125	5 7/16	45/64	.703125	8 7/16	61/64	.953125	11 7/16
	.2083	2 1/2		.4583	5 1/2		.7083	8 1/2		.9583	11 1/2
	.2135	2 9/16		.4635	5 9/16		.7135	8 9/16		.9635	11 9/16
7/32	.21875	2 5/8	15/32	.46875	5 5/8	23/32	.71875	8 5/8	31/32	.96875	11 5/8
	.2240	2 11/16		.4740	5 11/16		.7240	8 11/16		.9740	11 11/16
	.2292	2 3/4		.4792	5 3/4		.7292	8 3/4		.9792	11 3/4
15/64	.234375	2 13/16	31/64	.484375	5 13/16	47/64	.734375	8 13/16	63/64	.984375	11 13/16
	.2396	2 7/8		.4896	5 7/8		.7396	8 7/8		.9896	11 7/8
	.2448	2 15/16		.4948	5 15/16		.7448	8 15/16		.9948	11 15/16
1/2	.2500	3	1/2	.5000	6	3/4	.7500	9	1	1.0000	12

Table 4. Wire and Sheet Metal Gages

(Values in approximate decimals of an inch. As a number of gages are in use for various shapes and metals, it is advisable to state the thickness in thousandths when specifying gage number.)

Gage number	American (A.W.G.) or Brown & Sharpe (B. & S.) (for nonferrous wire and sheet)*	U. S. Steel Wire (Stl. W.G.) or Washburn & Moen or Roebling or Am. Steel & Wire Co. [A. (steel) W.G.] (for steel wire)	Birmingham (B.W.G.) (for steel wire) or Stubs Iron Wire (for iron or brass wire)†	Stubs (steel wire)	U. S. Standard (for sheet and plate metal, iron and steel)	Trenton Iron Co.	Standard Birmingham (B.G.) (for sheet and hoop metal)	British Imperial Standard Wire (S.W.G.), or New British Standard (N.B.S.) or English Legal Standard or Imperial Wire Gage	Gage number
0000000	—	0.4900	—	—	0.500	—	—	0.500	0000000
000000	—	.4615	—	—	.469	—	—	.464	000000
00000	—	.4305	—	—	.438	0.450	—	.432	00000
0000	0.460	.3938	0.454	—	.406	.400	—	.400	0000
000	.410	.3625	.425	—	.375	.360	0.5000	.372	000
00	.365	.3310	.380	—	.344	.330	.4452	.348	00
0	.325	.3065	.340	—	.312	.305	.3964	.324	0
1	.289	.2830	.300	0.227	.281	.285	.3532	.300	1
2	.258	.2625	.284	.219	.266	.265	.3147	.276	2
3	.229	.2437	.259	.212	.250	.245	.2804	.252	3
4	.204	.2253	.238	.207	.234	.225	.2500	.232	4
5	.182	.2070	.220	.204	.219	.205	.2225	.212	5
6	.162	.1920	.203	.201	.203	.190	.1981	.192	6
7	.144	.1770	.180	.199	.188	.175	.1764	.176	7
8	.128	.1620	.165	.197	.172	.160	.1570	.160	8
9	.114	.1483	.148	.194	.156	.145	.1398	.144	9
10	.102	.1350	.134	.191	.141	.130	.1250	.128	10
11	.091	.1205	.120	.188	.125	.1175	.1113	.116	11
12	.081	.1055	.109	.185	.109	.105	.0991	.104	12
13	.072	.0915	.095	.182	.094	.0925	.0882	.092	13
14	.064	.0800	.083	.180	078	.080	.0785	.080	14
15	.057	.0720	.072	.178	.070	.070	.0699	.072	15
16	.051	.0625	.065	.175	.062	.061	.0625	.064	16
17	.045	.0540	.058	.172	.056	.0525	.0556	.056	17
18	.040	.0475	.049	.168	.050	.045	.0495	.048	18
19	.036	.0410	.042	.164	.0438	.040	.0440	.040	19
20	.032	.0348	.035	.161	.0375	.035	.0392	.036	20
21	.0285	.0317	.032	.157	.0344	.031	.0349	.032	21
22	.0253	.0286	.028	.155	.0312	.028	.0313	.028	22
23	.0226	.0258	.025	.153	.0281	.025	.0278	.024	23
24	.0201	.0230	.022	.151	.0250	.0225	.0248	.022	24
25	.0179	.0204	.020	.148	.0219	.020	.0220	.020	25
26	.0159	.0181	.018	.146	.0188	.018	.0196	.018	26
27	.0142	.0173	.016	.143	.0172	.017	.0175	.0164	27
28	.0126	.0162	.014	.139	.0156	.016	.0156	.0148	28
29	.0113	.0150	.013	.134	.0141	.015	.0139	.0136	29
30	.0100	.0140	.012	.127	.0125	.014	.0123	.0124	30
31	.0089	.0132	.010	.120	.0109	.013	.0110	.0116	31
32	.0080	.0128	.009	.115	.0102	.012	.0098	.0108	32
33	.0071	.0118	.008	.112	.0094	.011	.0087	.0100	33
34	.0063	.0104	.007	.110	.0086	.010	.0077	.0092	34
35	.0056	.0095	.005	.108	.0078	.0095	.0069	.0084	35
36	.0050	.0090	.004	.106	.0070	.009	.0061	.0076	36
37	.0045	.0085	—	.103	.0066	.0085	.0054	.0068	37
38	.0040	.0080	—	.101	.0062	.008	.0048	.0060	38
39	.0035	.0075	—	.099	—	.0075	—	.0052	39
40	.0031	.0070	—	.097	—	.007	—	.0048	40
41	—	.0066	—	.095	—	—	—	.0044	41
42	—	.0062	—	.092	—	—	—	.0040	42
43	—	.0060	—	.088	—	—	—	.0036	43
44	—	.0058	—	.085	—	—	—	.0032	44
45	—	.0055	—	.081	—	—	—	.0028	45
46	—	.0052	—	.079	—	—	—	.0024	46
47	—	.0050	—	.077	—	—	—	.0020	47
48	—	.0048	—	.075	—	—	—	.0016	48
49	—	.0046	—	.072	—	—	—	.0012	49
50	—	.0044	—	.069	—	—	—	.0010	50

Metric wire gage is ten times the diameter in millimeters.

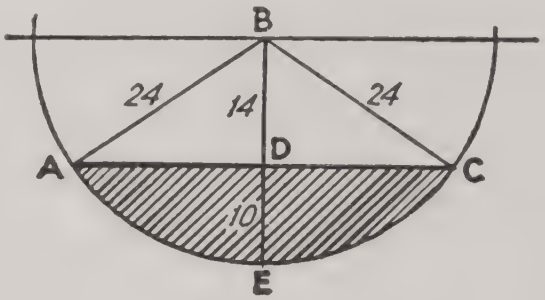
* Sometimes used for iron wire.

† Sometimes used for copper plate and for steel plate 12 gage and heavier and for steel tubes.

Table 5. Capacities of Horizontal Cylindrical Tanks

Method of Calculation

1. *Flat Ends.* Assume diameter = 48 in. and length = 120 in. Required the volume (in U. S. gallons) of shaded area of figure (ACE) of height 10 in.



Area ACE = area $ABCE$ − area ABC.

Area $ABCE = \left(\frac{2\angle ABD}{360}\right) \times$ area of circle, and $\angle ABD$ is found from its cosine which is 14/24.

$$\therefore \quad \angle ABD = 54.25°$$

Area $ABC = 14 \times 24 \times \sin \angle ABD$

or

$$14 \times 24 \times \sin 54.25 = 14 \times 24 \times .8116 = 272.7$$

$$\text{Area } ABCE = \frac{108.50}{360} \times \pi \times (24)^2 = 561.$$

$$\therefore \text{ Area } ACE = 561 - 272.7 = 288.3$$

$$\therefore \text{ Vol (U. S. gallons) per foot of length} = \frac{288.3 \times 12}{231} = 15$$

See Table 6.

To calculate the contents of a tank over one-half full, let h be the depth of the unfilled part of the tank. Find the volume corresponding to this depth from the table and subtract it from the volume of the full tank as given in the last column of each horizontal line of the table.

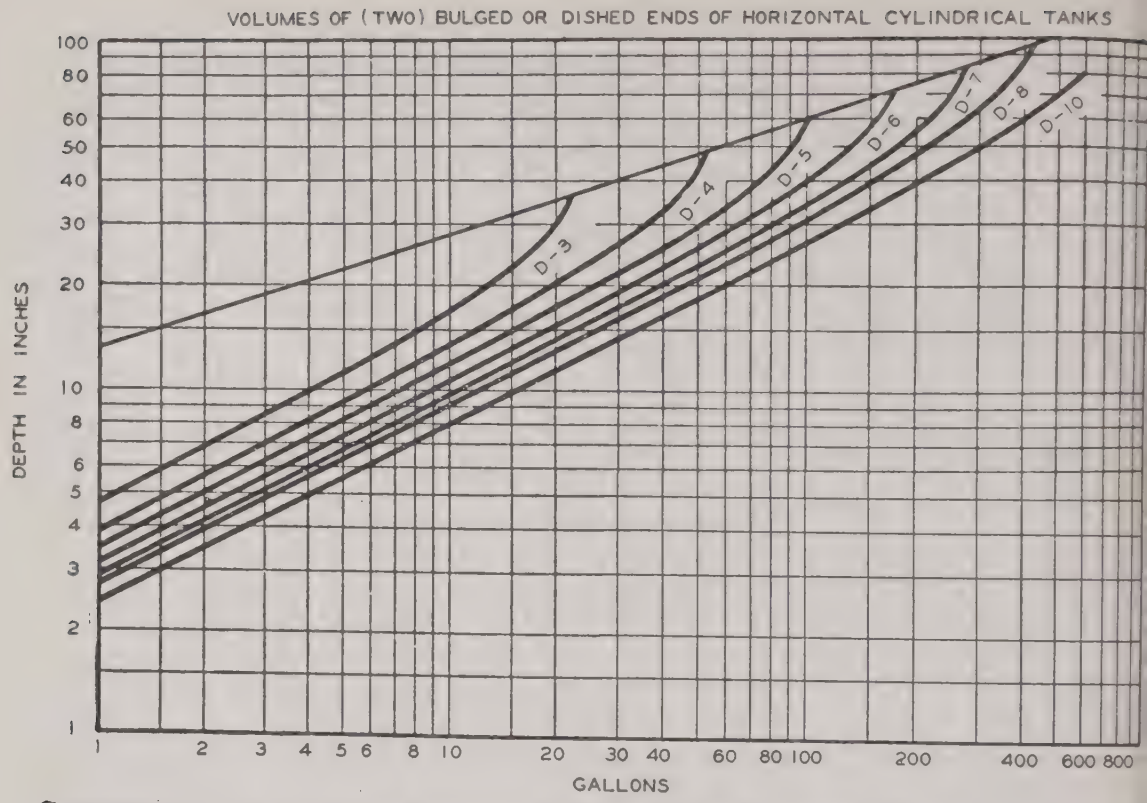
Table 6. Capacities of Horizontal Cylindrical Tanks in U. S. Gallons

Contents given for 1 ft of length

Flat ends

Diameter of tank, in.	Depth of liquid, in.																
	3	6	9	12	15	18	21	24	27	30	33	36	39	42	45	48	51
12	1.15	2.94	4.73	5.88													
18	1.45	3.86	6.61	9.36	11.77	13.22											
24	1.70	4.60	8.05	11.75	15.45	18.90	21.80	23.50									
30	1.91	5.23	9.27	13.72	18.36	23.00	27.45	31.49	34.81	36.72							
36	2.12	5.79	10.34	15.43	20.85	26.44	32.03	37.45	42.54	47.09	50.76	52.88					
42	2.28	6.31	11.31	16.97	23.07	29.46	35.99	42.51	48.90	55.00	60.65	65.66	69.69	71.97			
48	2.45	6.78	12.20	18.38	25.10	32.20	39.54	47.01	54.47	61.81	68.91	75.63	81.81	87.23	91.56	94.01	
54	2.60	7.22	13.04	19.68	26.97	34.72	42.80	51.08	59.49	67.90	76.18	84.26	92.01	99.30	105.94	111.76	116.38
60	2.75	7.64	13.82	20.91	28.72	37.06	45.82	54.88	64.11	73.45	82.78	92.01	101.07	109.83	118.17	125.98	133.07
66	2.89	8.04	14.56	22.07	30.37	39.28	48.65	58.39	68.42	78.59	88.87	99.14	109.31	119.34	129.08	138.45	147.36
72	3.02	8.42	15.26	23.17	31.92	41.36	51.32	61.71	72.45	83.41	94.54	105.76	116.98	128.11	139.07	149.81	160.20
78	3.15	8.78	15.94	24.21	33.41	43.34	53.86	64.87	76.27	87.97	99.90	111.97	124.12	136.27	148.34	160.27	171.97
84	3.26	9.12	16.57	25.24	34.85	45.24	56.29	67.87	79.91	92.30	104.98	117.85	130.87	143.95	157.03	170.05	182.92
90	3.43	9.46	17.20	26.20	36.21	47.05	58.61	70.75	83.39	96.43	109.81	123.46	137.28	151.22	165.25	179.27	193.21
96	3.50	9.79	17.80	27.13	37.52	48.81	60.84	73.52	86.73	100.39	114.44	128.79	143.40	158.17	173.07	188.01	202.95
102	3.61	10.10	18.37	28.01	39.00	50.49	62.99	76.18	89.94	104.20	118.89	133.92	149.25	164.81	180.53	196.36	212.25
108	3.71	10.39	18.94	28.90	40.03	52.14	65.09	78.74	93.04	107.87	123.17	138.87	154.89	171.19	187.71	204.37	221.15

Diameter of tank, in.	Depth of liquid, in.																
	54	57	60	63	66	69	72	75	78	81	84	87	90	93	96	102	108
12																	
18																	
24																	
30																	
36																	
42																	
48																	
54	118.98																
60	139.25	144.14	146.89														
66	155.66	163.17	169.69	174.84	177.73												
72	170.16	179.60	188.35	196.26	203.10	208.50	211.52										
78	183.37	194.38	204.90	214.83	224.03	232.30	239.46	245.09	248.24								
84	195.60	207.99	220.03	231.61	242.66	253.05	262.66	271.33	278.78	284.64	287.90						
90	207.03	220.68	234.06	247.10	259.74	271.88	283.44	294.28	304.29	313.29	321.03	327.06	330.49				
96	217.85	232.62	247.23	261.58	275.63	289.29	302.50	315.18	327.21	338.50	348.89	358.22	366.23	372.52	376.02		
102	228.12	243.95	259.67	275.23	290.56	305.59	320.28	334.54	348.30	361.49	373.99	385.48	396.47	406.11	414.38	424.48	
108	238.05	254.75	271.53	288.19	304.71	321.01	337.03	352.73	368.03	382.86	397.16	410.81	423.76	435.87	447.00	465.51	475.90



Constructed from Doolittle Formula. Ind. Eng. Chem., March, 1928.

V (both ends) $= 0.0009328h^2(3r - h)$ gallons.

h = depth in inches.

r = radius of tank in inches.

In curves D = diameter of tank in feet.

Table 7. Greek Alphabet

Greek letter	Greek name	English equivalent	Greek letter	Greek name	English equivalent
A α	Alpha	a	N ν	Nu	n
B β	Beta	b	Ξ ξ	Xi	x
Γ γ	Gamma	g	O o	Omicron	ŏ
Δ δ	Delta	d	Π π	Pi	p
E ϵ	Epsilon	ĕ	P ρ	Rho	r
Z ζ	Zeta	z	Σ σ	Sigma	s
H η	Eta	ē	T τ	Tau	t
Θ θ	Theta	th	Υ υ	Upsilon	u
I ι	Iota	i	Φ ϕ	Phi	ph
K κ	Kappa	k	X χ	Chi	ch
Λ λ	Lambda	l	Ψ ψ	Psi	ps
M μ	Mu	m	Ω ω	Omega	ō

14. CONVERSION TABLES

Equivalents of Measure

Use of all tables. To convert a known value to a desired measure, multiply by the known units equivalent.

Example: The wall of a room is given as 20.25 m; its length therefore must be $3.280333 \times 20.25 = 66.42674$ ft.

Notations: The following expressions will be used: $0^2\ 0^3\ 0^4\ 0^5$, etc., indicating the number of ciphers to be used after the decimal point.

Example: $1 \text{ sq rod} = 0.09766^5 = 0.000009766$ sq miles, or 1 grain $= 0.02083^2 = 0.002083$ oz troy. 1 grain $= 0.06480^4 = 0.00006480$ kg, or 1 kg/m^3 $= 0.03613^4 = 0.00003613$ lb/in^3.

Lengths

1 meter (m) = 10 decimeters (dm) = 100 centimeters (cm) = 1000 millimeters (mm) = 0.001 kilometer (km)
To convert metric units merely adjust the decimal point.
1 meter (m) = 39.37 in. (U. S. Standard) = 39.370113 in. (British Standard)
1 millimeter (mm) = 1000 microns (μ) = 0.03937 in. = 39.37 mils.
1 yard (U. S. Standard) = 1.0000029 yards (British standard)

Meters	Inches	Feet	Yard	Rods	Miles		Kilometers
					Statute	U. S. Nautical	
m	in.	ft	yd	4			km
1	39.37	3.280833	1.093611	0.198838	0.06214^3	0.05396^3	0.001
0.02540	1	0.08333	0.02778	0.05051	0.01578^4	0.01371^4	0.02540
0.30480	12	1	0.33333	0.06061	0.01894^3	0.01645^3	0.03048
0.91440	36	3	1	0.18182	0.05682^3	0.04934^3	0.09144
5.02921	198	16.5	5.5	1	0.03125^2	0.02714^2	0.05029
1,609.35	63,360	5,280	1,760	320	1	0.86839	1.60935
1,853.25	72,962.5	6,080.20	2,026.73	368.497	1.15155	1	1.85325
1,000	39,370	3,280.83	1,093.61	198.838	0.62137	0.53959	1

Surfaces and Areas

1 sq meter (m^2) = 100 sq decimeters (dm^2) = 10,000 sq centimeters (cm^2) = 0.0001 hectare (ha)
1 sq millimeter (mm^2) = 0.01 cm^2 = 0.00155 sq in. = 1,973.5 circular mils

Sq Meters m^2	Sq Inches sq in.	Sq Feet sq ft	Sq Yards sq yd	Acres A	Hectares ha	Sq Miles statute	Sq Kilometers km^2
				3		6	5
1	1549.99	10.7639	1.19599	0.02471	0.0001	0.03861	0.01
3		2	3	6	7	9	9
0.06452	1	0.06944	0.07716	0.01594	0.06452	0.02491	0.06452
				4	5	7	7
0.09290	144	1	0.11111	0.02296	0.09290	0.03587	0.09290
				3	4	6	6
0.83613	1,296	9	1	0.02066	0.08361	0.03228	0.08361
					2	5	4
						2	2
4,046.87	6,272,640	43,560	4,840	1	0.40469	0.01563	0.04047
						2	
10,000	15,499,969	107,639	11,959.9	2.47104	1	0.03861	0.01
2,589,999		27,878,400	3,097,600	640	259.000	1	2.59000
1,000,000		10,763,867	1,195,985	247.104	100	0.38610	1

1 sq rod, sq pole, or sq perch = 625 sq links = $\frac{1}{160}$ acre.

Volume and Capacity

1 cu meter (m^3) = 1,000 cu decimeters (dm^3) = 1,000,000 cu centimeters (cm^3)
1 liter (l) = 10 deciliters (dl) = 100 centiliters (cl) = 1,000 milliliters (ml) = 1,000 cu centimeters (cm^3 or cc)
1 liter (l) = 0.1 decaliter (dkl) = 0.01 hectoliter (hl) = 1 cu decimeter (dm^3)

Cubic Decimeters dm^3 1	Cubic Inches cu in.	Cubic Feet cu ft	Cubic Yard cu yd	U. S. Quarts		U. S. Gallons		U. S. Bushels bu
				liquid 1 qt	dry d qt	liquid 1 gal	dry d gal	
			2					
1	61.0234	0.035315	0.01308	1.05668	0.90808	0.264178	0.22702	0.02838
		3	4			2	2	3
0.01639	1	0.05787	0.02143	0.01732	0.01488	0.04329	0.03720	0.04650
28.3170	1,728	1	0.03704	29.9221	25.7140	7.48055	6.42851	0.80356
764.559	46,656	27	1	807.896	694.279	201.974	173.570	21.6962
			2					
0.94636	57.75	0.03342	0.01238	1	0.85937	0.25	0.21484	0.02686
			2					
1.10123	67.2006	0.03889	0.01440	1.16365	1	0.29091	0.25	0.03125
			2					
3.78543	231	0.13368	0.04951	4	3.43747	1	0.85937	0.10742
			2					
4.40492	268.803	0.15556	0.05761	4.65460	4	1.16365	1	0.125
35.2393	2,150.42	1.24446	0.04609	37.2368	32	9.30920	8	1
4.54374	277.274	0.160459	0.005942_9	4.30128	4.1267	1.20032	1.0317	0.12896

Bottom row lists the equivalents for one imperial gallon.

U. S. dry measure: 1 bushel = 4 pecks = 8 gallons = 32 quarts = 64 pints.
U. S. liquid measure: 1 gallon = 4 quarts = 8 pints = 32 gills = 128 fluid ounces
U. S. Apoth. measure: 1 fl. ounce (f$\overline{3}$) = 8 fl. drams (f$\overline{3}$) = 480 minims ($\overline{m}$) = 29.574 cm^3.
Example: For South America a pump has to be furnished of 1,000 liters per minute capacity. According to American practice this would be 1,000 × 0.264178 = 264.178 gpm.

Masses and Weights

1 gram (g) = 10 decigrams (dg) = 100 centigrams (cg) = 1,000 milligrams (mg) = 0.001 kilograms (kg)
1 kilogram (kg) = 1 liter or cu decimeter of water at 4 degrees C, 45° Lat. and sea level = 15,432.35639 grains.
1 U. S. liquid gal = 8.34545 lb, 1 Imperial gal = 10.0221 lb, 1 cu ft = 62.4283 lb water as above.

Kilograms kg	Grains gr	Ounces		Pounds		Tons		
		Troy oz t	Avoir oz av	Troy lb t	Avoir lb av	Net-short 2,000 lb	Gross-long 2,240 lb	Metric 1,000 kg
1	1,5432.4	32.1507	35.2740	2.67923	2.20462	0.01102	0.09842	0.001
0.06480	1	0.02083	0.02286	0.01736	0.01429	0.07143	0.06378	0.06480
0.03110	480	1	1.09714	0.08333	0.06857	0.03429	0.03061	0.03110
0.02835	473.5	0.91146	1	0.07595	0.06250	0.03125	0.02790	0.02835
0.37324	5,760	12	13.1657	1	0.82286	0.04114	0.03674	0.03732
0.45359	7,000	14.5833	16	1.21528	1	0.00050	0.04464	0.04536
907.185	14,000,000	29,166.7	32,000	2,430.56	2,000	1	0.89286	0.90719
1,016.05	15,680,000	32,666.7	35,840	2,722.22	2,240	1.12	1	1.01605
1,000	15,432,356	32,150.7	35,274.0	2,679.23	2,204.62	1.10231	0.98421	1

The pound, ounce and grain Apothecary have the same weight as those of Troy weight.
1 pound (Apoth.) = 12 ounces (℥) = 96 drams (ʒ) = 288 scruples (℈) = 5760 grains (gr)
1 pound (Troy) = 12 ounces (℥) = 240 pennyweight (dwt) = 5760 grains (gr)
1 hundredweight = 1/20 long ton = 4 quarters = 8 stone = 112 lbs = 50.8024 kg
Example: To make 10 metric tons of ice per day would require a plant with a capacity of 10 × 1.10231 = 11.023 ton American practice. When capacities in long tons are specified, then a 10-gross-ton plant requires 10 × 1.12 = 11.2 U. S. tons ice-making capacity.

Pressure

1 dyne per sq centimeter = 0.00101979 g/cm² = 0.000466646 poundals/in².
1 gram per sq centimeter = 980.5966 dynes/cm² = 0.457592 poundals/in².
1 poundal per sq in. = 2,142.95 dynes/cm² = 2.18536 g/cm² = 0.0310832 pound/in².

Kilograms per sq centimeter kg/cm²	Pounds per sq inch psi	Pounds per sq foot psf	Atmospheres, Standard 760 mm	Columns of Mercury Hg. 13.59593 sp g		Columns of Water, max density at 4C, 39F	
				Millimeters	Inches	Meters	Feet
1	14.2234	2,048.17	0.96778	735.514	28.9572	10	32.8083
0.07031	1	144	0.06804	51.7116	2.03588	0.70307	2.30665
0.04882	0.06944	1	0.04725	0.35911	0.01414	0.04882	0.01602
1.03329	14.6969	2,116.35	1	760	29.9212	10.3329	33.9006
0.01360	0.01934	2.78468	0.01316	1	0.03937	0.01360	0.04461
0.03453	0.49119	70.7310	0.03342	25.4001	1	0.34534	1.13299
0.10	1.42234	204.817	0.09678	73.5514	2.89572	1	3.28083
0.03048	0.43353	62.4283	0.02950	22.4185	0.88262	0.30480	1

Density—Specific Weights

1 dyne per cu centimeter = 0.00101979 gram/cm³ = 0.00118528 poundals/in³.
1 gram per cu centimeter = 980.5966 dynes/cm³ = 1.162283 poundals/in³.
1 poundal per cu in. = 843.683 dynes/cm³ = 0.860378 g/cm³ = 0.0310832 pound/in³.

Grams per cu centimeter g/cm³	Pounds per cu inch lb/in³	Pounds per cu foot lb/ft³	Pounds per cu yard lb/yd³	Kilograms per cu meter kg/m³	Pounds per bushel U. S.	Pounds per gallon, dry, U. S.	Pounds per gallon, liquid, U. S.
1	0.03613	62.4283	1,685.56	1,000	77.6893	9.71116	8.34545
27.6797	1	1,728	46,656	2,7679.7	2,150.42	268.803	231
0.01602	0.05787	1	27	16.0184	1.24446	0.15556	0.13368
0.05933	0.02143	0.03704	1	0.59327	0.04609	0.05762	0.04951
0.001	0.03613	0.06243	1.68556	1	0.07769	0.09711	0.08345
0.01287	0.04650	0.80356	21.6962	12.8718	1	0.125	0.10742
0.010297	0.03720	6.42851	173.570	102.974	8	1	0.85937
0.11983	0.04329	7.48052	201.974	119.826	9.30920	1.16365	1

The specific gravity of a substance is the ratio of its density to that of water at 4C, both densitie being expressed in the same units.
The density of a substance is its mass per unit volume.
The specific weight of a substance is its weight per unit volume.

Heat, Energy, Work

1 dyne-centimeter = 1 erg = 0.00101979 gram-centimeter = 0.0737612 foot-pound.
1 gram-centimeter = 980.5966 ergs = 0.07233 foot-pounds.
1 foot-pound, ft-lb = 13,557,300 ergs = 13,825.5 gram-centimeter.

Kilogram meters kg-m	Foot pounds ft-lb	Horsepower—Hours		Kilowatt hours kw-hr	Joules 10^7 ergs j	Thermal Units	
		U. S. hp-hr	Metric 75 kg-m-hr			Btu	kg-cal
1	7.23300	0.03653	0.03704	0.02724	9.80597	0.09296	0.02342
0.13826	1	0.05051	0.05121	0.03766	1.35573	0.01285	0.03239
273,745	1,980,000	1	1.01387	0.74565	2,684,340	2,544.65	641.240
270,000	1,952,910	0.98632	1	0.73545	2,647,610	2,509.83	632.467
367,123	2,655,403	1.34111	1.35972	1	3,600,000	3,412.66	859.975
0.10198	0.73761	0.03725	0.03777	0.02778	1	0.09480	0.02389
107.577	778.104	0.03930	0.03984	0.02930	1,054.90	1	0.25200
426.900	3,087.77	0.01559	0.01581	0.01163	4,186.17	3.96832	1

Power, Rate of Energy and Heat

1 erg per sec = 1 dyne-cm/sec = 0.0737612×10^{-7} foot-pound/sec.

1 gram-centimeter per second = 980.5966 ergs/sec = 0.07238×10^{-4} foot-pound/sec.

1 foot-pound per second = 13,557,300 ergs/sec = 13,825.5 gram-cm/sec.

Kilogram Meters per second kg-m/sec	Foot-pounds per second ft-lb/sec	Horsepower U. S. 550 ft-lb/sec	Horsepower Metric 75 kg-m/sec	Kilowatt kw	Watts 10^7 ergs/sec	Thermal Units per Sec Btu/sec	Thermal Units per Sec kg-cal/sec
1	7.23300	0.01315	0.01333	0.09806×10^{-2}	9.80597	0.09296×10^{-2}	0.02342×10^{-2}
0.13826	1	0.01818	0.01843	0.01356×10^{-2}	1.35573	0.01285×10^{-2}	0.03237×10^{-3}
76.0404	550	1	1.01387	0.74565	745.650	0.70685	0.17812
75	542.475	0.98632	1	0.73545	735.448	0.69718	0.17569
100	723.300	1.31509	1.33333	0.98060	980.597	0.92957	0.23425
101.979	737.612	1.34111	1.35972	1	1,000	0.94796	0.23888
0.10198	0.73761	0.01341×10^{-2}	0.01360×10^{-2}	0.001	1	0.09480×10^{-3}	0.02389×10^{-3}
107.577	778.104	1.41474	1.43436	1.05490	1,054.90	1	0.25200
426.900	3,087.77	5.61412	5.69200	4.18617	4,186.17	3.96832	1

The ton refrigeration is 288,000 Btu/24 hr = 12,000 Btu/hr = 200 Btu/min = 3.3333 Btu/sec.

Example: To absorb the heat developed in expending the energy of one hp/sec, $\frac{0.70685}{3.3333} = 0.21206$ ton refrigeration capacity is required.

Velocities and Accelerations

1 centimeter per sec = 0.0328083 ft per sec.

1 radian per second = 57.2958 degrees per sec = 0.159155 revolution per sec.

1 gravity = 980.5966 centimeters per sec per sec = 32.1717 ft per sec per sec.

Meters per second mps	Feet per second fps	Miles per hour mph	Knots U. S.	Kilometers per hour	Meters per sec/sec mps^2	Feet per sec/sec fps^2	Miles per hour/sec mph-s
1	3.28083	2.23693	1.94254	3.6	1	0.28083	2.23693
0.30480	1	0.68182	0.59209	1.09728	0.30480	1	0.68182
0.44704	1.46667	1	0.86839	1.60935	0.44704	1.46667	1
0.51479	1.68894	1.15155	1	1.85325	0.27778*	0.91134*	0.62137*
0.27778	0.91134	0.62137	0.53959	1			

* Equivalents for 1 Kilometer per hour per sec.

Temperature Conversion Table

Deg F	Deg C	Deg F	Deg C	Deg F	Deg C	Deg F	Deg C
−120.	−84.45	−77.80	−61.	−35.	−37.22	8.	−13.33
−119.20	−84.	−77.	−60.56	−34.60	−37.	8.60	−13.
−119.	−83.89	−76.	−60.	−34.	−36.67	9.	−12.78
−118.	−83.33	−75.	−59.44	−33.	−36.11	10.	−12.22
−117.40	−83.	−74.20	−59.	−32.80	−36.	10.40	−12.
−117.	−82.78	−74.	−58.89	−32.	−35.56	11.	−11.67
−116.	−82.22	−73.	−58.33	−31.	−35.	12.	−11.11
−115.60	−82.	−72.40	−58.	−30.	−34.44	12.20	−11.
−115.	−81.67	−72.	−57.78	−29.20	−34.	13.	−10.56
−114.	−81.11	−71.	−57.22	−29.	−33.89	14.	−10.
−113.80	−81.	−70.60	−57.	−28.	−33.33	15.	− 9.44
−113.	−80.56	−70.	−56.67	−27.40	−33.	15.80	− 9.
−112.	−80.			−27.	−32.78	16.	− 8.89
−111.	−79.45	−69.	−56.11	−26.	−32.22	17.	− 8.33
−110.20	−79.	−68.80	−56.	−25.60	−32.	17.60	− 8.
−110.	−78.89	−68.	−55.56	−25.	−31.67	18.	− 7.78
−109.	−78.33	−67.	−55.	−24.	−31.11	19.	− 7.22
−108.40	−78.	−66.	−54.44	−23.80	−31.	19.40	− 7.
−108.	−77.78	−65.20	−54.	−23.	−30.56	20.	− 6.67
−107.	−77.11	−65.	−53.89	−22.	−30.	21.	− 6.11
−106.60	−77.	−64.	−53.33	−21.	−29.44	21.20	− 6.
−106.	−76.67	−63.40	−53.	−20.20	−29.	22.	− 5.56
−105.	−76.11	−63.	−52.78	−20.	−28.89	23.	− 5.
−104.80	−76.	−62.	−52.22	−19.	−28.33	24.	− 4.44
−104.	−75.56	−61.60	−52.	−18.40	−28.	24.80	− 4.
−103.	−75.	−61.	−51.67	−18.	−27.78	25.	− 3.89
−102.	−74.45	−60.	−51.11	−17.	−27.22	26.	− 3.33
−101.20	−74.	−59.80	−51.	−16.60	−27.	26.60	− 3.
−101.	−73.89	−59.	−50.56	−16.	−26.67	27.	− 2.78
−100.	−77.33	−58.	−50.	−15.	−26.11	28.	− 2.22
− 99.40	−73.	−57.	−49.44	−14.80	−26.	28.40	− 2.
− 99.	−72.78	−56.20	−49.	−14.	−25.56	29.	− 1.67
− 98.	−72.22	−56.	−48.89	−13.	−25.	30.	− 1.11
− 97.60	−72.	−55.	−48.33	−12.	−24.44	30.20	− 1.
− 97.	−71.67	−54.40	−48.	−11.20	−24.	31.	− 0.56
− 96.	−71.11	−54.	−47.78	−11.	−23.89	32.	0.
− 95.80	−71.	−53.	−47.22	−10.	−23.33	33.	0.56
− 95.	−70.56	−52.60	−47.	− 9.40	−23.	33.80	1.
− 94.	−70.	−52.	−46.67	− 9.	−22.78	34.	1.11
− 93.	−69.45	−51.	−46.11	− 8.	−22.22	35.	1.67
− 92.20	−69.	−50.80	−46.	− 7.60	−22.	35.60	2.
− 92.	−68.89	−50.	−45.56	− 7.	−21.67	36.	2.22
− 91.	−68.33	−49.	−45.	− 6.	−21.11	37.	2.78
− 90.40	−68.	−48.	−44.44	− 5.80	−21.	37.40	3.
−90.	−67.78	−47.20	−44.	− 5.	−20.56	38.	3.33
−89.	−67.22	−47.	−43.89	− 4.	−20.	39.	3.89
−88.60	−67.	−46.	−43.33	− 3.	−19.44	39.20	4.
−88.	−66.67	−45.30	−43.	− 2.20	−19.	40.	4.44
−87.	−66.11	−45.	−42.78	− 2.	−18.89	41.	5.
−86.80	−66.	−44.	−42.22	− 1.	−18.33	42.	5.56
−86.	−65.56	−43.60	−42.	− 0.40	−18.	42.80	6.
−85.	−65.	−43.	−41.67	0	−17.78	43.	6.11
−84.	−64.44	−42.	−41.11	.1	−17.22	44.	6.67
−83.20	−64.	−41.80	−41.	1.40	−17.	44.60	7.
−83.	−63.89	−41.	−40.56	2.	−16.67	45.	7.22
−82.	−63.33	−40.	−40.	3.	−16.11	46.	7.78
−81.40	−63.	−39.	−39.44	3.20	−16.	46.40	8.
−81.	−62.78	−38.20	−39.	4.	−15.56	47.	8.33
−80.	−62.22	−38.	−38.89	5.	−15.	48.	8.89
−79.60	−62.	−37.	−38.33	6.	−14.44	48.20	9.
−79.	−61.67	−36.40	−38.	6.80	−14.	49.	9.44
−78.	−61.11	−36.	−37.78	7.	−13.89	50.	10.

Temperature Conversion Table (Concluded)

Deg F	Deg C	Deg F	Deg C	Deg F	Deg C	Deg F	Deg C
51.	10.56	91.40	33.	132.80	56.	173.	78.33
51.80	11.	92.	33.33	133.	56.11	174.	78.89
52.	11.11	93.	33.89	134.	56.67	174.20	79.
53.	11.67	93.20	34.	134.60	57.	175.	79.44
53.60	12.	94.	34.44	135.	57.22	176.	80.
54.	12.22	95.	35.	136.	57.78	177.	80.56
55.	12.78	96.	35.56	136.40	58.	177.80	81.
55.40	13.	96.80	36.	137.	58.33	178.	81.11
56.	13.33	97.	36.11	138.	58.89	179.	81.67
57.	13.89	98.	36.67	138.20	59.	179.60	82.
57.20	14.	98.60	37.	139.	59.44	180.	82.22
58.	14.44	99.	37.22	140.	60.	181.	82.78
59.	15.	100.	37.78	141.	60.56	181.40	83.
60.	15.56	100.40	38.	141.80	61.	182.	83.33
60.80	16.	101.	38.33	142.	61.11	183.	83.89
61.	16.11	102.	38.89	143.	61.67	183.20	84.
62.	16.67	102.80	39.	143.60	62.	184.	84.44
62.60	17.	103.	39.44	144.	62.22	185.	85.
63.	17.22	104.	40.	145.	62.78	186.	85.56
64.	17.78	105.	40.56	145.40	63.	186.80	86.
64.40	18.	105.80	41.	146.	63.33	187.	86.11
65.	18.33	106.	41.11	147.	63.89	188.	86.67
66.	18.89	107.	41.67	147.20	64.	188.60	87.
66.20	19.	107.60	42.	148.	64.44	189.	87.22
67.	19.44	108.	42.22	149.	65.	190.	87.78
68.	20.	109.	42.78	150.	65.56	190.40	88.
69.	20.56	109.40	43.	150.80	66.	191.	88.33
69.80	21.	110.	43.33	151.	66.11	192.	88.89
70.	21.11	111.	43.89	152.	66.67	192.20	89.
71.	21.67	111.20	44.	152.60	67.	193.	89.44
71.60	22.	112.	44.44	153.	67.22	194.	90.
72.	22.22	113.	45.	154.	67.78	195.	90.56
73.	22.78	114.	45.56	154.40	68.	195.80	91.
73.40	23.	114.80	46.	155.	68.33	196.	91.11
74.	23.33	115.	46.11	156.	68.89	197.	91.67
75.	23.89	116.	46.67	156.20	69.	197.60	92.
75.20	24.	116.60	47.	157.	69.44	198.	92.22
76.	24.44	117.	47.22	158.	70.	199.	92.78
77.	25.	118.	47.78	159.	70.56	199.40	93.
78.	25.56	118.40	48.	159.80	71.	200.	93.33
78.80	26.	119.	48.33	160.	71.11	201.	93.89
79.	26.11	120.	48.89	161.	71.67	201.20	94.
80.	26.67	120.20	49.	161.60	72.	202.	94.44
80.60	27.	121.	49.44	162.	72.22	203.	95.
81.	27.22	122.	50.	163.	72.78	204.	95.56
82.	27.78	123.	50.56	163.40	73.	204.80	96.
82.40	28.	123.80	51.	164.	73.33	205.	96.11
83.	28.33	124.	51.11	165.	73.89	206.	96.67
84.	28.89	125.	51.67	165.20	74.	206.60	97.
84.20	29.	125.60	52.	166.	74.44	207.	97.22
85.	29.44	126.	52.22	167.	75.	208.	97.78
86.	30.	127.	52.78	168.	75.56	208.40	98.
87.	30.56	127.40	53.	168.80	76.	209.	98.33
87.80	31.	128.	53.33	169.	76.11	210.	98.89
88.	31.11	129.	53.89	170.	76.67	210.20	99.
89.	31.67	129.20	54.	170.60	77.	211.	99.44
89.60	32.	130.	54.44	171.	77.22	212.	100.
90.	32.22	131.	55.	172.	77.78	Deg $C=\frac{5}{9}(F-32)$;	
91.	32.78	132.	55.56	172.40	78.	Deg $F=32+\frac{9}{5}C$	

Viscosity

Dynamic or Absolute Viscosity, μ (mu)

gm/cm sec or dyne-sec/cm^2 Poise	gm/cm sec or dyne-sec/cm^2 Centipoises	lb/ft sec or poundal sec/ft^2	Slug/ft sec lb$_f$ sec/ft^2	kg/m hr
			2	
1	100	0.0672	0.0209	360
14.88	1488	1	0.03105	5356
478.5	47,850	32.174	1	172,300
2		3	5	
0.02778	0.2778	0.01867	0.05804	1

Kinematic viscosity $\nu = \mu/\rho$ where μ is the dynamic or absolute viscosity and ρ is the density of the fluid. Kinematic viscosity is termed a centistoke in the metric system.

Relative viscosity is the ratio of dynamic viscosity of the fluid to that of water at a definite reference temperature, usually 20 C or 68 F, where the viscosity of water is practically 1/100 poise or 1 centipoise.

Specific viscosity: Saybolt is a very common method of measuring and stating viscosity and the results stated in Saybolt seconds. Since the rate of flow through the nozzle of the Saybolt viscometer depends on the dynamic viscosity and the head pressure, and since the pressure head is a function of the density of the fluid, Saybolt viscosity is a function of the kinematic viscosity.

$$\nu = 0.0022 \times t - \frac{1.8}{t} \text{ poise/gram,}$$

$$\nu = \left(0.0022 \times t \times \rho - \frac{1.8}{t}\rho\right) \text{ poise,}$$

Also

$$\nu = 0.00000237t - \frac{0.00194}{t} \text{ sq ft per sec}$$

Some commercial measurements of viscosity are made with the Engler viscometer and the results are stated in Engler seconds.

$$\nu = 0.00147t - \frac{3.74}{t} \text{ poise/gram}$$

$$\nu = 0.00147t\rho - 3.74\frac{\rho}{t} \text{ poise}$$

where μ, ν and ρ are as above, and t is the time in Saybolt seconds or Engler seconds for the respective methods.

Enthalpy

1 kg-cal/kg = 1.8 Btu/lb
1 Btu/lb = 0.555 kg-cal/kg

Entropy

1 kg-cal/kg deg C = 1 Btu/lb deg F

Thermal Conductivities

$\frac{\text{Btu ft}}{\text{hr ft}^2\text{ F}}$*	$\frac{\text{Btu in.}}{\text{hr ft}^2\text{ F}}$	$\frac{\text{Chu in.}}{\text{hr ft}^2\text{ C}}$†	$\frac{\text{g cal cm}}{\text{sec cm}^2\text{ C}}$	$\frac{\text{kilo ergs}}{\text{sec cm}^2\text{ C}}$	$\frac{\text{w cm}}{\text{cm}^2\text{ C}}$	$\frac{\text{kg cal m}}{\text{hr m}^2\text{ C}}$
1.000	12.000	12.000	0.0413_2	173.	0.0173_2	1.49
0.0833	1.000	1.000	0.0344_3	14.4	0.0144_2	0.124
0.0833	1.000	1.000	0.0344_3	14.4	0.0144_2	0.124
242.	2,905.	2,905.	1.000	41,900.	4.19	361.
0.0578_2	0.0694	0.0694	0.0239_4	1.000	0.01_3	0.0861_2
57.8	694.	694.	0.239	10,000.	1.000	86.1
0.671	8.05	8.05	0.0277_2	116.	0.0116	1.000

* Read: Btu foot per hour square foot degree Fahrenheit.
† The pound-centigrade unit, PCu, or centigrade heat unit Chu, is the heat required to raise one pound of water one degree centigrade and equals 1.8 Btu.

Coefficients of Heat Transfer

$\frac{\text{Btu}}{\text{hr ft}^2\text{ F}}$	$\frac{\text{kg-cal}}{\text{hr m}^2\text{ C}}$	$\frac{\text{Chu}}{\text{hr ft}^2\text{ C}}$	$\frac{\text{g-cal}}{\text{sec cm}^2\text{ C}}$	$\frac{\text{Watts}}{\text{cm}^2\text{ C}}$	$\frac{\text{Watts}}{\text{m}^2\text{ F}}$	$\frac{\text{hp}}{\text{ft}^2\text{ F}}$
1.000	4.88	1.000	0.01355_3	0.0568_3	0.0204_2	0.0394_3
.205	1.000	.205	0.0278_4	0.0116_3	0.0418_3	0.0807_4
1.000	4.88	1.000	0.01355_3	0.0568_3	0.0204_2	0.0394_3
7,380.	36,000.	7,380.	1.000	4.19	15.1	2.91
1,761.	8,592.	1,761.	0.239	1.000	3.697	0.694
490.	2,392.	490.	0.0664	0.278	1.000	0.193
2,538.	12,390.	2,538.	0.344	1.44	5.18	1.000

Absolute Pressure and Equivalent Vacuum Values

Inches of mercury at the density of 32 F

psia	in. Hg vac	psia	in. Hg vac	psia	in. Hg vac	psia	in. Hg vac
0	29.921	3.6	22.590	7.2	15.261	10.767	8.0
0.1	29.716	3.645	22.5	7.3	15.057	10.8	7.932
0.2	29.512	3.7	22.387	7.329	15.0	10.9	7.728
0.206	29.5	3.8	22.183	7.4	14.854	11.0	7.525
0.3	29.309	3.890	22.0	7.5	14.650	11.013	7.5
0.4	29.105	3.9	21.979	7.574	14.5	11.1	7.321
0.452	29.0	4.0	21.776	7.6	14.447	11.2	7.117
0.5	28.901	4.1	21.572	7.7	14.243	11.258	7.0
0.6	28.698	4.136	21.5	7.8	14.039	11.3	6.914
0.698	28.5	4.2	21.369	7.820	14.0	11.4	6.710
0.7	28.494	4.3	21.165	7.9	13.836	11.5	6.507
0.8	28.291	4.381	21.0	8.0	13.632	11.504	6.5
0.9	28.087	4.4	20.961	8.065	13.5	11.6	6.303
0.943	28.0	4.5	20.758	8.1	13.429	11.7	6.100
1.0	27.883	4.6	20.554	8.2	13.225	11.749	6.0
1.1	27.680	4.627	20.5	8.3	13.022	11.8	5.896
1.189	27.5	4.7	20.351	8.311	13.0	11.9	5.692
1.2	27.476	4.8	20.147	8.4	12.818	11.995	5.5
1.3	27.273	4.873	20.0	8.5	12.614	12.0	5.489
1.4	27.069	4.9	19.944	8.557	12.5	12.1	5.285
1.434	27.0	5.0	19.740	8.6	12.411	12.2	5.082
1.5	26.866	5.1	19.536	8.7	12.207	12.241	5.0
1.6	26.662	5.118	19.5	8.8	12.044	12.3	4.878
1.680	26.5	5.2	19.333	8.802	12.0	12.4	4.674
1.7	26.458	5.3	19.129	8.9	11.780	12.486	4.5
1.8	26.255	5.364	19.0	9.0	11.596	12.5	4.471
1.9	26.051	5.4	18.926	9.048	11.5	12.6	4.267
1.926	26.0	5.5	18.722	9.1	11.393	12.7	4.064
2.0	25.848	5.6	18.518	9.2	11.189	12.732	4.0
2.1	25.644	5.609	18.5	9.293	11.0	12.8	3.860
2.171	25.5	5.7	18.315	9.3	10.986	12.9	3.656
2.2	25.440	5.8	18.111	9.4	10.782	12.977	3.5
2.3	25.237	5.855	18.0	9.5	10.578	13.0	3.453
2.4	25.033	5.9	17.908	9.539	10.5	13.1	3.249
2.417	25.0	6.0	17.704	9.6	10.375	13.2	3.046
2.5	24.830	6.1	17.500	9.7	10.171	13.223	3.0
2.6	24.626	6.101	17.5	9.785	10.0	13.3	2.842
2.662	24.5	6.2	17.297	9.8	9.968	13.4	2.639
2.7	24.422	6.3	17.093	9.9	9.764	13.468	2.5
2.8	24.219	6.346	17.0	10.0	9.561	13.5	2.435
2.9	24.015	6.4	16.890	10.030	9.5	13.6	2.231
2.908	24.0	6.5	16.686	10.1	9.357	13.7	2.028
3.0	23.812	6.592	16.5	10.2	9.153	13.714	2.0
3.1	23.608	6.6	16.483	10.276	9.0	13.8	1.824
3.153	23.5	6.7	16.279	10.3	8.950	13.9	1.621
3.2	23.405	6.8	16.075	10.4	8.746	13.960	1.5
3.3	23.201	6.837	16.0	10.5	8.543	14.0	1.417
3.399	23.0	6.9	15.872	10.521	8.5	14.1	1.213
3.4	22.997	7.0	15.668	10.6	8.339	14.2	1.010
3.5	22.794	7.083	15.5	10.7	8.135	14.205	1.0
		7.1	15.864			14.3	0.8062
						14.4	0.6026
						14.45	0.5
						14.6	0.3990
						14.696	0

PART III

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15. INSULATION AND MOISTURE-PROOFING

Economic Factors

THE costs of insulating refrigerated rooms or pipes, vessels and other equipment are large enough to warrant careful consideration of type and quantity of material to use. **First cost** is important, but even more so is the **continuing cost** made up of operating expense, maintenance and repairs and depreciation.

In considering insulation of heated surfaces it is conventional to compare the cost of heat loss from bare surfaces with that from insulated ones, thus determining the sum economically available to offset the cost of insulation. Exactly the same approach is not possible for refrigeration as it is seldom practical to operate without any insulation. Similar basic analysis is appropriate however. As the amount or thickness of insulation increases its cost goes up but the cost of refrigeration decreases. The most economical thickness is therefore the point where the sum of these costs per year is least. **Fig. 1** illustrates this, where the annual costs of insulation and refrigeration are respectively curves N and M. (Mathematical form and derivation by McMillan shown in Appendix A.)

It is important to note that the thickness so determined may not be sufficient to withstand the conditions of service. Other factors may modify purely economic considerations.

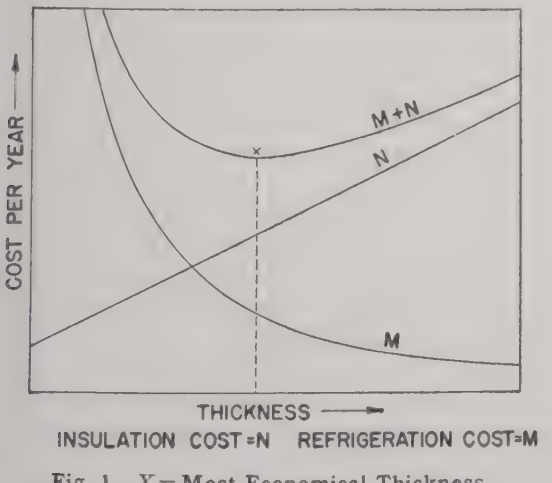


Fig. 1. X = Most Economical Thickness

Insulation Thickness in Practice

Extensive studies of all factors affecting the true economic thickness are warranted only for large permanent structures. The great majority of transient food storages held at 32 F to 40 F are insulated with 4 in. of sheet or block type insulation having a conductivity between 0.28 and 0.33 Btu per hr per sq ft per deg F per in. of thickness.

For other conditions, the thicknesses given in **Table 1** are in accord with general practice applying to the same type of insulation. Fill type insulations are generally used from ⅓ to ½ thicker. Manufacturers of different insulants have their own recommendations and will supply also information on conductivity, moisture resistance and other properties.

Table 1

Storage temp deg F	Insulation thickness in inches	
	Northern U. S.	Southern U. S.
−45 to −15	9	10
−15 to 0	7	8
0 to 15	6	7
15 to 25	5	6
25 to 35	4	5
35 to 50	3	4
50 to 60	2	3

Unusual conditions of climate or use should modify the normal recommendations. Roofs should have 1 in. more insulation than walls in northern latitudes and 2 in. more in the South because of the greater equivalent temperature difference where the roof is exposed to the sun. Floors, except in first-floor freezers, may have 1 in. less than walls.

One rule of thumb method of estimating load or overall heat loss and hence refrigerating capacity required, that should not be used unless properly modified by size factors, is "so many Btu per 100 cubic feet of capacity." Heat entry through insulation depends on total surface area times rate, and a cube of 1,000,000 cu ft has one-tenth the proportionate surface of a cube containing 1,000 cu ft.

Types of Structure

Three possible types of structure present as many problems for consideration:

1. Experimental construction, used only through the life of the test.
2. Temporary construction, useful up to five years.
3. Permanent construction, for the life of the structure housing the insulation.

Experimental construction obviously calls for the lowest possible investment. Failures due to condensation and freezing are unimportant, so long as they do not interfere with the experiment. For this purpose many fill type insulations, felted blankets and other cheap blocks are available. Wood framing and either wood, compo-board, or light metal sheathing may be employed and packed with the fill material.

Temporary construction including insulation should not be undertaken, unless low cost factors are paramount. For more than a brief period it will be found more costly in the end.

For **permanent construction,** insulating materials used should have lasting properties equal to those of the structure itself, which should be of brick and steel or concrete. If wood is used it must be specially protected against rot, vermin, and other destructive agencies.

Thermal Factors

As stated in Chapter 5, "Fundamental Concepts of Heat Transfer," **conductivities** of all insulations depend upon the density and vary directly with mean temperature and with respect to moisture content. As here used, **mean temperature** of the material is the arithmetic mean of surrounding air temperature and the operating temperature of the room or piece of equipment under consideration. Rowley, Jordan and Lander have reported[1] that conductivities of typical sheet or block type insulants decrease about 12.6% per 100 deg decrease in mean temperature. Except for a few chemical process and ultra-low temperature applications, the maximum difference in mean temperatures encountered will not exceed 50 deg F, so that the conductivity of insulation will not vary more than about 6% from one type of job to another.

There will be more variation than this in any carload of commercial insulation and still more is likely as a result of differing degrees of skill and care in applicators. Consequently there is no need in most cases to take the time to compute variation in conductivity due to mean temperature. Values determined at a mean of 50 or 60 F will be within 3 or 4% of accuracy for almost any job.

Variation in conductivity due to **moisture content** cannot be dismissed so easily. While there has not been general agreement on the extent to which conductivity is increased as the moisture content increases, there is full agreement that wet insulation is useless and a structural hazard. It is also agreed that unless properly protected against the entry of moisture, usually in the form of water vapor, nearly all insulations will soon become saturated, and structural damage either by corrosion or the expansive action of freezing water will follow.

The entry of water vapor into insulation is primarily the result of difference in pressure due to temperature. The temperature gradient through an insulated wall is proportional to the thermal resistance of the layers or lamina considered, in relation to the total resistance of all in series. This is shown in **Fig. 2,** from which it is apparent also that the place to put the protection against moisture, usually called the **vapor barrier,** is at the warm side of the insulation, since most of the material between that point and the cold surface is below the dew point. Vapor which gets through the barrier may, if its passage is not seriously impeded, continue traveling toward the region of still lower temperature

[1] *Refrig. Eng.*, Dec. 1945.

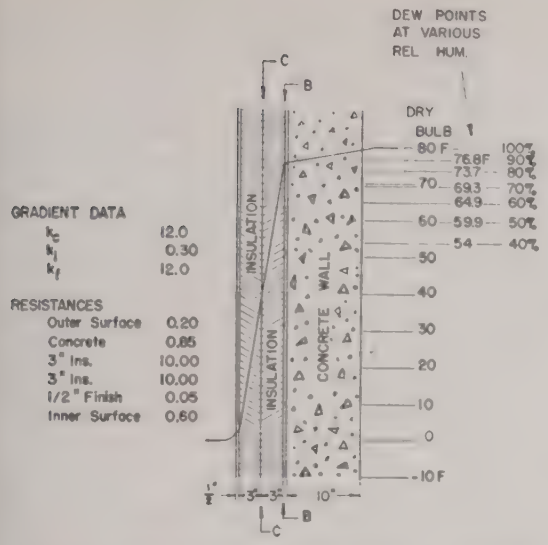


Fig. 2. Conventional Temperature Gradient. (Shows that vapor barrier at B-B hear warm side is preferable to barrier at C-C closer to cold side)

and pressure, and wind up as frost on the cooling surfaces. Whether it does so or collects in the insulation depends on the characteristics of the latter, the amount of vapor present and the presence or absence of more or less effective barriers. The only safe practice is to provide as good a seal or barrier on the warm side of the insulation as is practicable. It follows that the absence of vapor barriers or surfaces of low permeability between the warm side and the cold surface is desirable.

One type of installation that requires special treatment is an enclosure which is alternately warmer and cooler than its surroundings. If the temperature differences are large, and particularly if high humidities are involved, present state of knowledge indicates that light sheet metal surfacings with joints soldered or welded are required. In railroad car design there is a tendency toward assuming that because of vibratory strains, sealed surfaces will not remain tight, moisture will enter and must be allowed to escape.

All materials resist the passage of gases, yet all materials allow it to some degree. Moisture in the atmosphere performs as a gas, having the property of diffusing throughout the gases of the air, and through most materials at a fairly rapid rate. Moisture will diffuse and pass through barriers at a rate proportional to its vapor pressure and inversely proportional to the resistance of the barrier. **Vapor pressure** refers to the pressure of the moisture (or steam) only, independent of the air. The rate at which moisture will pass through some materials is shown on page 246.

Table 2. Vapor Pressure Values

Temperature, deg F	Water vapor pressure at saturation, lb/sq ft	Temperature, deg F	Water vapor pressure at saturation, lb/sq ft
−45	0.205	30	11.62
−40	0.268	35	14.40
−35	0.367	40	17.50
−30	0.496	45	21.25
−25	0.669	50	25.61
−20	0.892	55	30.82
−15	1.182	60	36.9
−10	1.562	65	43.9
− 5	2.05	70	52.3
0	2.67	75	61.8
5	3.46	80	73.0
10	4.45	85	85.7
15	5.72	90	100.7
20	7.26	95	117.2
25	9.22	100	136.7

From **Table 2** it is apparent that quite considerable pressures are involved in most refrigerated structures. The pressure differential between water vapor in the air at 80 F and 80% relative humidity and at 0 F and 85% relative humidity is slightly over 56 lb per sq ft.

Fig. 3, representing the same construction as in Fig. 2 but with two temperature gradients ((a) as in Fig. 2 for 6-in. insulation, with a conductivity of 0.27, and (b) for 6-in. insulation with a conductivity of 0.60), illustrates that within the conductivity range of the commercial insulants (0.27 to 0.45) the location of the vapor barrier need not change, even though there is a considerable change in the amount of heat flowing. However, if the structural part of the wall supplies enough of the total resistance to cause a temperature drop through it equal to or exceeding the assumed dew point depression (as in gradient b if relative humidity is 80%), then the vapor protection must be on its outside

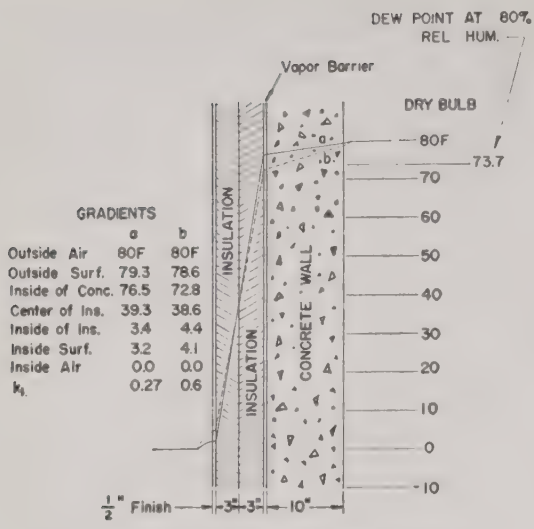


Fig. 3. Gradient Variation with Change in k of 120%

surface. For Fig. 3, the gradient figures and heat flow rates for both insulations and for the wall uninsulated are:

	Resistances			*Temperature Drops*		
	a *Ins.* ($k=.27$)	*b* *Ins.* ($k=.60$)	*c* *Unins.*	*a*	*b*	*c*
Outside surface	.20	.20	.20	0.7°F	1.4°F	9.8°F
Concrete wall 10/12	.85	.85	.85	2.8	5.8	41.2
6-in. insulation	22.2	10.00	—	74.3	68.4	—
½-in. cement finish	.05	.05	—	0.2	0.4	—
Inside surface	.60	.60	.60	2.0	4.0	29.0
Totals	23.9	11.7	1.65	80.0°F	80.0°F	80.0°F
Heat flow $=\Delta t/R$ = Btu/hr/sq ft	3.35	6.84	48.5	—	—	—

If only one point of the gradient needs to be determined, as for example the temperature of the cooler surface of the concrete wall in Fig. 3, a simple proportion will give it. If insulation of a conductivity of 0.45 is being considered ($R=15.0$ and the resistance of surface and wall is 1.05), then $80°—(1.05/15.0\times80)=74.4°$, which is just above the dew point for the assumed conditions.

Surface Condensation

To prevent condensation on the outside surface of insulated equipment or rooms requires the use of a sufficient thickness of insulation to insure that the temperature drop from the surrounding air to the insulation surface is less than the dew-point depression. For ordinary conditions this means about 1 in. of insulation for each 20 to 25 deg F of overall temperature difference.

The amount required varies considerably depending on the location. In humid climates more insulation is needed, as will be seen from **Fig. 4.** As an example, with the air temperature at 80 F, and 85% relative humidity, the dew-point depression is approximately 4.6°, while if the relative humidity is 60% the dew-point depression is about 14.5°. To get the minimum thickness of insulation for each condition, pick out the dew-point depression at left side of figure, run a line horizontally to the overall temperature difference line, say 80° if refrigeration temperature is 0 F, and drop to bottom of chart. For flat surfaces and 4.6° dew-point depression the thickness is 3.5 in., but for 14.5° dew-point depression it is just under 1 in.

One thing that cannot be done is to insulate heavily enough to prevent condensation under any conditions. To stop condensation at 100% relative humidity would require an infinite thickness of insulation, since the heat flow would have to be nil.

Foundations and Freezer Floors

For storage temperatures of 25 F and higher, foundations and areas beneath floors present no different problems than in non-refrigerated buildings. When the sub-floor is on the ground it is particularly desirable to use an insulation that absorbs a minimum of moisture, or provide thorough waterproofing beneath it. If freezer space is contemplated on or below grade, consideration should be given to the pos-

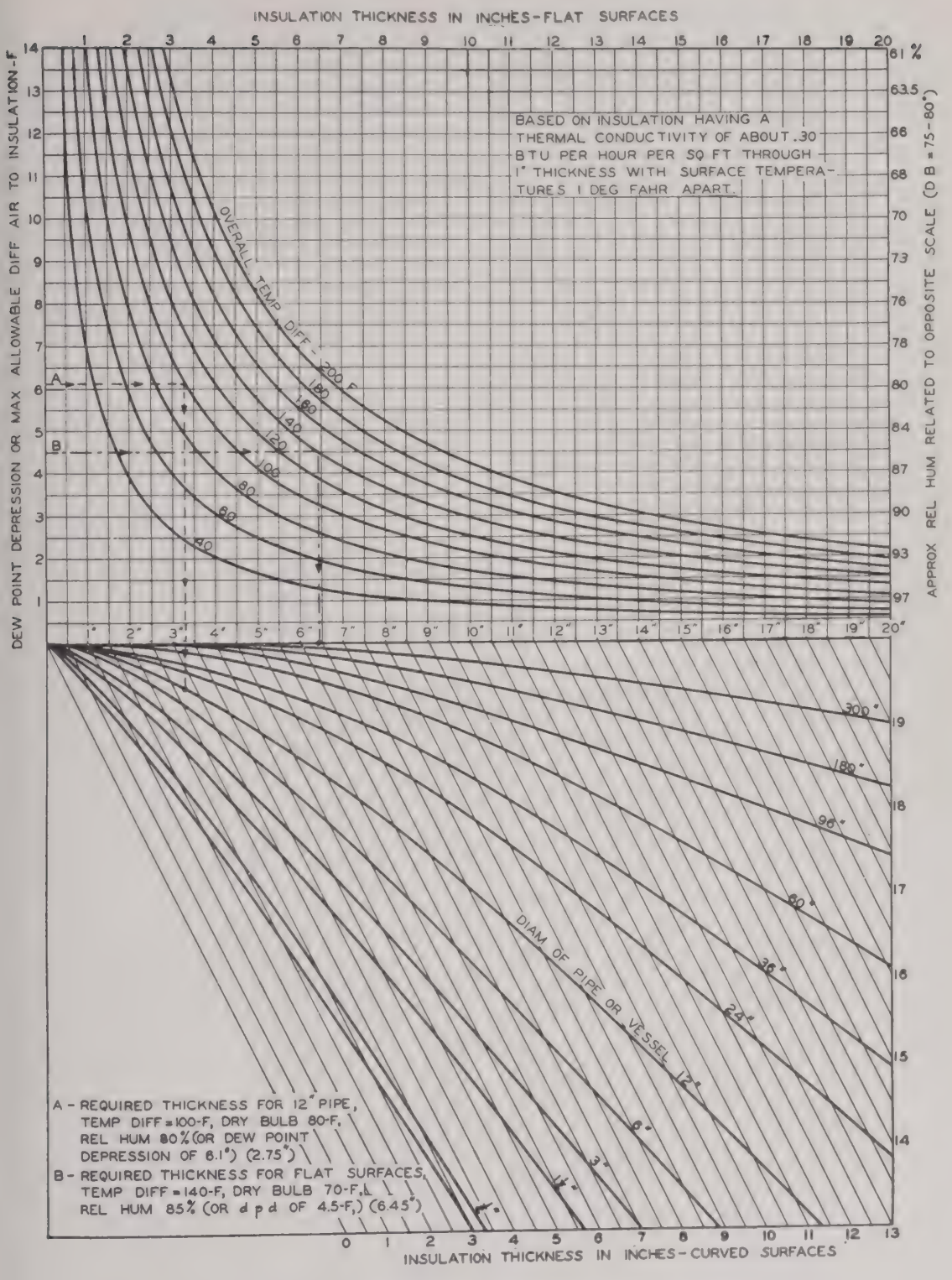


Fig. 4. Insulation Thickness to Prevent Surface Condensation

sibility of freezing the ground and heaving the floor or column and wall footings. According to Cooling and Ward[2] the risk of this happening becomes considerable with storage temperatures below 15 F, of course being greater as the temperatures go lower. The most unfavorable soils are of the fine-grained type such as fine sands, silts and clays.

Positive **drainage** is essential and must be provided for by the use of gravel or cinder fills plus drain tile, if there is any reason to fear seepage of ground water. For low-temperature freezing or holding rooms a depth of dry fill of 6 ft beneath the concrete floor slab is not excessive, although the use of heavier insulation on the floor would be a way of reducing the depth of fill. What is needed is to balance the heat subtracted from the ground beneath the floor with the heat flowing into the cooled area from the surrounding earth, so that a temperature of 32 F or lower is not reached in the ground beneath the dry fill. If it is, and there is moisture present, layers of ice form and build up in thickness, mainly drawing water from the surrounding area by capillary action.

In extreme cases where freezer rooms have to be on the lowest floor and the ground water level may approach the subfloor slab, the only way to be safe is to provide for supplying enough heat beneath the floor insulation to balance that flowing up into the cold room when the ground temperature is 33 to 35 F.

This can be done by making the subfloor slab a form of panel heating. Noncorroding pipe or tubing in the concrete carrying heated oil is being used by one engineer in the Mid-West to meet such conditions and seems a simple, effective and reliable method. Other suggestions include electric strip heaters and air channels beneath the slab carrying heated air. The designer confronted by such a problem will find it helpful to consult Professor C. F. Kayan's paper on "Temperatures and Heat Flow for a Concrete Slab with Imbedded Pipes."[3]

[2] L. F. Cooling, M.Sc., and W. H. Ward, B.Sc., A.C.G.I., A.M.I.C.E., Proceedings of the Institute of Refrigeration, (Great Britain) Vol. XLI. 1944–5.

[3] *Refrig. Eng.* 54, 2, 143 (Aug. 1947).

Piping and Vessels

The insulation of piping, tanks and vessels differs in one important respect from insulating rooms and buildings. In the case of the latter, the applicator is nearly always building inwards, that is, from what will be the warm surface of the insulation toward what will be the cold side. His problem is primarily to make vapor-tight, or nearly so, the surface of the first layer of insulation itself. Exactly the reverse is true on piping and tanks, and it is the outer finish or protection that will determine to a large degree the permanence of the job.

Only two general types of insulation have been used to date with any success on piping and vessels:

1. The relatively **solid type** made commercially in the form of semi-cylindrical sections for pipes and as bevelled lags and blocks for large pipes, vessels and tanks;

2. The comparatively soft, springy **blanket type** typified by hair felt. Recently there has been some use of felted mineral wool blankets, applied in much the same manner as hair felt.

Engineers designing a job, or owners buying insulation, must make their own choice between the two, for undeniably there have been first class installations of both types giving long continued good service. It is equally undeniable that there is considerably more difficulty in making sure of a good protective seal against moisture over the softer type of material.

Fittings are insulated either by covering them with molded or milled jackets of the same material as the sections of pipe insulation, or by wrapping them with one of the blanket type materials. In either case it is necessary to take particular precautions that the sealing be good where there are projections of metal through the insulation, such as valve stems, hangers, etc.

Table 3 gives heat transmission values for **solid sectional type insulation** having a nominal thermal conductivity of the order of 0.30 Btu per hr per sq ft of surface per deg F temperature difference between surfaces. **Table 4** shows minimum **spacing** that must be allowed to permit proper application and sealing of pipe insulation.

Table 3. Heat Transmission through Sectional Pipe Insulation

The rates of heat transmission given below are expressed in Btu per sq ft (and also per linear ft), per hr, per degree temperature difference between fluid in the pipe and air surrounding the pipe. In the case of screwed fittings, where the proportion of fittings to pipe is not unusually high, no great error will be introduced by figuring small fittings, such as ells and tees, as 1 linear ft of pipe; and larger fittings, valves, crosses, etc., as 1½ linear ft. Reference to a table of areas will be necessary in computing radiation equivalents for flanged fittings. Conductances in this table are based on an insulation conductivity of 0.30 Btu per sq ft hr deg F per in. of thickness.

Pipe size in.	Ice water thickness			Brine thickness			Special or heavy brine thickness		
	Btu per lin ft	Btu per sq ft pipe surface	Btu per sq ft outer surface	Btu per lin ft	Btu per sq ft pipe surface	Btu per sq ft outer surface	Btu per lin ft	Btu per sq ft pipe surface	Btu per sq ft outer surface
½	.110	.502	.105	.098	.446	.075	.087	.394	.051
¾	.119	.431	.101	.111	.405	.085	.094	.340	.051
1	.139	.403	.118	.124	.352	.086	.104	.294	.053
1¼	.155	.357	.118	.131	.300	.077	.113	.260	.054
1½	.174	.351	.133	.134	.270	.073	.118	.238	.053
2	.200	.322	.139	.151	.244	.077	.134	.214	.057
2½	.228	.303	.145	.170	.226	.081	.147	.197	.059
3	.269	.293	.158	.186	.202	.079	.162	.176	.059
3½	.295	.282	.161	.191	.183	.073	.176	.167	.061
4	.294	.248	.140	.209	.176	.076	.182	.154	.058
5	.349	.239	.148	.241	.165	.080	.202	.138	.057
6	.404	.233	.154	.259	.150	.076	.228	.130	.060
8	.455	.201	.139	.318	.140	.081	.263	.116	.059
10	.559	.198	.147	.383	.135	.086	.309	.110	.062
12	.648	.194	.150	.438	.131	.088	.364	.108	.067
14	.670	.182	.142	.487	.132	.093	.392	.107	.068
16	.750	.179	.143	.542	.130	.094	.434	.104	.069
18	.833	.176	.145	.604	.127	.096	.476	.102	.070
20	.919	.175	.146	.661	.126	.097	.520	.100	.071

Table 4

Insulation thickness	Space required between parallel pipes in.	Space required between pipe and adjacent surfaces in.
Ice Water Thickness (1½″ to 2″)		
Up to 6-in. pipe—screwed fittings	7	5
Larger than 6-in. pipe—screwed fittings	11	6
All pipe sizes—flanged fittings	12	8
Brine Thickness (2″ to 3″)		
Up to 6-in. pipe—screwed fittings	9	7
Larger than 6-in. pipe—screwed fittings	15	9
All pipe sizes—flanged fittings	16	10
Heavy Brine Thickness (3″ to 4″)		
Up to 3-in. pipe—screwed fittings	11	9
Larger than 3-in. pipe—screwed fittings	19	13
All pipe sizes—flanged fittings	20	14

The insulation of **water piping** required to prevent freezing when the pipe is exposed to low atmospheric temperatures is shown in **Table 5** for both types of insulation. It must be remembered that if there is no flow of water and the surrounding temperatures remain low, no amount of insulation will prevent freezing.

Insulation thicknesses to prevent **surface condensation** are shown in Fig. 4. To figure any special case it is necessary to know the dew-point depression in degrees

Table 5. Insulation to Prevent Freezing of Water in Pipes

No thickness of any type of insulation will prevent freezing if there is no flow of water and the surrounding temperature remains low long enough.

The two tables shown are based upon (A) use of sectional pipe insulation having a conductivity of approximately .31 Btu, and (B) hair felt or other flexible blanket having a conductivity of about .27 Btu.

Figures are based on air at −20 F, or a temperature difference of 62 deg, with entering water at 42 F. The time shown to lower water to 32 F is much less than would be needed to freeze all the water in the pipe, but when freezing starts the danger point has been reached.

Steel pipe size, in.	A				B			
	Insulation thickness	Btu per deg temp diff, per hr per lin ft	Hours to cool from 42 to 32 deg F—no flow	Lb water flow per hr per lin ft to prevent freezing	Insulation, no. of layers each nom. 1 in. thick	Btu per deg temp diff, per hr per lin ft	Hr to cool from 42 to 32 deg F—no flow	Lb water flow per hr per lin ft to prevent freezing
½	Ice Water	0.110	0.43	0.63	2	.0895	.417	.537
	Brine	0.098	0.52	0.56	3	.0747	.500	.448
	Heavy Brine	0.087	0.73	0.50	4	.0660	.565	.396
1	Ice Water	0.139	0.78	0.79	2	.1125	.825	.675
	Brine	0.124	0.90	0.71	3	.0911	1.02	.548
	Heavy Brine	0.104	1.20	0.59	4	.0798	1.16	.480
1½	Ice Water	0.174	1.27	0.98	2	.1400	1.40	.840
	Brine	0.134	1.72	0.76	3	.1126	1.74	.676
	Heavy Brine	0.118	2.04	0.67	4	.0972	2.02	.583
2	Ice Water	0.200	1.69	1.14	2	.1586	1.94	.952
	Brine	0.151	2.31	0.86	3	.1244	2.48	.747
	Heavy Brine	0.134	2.71	0.76	4	.1063	2.90	.638
3	Ice Water	0.269	2.73	1.53	2	.2062	3.25	1.237
	Brine	0.186	4.02	1.06	3	.1572	4.27	.943
	Heavy Brine	0.162	4.68	0.93	4	.1322	5.08	.793
4	Ice Water	0.294	4.08	1.68	2	.2450	4.55	1.470
	Brine	0.209	5.84	1.19	3	.1850	6.02	1.110
	Heavy Brine	0.182	6.77	1.04	4	.1548	7.20	.929
6	Ice Water	0.404	6.45	2.30	2	.3302	7.35	1.981
	Brine	0.259	10.01	1.48	3	.2434	9.88	1.460
	Heavy Brine	0.228	11.61	1.30	4	.1984	12.20	1.191
8	Ice Water	0.455	9.75	2.60	2	.4100	10.05	2.460
	Brine	0.318	14.00	1.82	3	.2960	13.90	1.776
	Heavy Brine	0.263	17.00	1.50	4	.2390	17.25	1.434
10	Ice Water	0.559	12.22	3.20	2	.4930	13.00	2.960
	Brine	0.383	17.95	2.20	3	.3536	18.10	2.122
	Heavy Brine	0.309	22.32	1.76	4	.2830	22.70	1.698
12	Ice Water	0.648	14.80	3.70	2	.5720	15.80	3.432
	Brine	0.438	22.12	2.50	3	.4090	22.20	2.454
	Heavy Brine	0.364	26.80	2.08	4	.3222	28.10	1.933

below the expected ambient, the overall temperature difference and the commercial conductivity of the insulation it is proposed to use. The surface resistance value can be taken as 0.65, which gives satisfactory results and provides a slight margin of safety in most cases. For flat or large diameter surfaces the equation is

$$x = 0.65k\left(\frac{t_a - t_o}{t_a - t_s} - 1\right),$$

where x = Thickness of insulation
k = Conductivity of insulation
t_a = Air temperature (ambient)
t_o = Temperature of surface to be insulated, frequently called operating temperature
t_s = allowable temperature of outer surface of insulation.

For pipes or vessels of moderate and small diameter there is no simple equation as the equivalent thickness, $r_2 \log_e (r_2/r_1)$ must be used in place of x. In this expression r_2 is the radius of the outer surface of the insulation and r_1 is the radius of the surface to be insulated.

The effect of diameter is small for small temperature differences but becomes considerable as $t_a - t_0$ increases. **Fig. 7** illustrates this for one value of $t_a - t_s$.

Several applications of insulation to tanks and vessels are shown in **Figs. 5** and **6**. Recommended methods vary somewhat

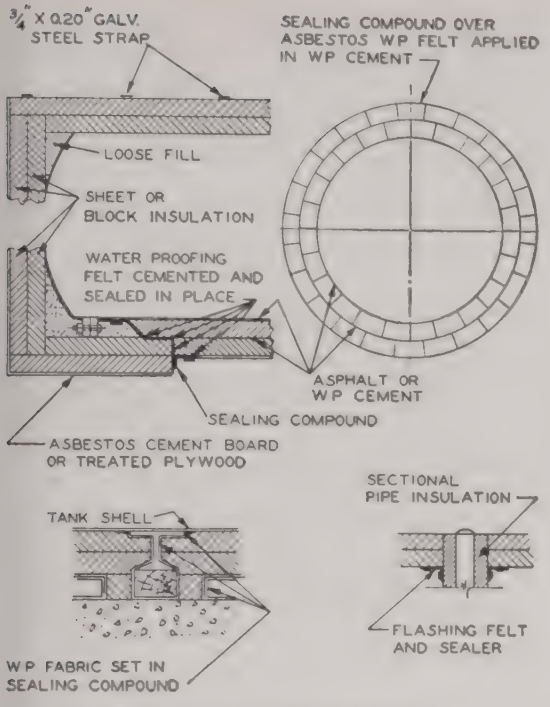


Fig. 5. Tank Insulation Details

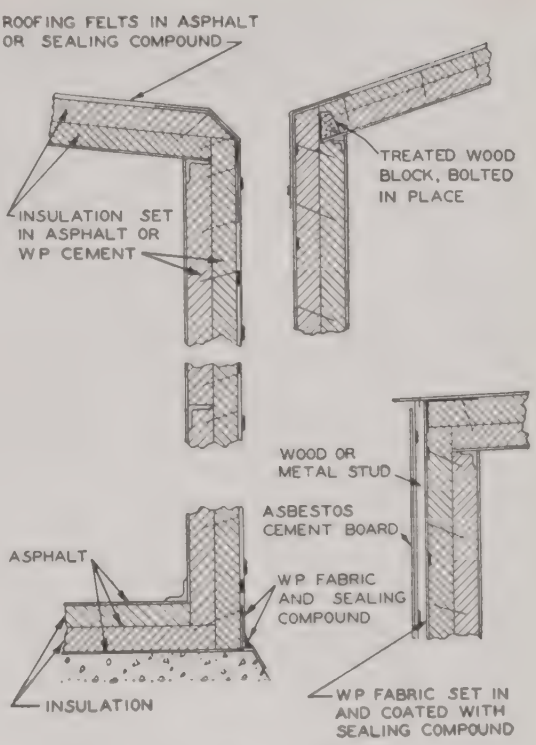


Fig. 6. Large Vertical Tank Insulation Details

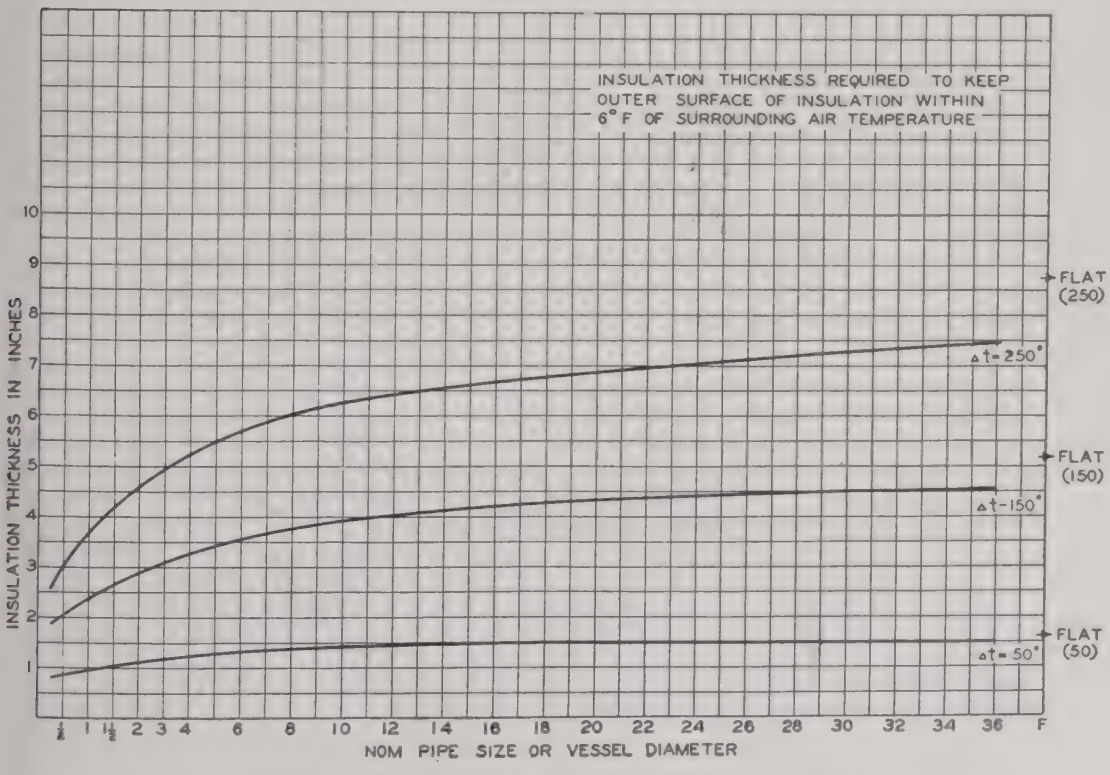


Fig. 7. Effect of Diameter on Insulation Thickness Required to Prevent Condensation

between manufacturers of different insulants but the examples shown are quite representative.

Application Specifications

Typical specifications are as follows:

The surface to be insulated shall be brushed free of all loose and foreign materials and shall be thoroughly dry. Rough surfaces shall be filled out with a coat of Portland cement plaster consisting of one part Portland cement and two and one half parts clean sharp, sand. When dry the surface shall be primed with a thin coat of asphalt primer. For each 10 ft of wall height, a horizontal support shall be bolted to the wall to carry weight of insulation above.

Insulation shall consist of — inches of ———— applied in — layers. Each sheet shall be dipped on back surface only in hot asphalt at a temperature of 350 to 400 F, and immediately pressed firmly into place. Adhesion shall be uniform over entire area.

The second layer is to be similarly applied to wall over the first layer, with all joints staggered in relation to each other. The sheets may be additionally secured to each other by wood skewers and with galvanized nails through caps to treated wood grounds bolted to the wall.

All joints must be tightly butted and all voids eliminated. Surfaces of the insulation shall be made smooth to receive finish.

Finish over the exposed surface of insulation shall be a Portland cement finish, applied in two coats. The first coat shall be thin and well scratched. The second coat shall be float finished and blocked off in squares of suitable size.

Appendix A

Derivation of McMillan's formula for economic thickness.

Let cost of refrigeration or heat entry equal M, and

$$M = \frac{a}{\frac{x}{k} + R}$$

where

a = hours operation per year $\times$ Δt $\times$ cost of refrig per ton day / 288,000

x = Thickness of insulation in inches

k = Conductivity of insulation in Btu/hr ft²F/in.

R = Sum of all thermal resistances aside from insulation

b = Cost per year per inch of insulation per sq ft

Let the cost of insulation per square foot per year equal N, and $N = bx + C$. Let y = total cost per year $= M + N$

$$M + N = \frac{a}{\frac{x}{k} + R} + bx + C = \frac{ak}{x + Rk} + bx + C$$

Differentiating and equating the first derivative to zero:

$$\frac{dy}{dx} = \frac{-ak}{(x + Rk)^2} + b = 0$$

$$-ak + b(x + Rk)^2 = 0 \quad \text{or} \quad ak = b(x + Rk)^2$$

Dividing through by b and extracting the square root of both terms,

$$\left(\frac{ak}{b}\right)^{1/2} = x + Rk$$

Therefore for most economical thickness, $x = (ak/b)^{1/2} - Rk$.

16. PIPE AND PIPING

THIS chapter deals with piping as a means of conveying fluid. Piping is also used as heat transfer surface, which use is treated in Chapter 5. Dimensions and construction details are the same for both uses, but other considerations differ.

The **design** of piping as a means of conveying fluid involves consideration of the following factors: Size, materials, strength, method of assembly, expansion and contraction, supports, protection against damage, control of flow, convenience, and restrictions imposed by codes, laws and regulations.

1. The **size** of pipe is determined principally by the permissible **frictional pressure drop.** How much drop is permissible depends on the cost of overcoming it. In refrigerant **suction lines,** whatever pressure drop occurs increases the power input per unit of refrigeration, and decreases capacity of apparatus, and it is generally eco-

Table 1. Dimensions of Welded Seamless Steel Pipe (ASA B36.1019–39)

Nominal pipe size, in.	Outside diameter, in.	Nominal Wall Thickness for Schedule Numbers, in.									
		10	20	30	40	60	80	100	120	140	160
⅛	0.405				0.068		0.095				
¼	0.540				0.088		0.119				
⅜	0.675				0.091		0.126				
½	0.840				0.109		0.147				0.187
¾	1.050				0.113		0.154				0.218
1	1.315				0.133		0.179				0.250
1¼	1.660				0.140		0.191				0.250
1½	1.900				0.145		0.200				0.281
2	2.375				0.154		0.218				0.343
2½	2.875				0.203		0.276				0.375
3	3.5				0.216		0.300				0.437
3½	4.0				0.226		0.318				
4	4.5				0.237		0.337		0.437		0.531
5	5.563				0.258		0.375		0.500		0.625
6	6.625				0.280		0.432		0.562		0.718
8	8.625		0.250	0.277	0.322	0.406	0.500	0.593	0.718	0.812	0.906
10	10.75		0.250	0.307	0.365	0.500	0.593	0.718	0.843	1.000	1.125
12	12.75		0.250	0.330	0.406	0.562	0.687	0.843	1.000	1.125	1.312
14 OD	14.0	0.250	0.312	0.375	0.437	0.593	0.750	0.937	1.062	1.250	1.406
16 OD	16.0	0.250	0.312	0.375	0.500	0.656	0.843	1.031	1.218	1.437	1.562
18 OD	18.0	0.250	0.312	0.437	0.562	0.718	0.937	1.156	1.343	1.562	1.750
20 OD	20.0	0.250	0.375	0.500	0.593	0.812	1.031	1.250	1.500	1.750	1.937
24 OD	24.0	0.250	0.375	0.562	0.687	0.937	1.218	1.500	1.750	2.062	2.312
30 OD	30.0	0.312	0.500	0.625							

The decimal thicknesses listed for the respective pipe sizes represent their nominal or average wall dimensions. For tolerances on wall thicknesses see appropriate material specifications.

Thicknesses shown in bold face type in Schedules-30 and 40 are identical with thicknesses for "standard weight" pipe in former lists; those in Schedules-60 and 80 are identical with thicknesses for "extra strong" pipe in former lists.

Table 2. Dimensions of Brass Pipe
(ASTM B43)

Nominal pipe size, in.	Outside diameter, in.	Wall thickness, in.	
		Regular	Extra strong
$\frac{1}{8}$	0.405	.062	.100
$\frac{1}{4}$	0.540	.082	.123
$\frac{3}{8}$	0.675	.090	.127
$\frac{1}{2}$	0.840	.107	.149
$\frac{3}{4}$	1.050	.114	.157
1	1.315	.126	.182
$1\frac{1}{4}$	1.660	.146	.194
$1\frac{1}{2}$	1.900	.150	.203
2	2.375	.156	.221
$2\frac{1}{2}$	2.875	.187	.280
3	3.500	.219	.304
$3\frac{1}{2}$	4.000	.250	.321
4	4.500	.250	.341
5	5.563	.250	.375
6	6.625	.250	.437
8	8.625	.312	.500
10	10.750	.365	.500
12	12.750	.375	—

Table 3. Dimensions of Copper Tubing
(ASA H23.1-1941 or ASTM B88)

Nominal size, in.	Outside diameter, in.	Wall thickness, in.		
		Type K	Type L	Type M
$\frac{1}{8}$	.250	.032	.025	.025
$\frac{1}{4}$	.375	.032	.030	.025
$\frac{3}{8}$	.500	.049	.035	.025
$\frac{1}{2}$	.625	.049	.040	.028
$\frac{5}{8}$	.750	.049	.042	.030
$\frac{3}{4}$	.875	.065	.045	.032
1	1.125	.065	.050	.035
$1\frac{1}{4}$	1.375	.065	.055	.042
$1\frac{1}{2}$	1.625	.072	.060	.049
2	2.125	.083	.070	.058
$2\frac{1}{2}$	2.625	.095	.080	.065
3	3.125	.109	.090	.072
$3\frac{1}{2}$	3.625	.120	.100	.083
4	4.125	.134	.110	.095
5	5.125	.160	.125	.109
6	6.125	.192	.140	.122
8	8.125	.271	.200	.170
10	10.125	.338	.250	.212
12	12.125	.405	.280	.254

nomical to hold the pressure drop to a minimum. In refrigerant **liquid lines** a substantial pressure drop is permissible and involves no penalty, so long as the line will deliver the required amount of liquid to the evaporator. In **water** and **brine lines,** the cost of extra pumping head to overcome the **pressure drop** must be balanced against the cost of providing larger lines. In all cases where an economic balance is involved, the load factor must be taken into account.

Pressure drop depends on these factors: quantity, density and viscosity of the fluid, diameter, roughness and length of the pipe. Determination of pressure drop as a function of these factors is treated at length in Chapter 6. Length must include the equivalent length of valves, fittings, etc., as given in **Fig. 1.** Pipe and tubing come in certain "nominal" sizes which are given in **Tables 1, 2** and **3.**

Tables 4a, 4b and **4c** give quantities (tons of refrigeration for refrigerants; gpm for water and brine) which will give a pressure drop of 1 psi in flowing through a 2-in. Schedule-40 steel pipe 100 ft long (including equivalent lengths). **Table 5** gives the multiplier, M_1, to apply to the flow in the 2-in. Schedule-40 steel pipe to obtain the same pressure drop in other nominal sizes of pipe and tubing. This pressure drop is rather high for refrigerant suction gases except for very short runs, and somewhat low for liquids except for very long runs. **Table 6** gives the multiplier, M_2, to apply to flow for obtaining other pressure drops.

For example, suppose it is desired to determine the quantity of flow of saturated ammonia vapor at 20 psig which will give a pressure drop of 1 psi in a 4-in. Schedule-40 steel pipe 200 ft long, with four elbows and one globe valve. The computation is as follows:

Total equivalent length in linear ft

Pipe	200
4 elbows at 11 ft (Fig. 1)	44
1 globe valve at 120 (Fig. 1)	120
Total	364

Permissible pressure drop per 100 lin ft $= 1/3.64 = .275$.

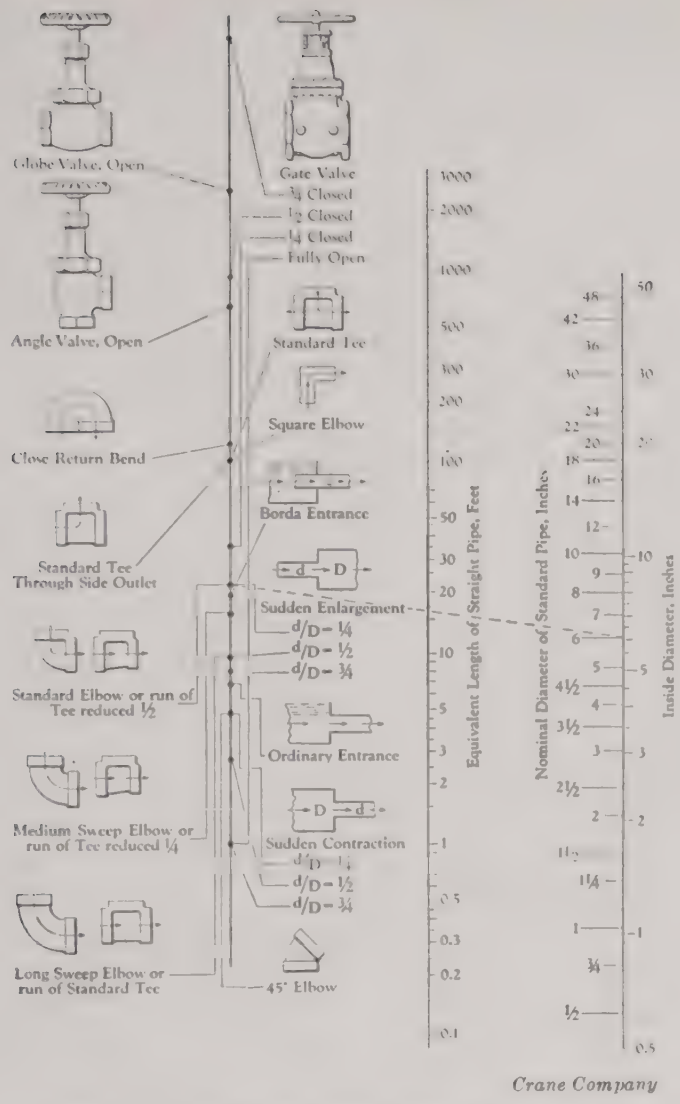


Fig. 1. Pressure Drop, Water Flow in Various Fittings

Example: 6-in. elbow is equivalent to 16½ ft of straight pipe

Multiplier for pressure drop (Table 6) $M_2 = .49$.

Multiplier for 4-in. Schedule-40 pipe (Table 5) $M_1 = 5.9$.

Tonnage in 2-in. Schedule-40 pipe for 1 psi, pressure drop per 100 lin ft (Table 4a) = 32.4 tons.

Tonnage in 4-in. Schedule-40 pipe for conditions given, $32.4 \times 0.49 \times 5.9 = 93.7$ tons.

A similar computation can be used to determine pipe size by solving for M_1 and referring to Table 5, although this involves trial and error in determining equivalent lengths, since these are different for the same type of fitting in different sizes.

The above formulation must be used with judgment and discretion, and if exact pressure drops are important, the method in Chapter 6 should be used, after determining approximate size by the method given. However, for ordinary sizing of pipe, there is such a great difference in pressure drop for a given flow between adjacent sizes that by using the above method, and picking the size which most nearly *exceeds* that indicated, the proper result will be obtained. In cases of doubt, oversizing is preferable to undersizing.

2. Materials commonly used in re-

Table 4a. Saturated Vapors. Flow Giving 1-psi Pressure Drop per 100 lin ft in 2-in. Schedule-40 ("Standard") Steel Pipe

Absolute pressure	Ammonia			Freon (F-12)			Freon (F-22)			Methyl Chloride			Sulfur Dioxide			Carbon Dioxide		
	Temp sat vapor	Lb per hr	Tons refrig	Temp sat vapor	Lb per hr	Tons refrig	Temp sat vapor	Lb per hr	Tons refrig	Temp sat vapor	Lb per hr	Tons refrig	Temp sat vapor	Lb per hr	Tons refrig	Temp sat vapor	Lb per hr	Tons refrig
in. Hg	F			F			F			F			F			F		
5	−83.0	219	8.0	−84.7	626	2.1	−99.3	538	2.6									
10	−63.7	310	11.6	−63.1	886	3.2	−79.2	757	3.8				−26.2	593	6.9			
15	−51.3	380	14.4	−48.8	1,076	4.0	−66.1	928	4.7				−12.5	727	8.5			
20	−41.9	439	16.7	−38.0	1,238	4.8	−56.4	1,069	5.6				− 2.0	840	9.9			
25	−34.3	490	18.8	−29.0	1,383	5.4	−48.4	1,192	6.3	−18.5	838	10.3	6.6	945	11.1			
psi																		
15	−27.3	540	20.8	−20.8	1,525	6.1	−40.8	1,314	7.1	−10.0	927	11.4	14.4	1,040	12.3			
20	−16.6	623	24.2	− 8.2	1,753	7.2	−29.4	1,507	8.2	2.9	1,073	13.4	26.4	1,202	14.3			
25	− 8.0	696	27.2	2.2	1,953	8.3	−20.0	1,685	9.4	13.3	1,204	15.2	36.3	1,345	16.0			
30	− 0.6	762	30.0	11.1	2,135	9.2	−11.7	1,854	10.5	22.3	1,325	16.9	44.8	1,472	17.6			
35	5.9	822	32.5	18.9	2,305	10.1	− 4.9	1,994	11.4	29.7	1,430	18.3	52.2	1,595	19.1			
40	11.7	877	34.8	25.9	2,465	11.0	1.5	2,130	12.3	36.6	1,535	19.8	58.8	1,707	20.4			
50	21.7	980	39.2	38.3	2,760	12.6	12.4	2,400	14.1	48.4	1,720	22.4	70.4	1,915	22.8			
60	30.2	1,072	43.1	48.7	3,020	14.1	21.8	2,610	15.5	58.6	1,892	24.8	80.3	2,095	24.9			
80	44.4	1,239	50.0	66.3	3,495	16.8	37.4	3,030	18.4	76.1	2,220	29.4	96.9	2,445		−67.4	2,320	10.2
100	56.0	1,386	56.2	80.9	3,925	19.3	50.4	3,400	21.0	88.7	2,440		110.2	2,750		−57.6	2,610	11.6
120	66.0	1,520	62.0	93.4	4,320		61.5	3,750	23.4	101	2,655		121.5	3,040		−49.3	2,865	12.8
140	74.8	1,643	67.2	104.5	4,710		71.3	4,070	25.6	111	2,805		131.6	3,310		−42.0	3,100	14.0
160	82.6	1,760	72.2	114.5	5,070		80.1	4,380	27.8	120	2,910					−35.4	3,320	15.1
180	89.8	1,868		123.7	5,420		88.2	4,680		128	3,140					−29.4	3,530	16.1
200	96.3	1,970		132.1	5,770		95.6	4,940		140	3,325					−23.8	3,730	17.1
250	110.8	2,195					112.0	5,650								−11.6	4,190	19.3
300	123.2	2,390														− 1.1	4,630	21.5
400																16.6	5,470	25.2
500																31.3	6,240	28.2
600																43.9	7,030	30.9
700																55.0	7,830	32.8
800																65.0	8,670	33.1
900																74.2	9,670	33.1
1,000																82.6	11,100	30.0

"Lb per hr" is in 2-in. Schedule-40 steel pipe with 1-psi pressure drop per 100 linear feet.
"Tons refrig" is based on 86 F liquid up to expansion valve.
Tables 4a, 4b, and 4c, except data on Freon-11, were prepared by Mr. M. B. Golber, and are based on viscosity and density data and formulation appearing in 1942 *Refriger-*

Table 4b. Superheated Vapors. Flow Giving 1-psi Pressure Drop per 100 lin ft in 2-in. Schedule-40 ("Standard") Steel Pipe

Absolute pressure	Ammonia			Freon (F-12)			Freon (F-22)			Methyl Chloride			Sulfur Dioxide			Carbon Dioxide		
	Temp comp vapor	Lb per hr	Tons refrig	Temp comp vapor	Lb per hr	Tons refrig	Temp comp vapor	Lb per hr	Tons refrig	Temp comp vapor	Lb per hr	Tons refrig	Temp comp vapor	Lb per hr	Tons refrig	Temp comp vapor	Lb per hr	Tons refrig
in. Hg	F			F			F			F			F			F		
25													8	938	13.1			
psi																		
15													28	1,021	14.1			
20													57	1,144	15.4			
25										25	1,203	17.8	81	1,254	16.6			
30				13	2,140	12.1				45	1,283	18.6	98	1,325	17.2			
35	7	822	38.6	22	2,310	12.7				65	1,367	19.5	115	1,413	18.0			
40	22	869	40.4	31	2,465	13.3				75	1,448	20.4	132	1,522	19.1			
50	47	965	44.0	46	2,730	14.2	16.4	2,370	18.1	100	1,612	22.1	158	1,667	20.4			
60	69	1,027	46.0	58	2,990	14.9	31.7	2,580	19.1	125	1,760	23.6	179	1,835	21.9			
80	105	1,153	50.1	78	3,450	16.0	56.7	2,940	20.8	155	1,990	25.6	215	2,030	23.3			
100	135	1,260	53.3	94	3,850	16.8	77.0	3,260	21.9	180	2,215	27.5	243	2,210	24.5			
120	160	1,354	56.1	109	4,220	17.4	94.0	3,550	22.8	200	2,430	29.2	265	2,405	26.0			
140	182	1,440	58.5	121	4,560	17.7	109.0	3,820	23.6	220	2,550	29.8	284	2,550	26.8			
160	201	1,520	60.6	133	4,890	17.9	122.3	4,070	24.2	235	2,680	30.5						
180	219	1,595	62.5	143	5,220	18.1	134.7	4,300	24.6	250	2,805	30.9						
200	235	1,660	64.0	152	5,520	18.2	145.7	4,530	25.0	270	2,940	29.8						
250	270	1,820	67.5				168.8	5,030	25.3									
300	300	1,955	70.1															
400																28	5,330	49.5
500																54	5,860	50.1
600																82	6,430	50.7
700																98	6,750	48.8
800																118	7,100	47.0
900																133	7,500	44.5
1,000																148	7,750	39.7

Temperature of compressed vapor based upon isentropic compression from saturated vapor at 5 F.
Tons of refrigeration based upon liquid at temperature corresponding to pressure to saturated vapor at 5 F.

Table 4c. Liquids. Flow Giving 1-psi Pressure Drop per 100 lin ft in 2-in. Schedule-40 ("Standard") Steel Pipe

Fluid	Temp, F	Density, lb/cu ft	lb/hr	Flow, gpm	Tons†
Water	32	62.4	15,900	31.8	
	100	62.0	17,100	34.5	
	212	59.8	17,800	37.2	
Water, average pipe*				25.0	
Sodium brine —10% NaCl	25	67.2	16,200	31.2	
—15% NaCl	15	69.9	16,100	28.8	
	25	69.8	16,300	29.2	
—20% NaCl	5	72.7	15,500	26.6	
	15	72.5	15,900	27.4	
	25	72.2	16,350	28.3	
Calcium brine—10% $CaCl_2$	30	68.2	16,350	30.0	
—20% $CaCl_2$	0	74.7	15,200	25.4	
	15	74.5	15,800	26.5	
	30	74.3	16,500	27.7	
—30% $CaCl_2$	−15	81.9	13,500	20.6	
	0	81.7	14,600	22.3	
	15	81.5	15,400	23.6	
	30	81.3	16,100	24.7	
Ammonia	86	37.2	14,100		556.
Carbon dioxide	86	37.4	17,900		82.8
Freon (F-12)	86	80.6	21,000		89.3
Freon-22 (F-22)	86	73.4	2,000		103.9
Methyl chloride	86	56.3	17,400		218.
Sulfur dioxide	86	84.4	21,300		251.

* Water pipe is subject to scale, corrosion and tuberculation. This value gives a close approximation to the Williams and Hazen tables, using $C = 100$.

† Tons refrigeration based on 5 F evaporation, saturated gas, and 86 F liquid.

frigerant, brine and water piping are steel, wrought iron, malleable iron, cast iron, copper, brass, and bronze, as furnished under the specifications listed in **Table 7,** which is taken from the ASA B31 Piping Code. Copper and brass must not be used with ammonia. Approval of use of any of the materials is predicated on refrigerants which are free from moisture and air, and on brine or water in which the *p*H value is properly controlled or in which proper **corrosion inhibitors** are used.

Other materials are not barred by the code, providing they are of suitable strength and chemically are suitable for use with the fluid conveyed. The code bars the use of aluminum base alloys with F-12 or F-114, or magnesium base alloys with any Freon refrigerant; also the use of zinc, aluminum, die-castings and magnesium alloys with methyl chloride.

For temperatures below 0 F, the code provides that materials shall have a Charpy **impact value** at the operating temperature of not less than 15 ft-lb using a keyhole notch, but permits copper, brass and bronze without restrictions, and permits ferrous materials not meeting this requirement down to −100 F, provided the design pressure is increased 2% for each degree below 0 F. This permits the ordinary lines of cast iron refrigerant fittings down to −50 F without penalty. As to what materials are suitable for low temperatures, the code has the following to say:

"There is a large and constantly growing list of alloy compositions which meet

Table 5. Values of M_1 Multipliers to be Applied to Flow in 2-in. Schedule-40 Steel Pipe to Find Flow Giving Same Pressure Drop

Nominal size, in.	Steel or wrought iron		Brass		Copper tubing		
	Schedule-40, Standard	Schedule-80, Extra Strong	Regular	Extra Strong	Type K	Type L	Type M
$\frac{1}{8}$	.0041	.0022	.0052	.0022	.00175	.0021	.0021
$\frac{1}{4}$	.0090	.0055	.0115	.0059	.0068	.0071	.0077
$\frac{3}{8}$	.021	.0138	.025	.0157	.0138	.0166	.0188
$\frac{1}{2}$	.039	.028	.047	.032	.029	.032	.036
$\frac{5}{8}$	—	—	—	—	.052	.055	.060
$\frac{3}{4}$	.086	.064	.098	.073	.075	.086	.094
1	.164	.127	.193	.144	.162	.176	.190
$1\frac{1}{4}$	.33	.27	.38	.31	.29	.30	.32
$1\frac{1}{2}$	.51	.42	.57	.48	.47	.49	.50
2	1.00	.84	1.13	.95	.99	1.02	1.04
$2\frac{1}{2}$	1.65	1.37	1.90	1.55	1.77	1.83	1.89
3	2.8	2.4	3.3	2.8	2.8	2.9	3.0
$3\frac{1}{2}$	4.2	3.6	4.6	4.2	4.2	4.4	4.5
4	5.9	5.1	6.6	5.8	6.0	6.2	6.3
5	10.7	9.5	12.1	10.6	10.6	11.0	11.2
6	17.5	15.3	19.9	16.9	16.8	17.6	17.9
8	36	32	40	35	35	36	37
10	67	58	71	66	61	64	66
12	108	93	114	—	97	103	104

these requirements, and it has not been deemed advisable to infer any limitation by including some few of them in Table 7. Depending on composition and heat treatment, some 'carbon steels' meet the requirements for a considerable range of temperature. In general, the 3 to 4 per cent nickel steels are acceptable for quite low temperatures, and the 18 per cent chrome-8 per cent nickel stainless steels meet requirements to very low temperatures. Various other classes of alloys are also acceptable. Apparently minor variations in composition and heat treatment make large differences in toughness at low temperatures."

A summary of available data is contained in "Properties of Metals at Subatmospheric Temperatures" issued in 1941 by the American Society of Testing Materials.

3. The **strength** of pipe, that is, the thickness required in a given material for a given pressure, is fixed by the Piping Code as follows:

$$t_m = \frac{PD}{2S + 0.8P} + C$$

where

t_m = minimum pipe wall thickness in inches,

P = maximum internal service pressure in pounds per square inch gage (plus allowance for brittleness at low temperatures and water hammer allowance for cast iron pipe),

D = outside diameter of pipe in inches,

Table 6. Values of M_2 Multipliers for Flow to Give Other Pressure Drops

Pressure drop, psi	.0	.2	.4	.6	.8
0.	.00	.42	.61	.77	.89
1.	1.00	1.10	1.20	1.29	1.37
2.	1.45	1.52	1.60	1.67	1.73
3.	1.80	1.86	1.92	1.98	2.04
4.	2.10	2.15	2.21	2.26	2.31
5.	2.36	2.41	2.46	2.51	2.56

Table 7. List of Material Specifications

Material	Specification
Bolts	
Alloy steel	ASTM A96
Commercial steel (bar stock)	ASTM A107
Wrought iron (bar stock)	ASTM A84
Fittings, valves and flanges	
Cast iron flanges, flanged fittings and valves	ASTM A126
Cast iron bell and spigot fittings, and valves	AWWA
Malleable iron	ASTM A197
Steel (cast carbon)	ASTM A27 or A95
Steel (forged)	ASTM A105 or A181
Brass castings	ASTM B62
Bronze castings	ASTM B61
Pipe and tubing	
Steel, seamless or welded, for coiling or bending	ASTM A53
Steel, seamless or welded, for ordinary uses	ASTM A120
Steel, seamless or welded	API 5L
Steel, seamless or welded	ASTM A106
Steel, seamless	ASTM A83 or A179 or A192
Steel, seamless, medium carbon	ASTM A210
Steel, seamless, alloy	ASTM A213
Steel, electric resistance welded	ASTM A135 or A178 or A214 or A226
Steel, electric resistance welded, alloy	ASTM A249
Wrought iron, welded	ASTM A72
Wrought iron, welded	API 5L
Cast iron (pit cast)	ASA A21.2 (ASTM A44)
Cast iron (centrifugally cast or cast horizontally in green sand molds)	FSEC WWP-P-421
Brass (seamless)	ASTM B43
Copper pipe (seamless)	ASTM B42
Copper tubing	ASTM B88

S = allowable stress in material due to internal pressure in pounds per square inch, and

C = allowance for threading, mechanical strength and for corrosion, in inches, as follows:

Cast iron pipe centrifugally cast or cast horizontally in green sand molds, 0.14

Cast iron pipe, pit-cast, 0.18

Threaded steel, wrought iron or non-ferrous pipe, $\frac{3}{8}$-in. and smaller, 0.05; for $\frac{1}{2}$-in. and larger, use depth of thread.

Grooved steel, wrought iron or non-ferrous pipe, use depth of groove.

Plain end steel or wrought iron pipe, 1-in. size and smaller, 0.05; for sizes larger than 1 in., 0.065.

Plain end non-ferrous pipe or tube, zero.

Values of S to be used for the materials listed in Table 7 are given in **Table 8.** For other materials, the Piping Code gives rules for figuring S values. Values of service pressures to be used in the formula are two-thirds of the test pressures given in **Table 9,** but in no case are values of P less than 100 to be used.

The thicknesses of ferrous fittings and valves are covered by the various ASA B16 standards for various services, except that refrigerant fittings and valves have not been standardized, and "manufacturers standards of long standing" are recognized by the Code. However, ASA B16, 16 covering refrigerant fitting and flanges, was promulgated in 1948 and will be followed by all manufacturers after a transition period. There are ASA A40 standards on non-ferrous soldered and flared compression fittings.

There are also certain specific requirements set forth in the Piping Code and in the ASA B9 Safety Code for Mechanical Refrigeration. Chief of these rules are:

Refrigerant working pressure shall not exceed 250 psi in Schedule-40 ferrous pipe and butt welded pipe shall not exceed 2 in. in size.

Refrigerant liquid ferrous pipe, $1\frac{1}{2}$-in. and smaller, shall be of Schedule-80.

Refrigerant tubing shall not be lighter than Type L for field assembly, or Type M for shop assembly.

Soft annealed copper tubing erected on the

Table 8. Allowable *S* Values to Be Used in Formula to Determine Pipe Thickness

Material	Specification	Value of *S*, psi
Steel pipe, seamless, Grade A	ASTM A53 or ASTM A120 or ASTM A106 or ASTM A83 or ASTM A179 or ASTM A192 or API 5L	12,000
Steel pipe, seamless, Grade B	ASTM A53 or ASTM A106 or ASTM A210	15,000
Steel, seamless alloy, grades T18, T19	ASTM A213	18,750
Steel, electric resistance welded, alloy grades T18, T19	ASTM A249	15,900
Steel pipe, lap welded	ASTM A53 or ASTM A106 or ASTM A120 or API 5L	9,000
Steel pipe, butt welded	ASTM A53 or ASTM A120	6,800
Steel pipe, electric resistance welded	ASTM A135 or ASTM A178 or ASTM A214 or ASTM A226	10,200
Wrought iron, lap welded	ASTM A72 or API 5L	8,000
Wrought iron, butt welded	ASTM A72	6,000
Cast iron pipe, pit cast	AWWA	4,000
Cast iron pipe, centrifugally cast or cast horizontally in green sand molds	FSEC WW-P-421	6,000
Brass pipe, seamless red brass	ASTM B43	4,700
Copper pipe, seamless	ASTM B42	4,000
Copper tubing, seamless	ASTM B88	4,000

premises shall not be used in sizes exceeding $\frac{3}{4}$ in.

Pipe thinner than Schedule-40 ferrous, or regular brass, shall not be threaded.

For refrigerants, cast iron shall conform to ASTM A126, Class B higher strength grey iron with not less than 31,000 psi tensile strength.

4. The **method of assembly** of piping refers to the type of joints, which may be screwed, flanged, welded, brazed, soldered, or in small tubing by flared compression fittings.

a. Screwed joints are limited by the Piping Code to not over 3 in. nominal size for refrigerant pressures 250 psi or less, $1\frac{1}{4}$ in. for refrigerant pressures above 250 psi and not over 4 in. for brine. Screwed-on flanges are not permitted in sizes larger than the above. Bushings are not permitted for reductions of a single pipe size. For screwed brass or copper refrigerant pipe, extra heavy fittings are required. **Table 10** gives dimensions of 125 and 250 psi screwed cast iron fittings up to 4 in. in size.

b. Flanged joints are used for sizes greater than the above, and in many cases for smaller sizes. There are various ASA B16 standards for flanged fittings for various services. The most commonly used are the 125 and 250 psi cast iron fittings,

Table 9. Test Pressures

Refrigerant	Chemical formula	Minimum test pressure, psi	
		High side	Low side
Ammonia	NH_3	300	150
Butane	C_4H_{10}	90	50
Carbon dioxide	CO_2	1,500	1,000
Dichlorodifluoromethane (Freon-12)	CCl_2F_2	235	145
Dichlorotetrafluoroethane (Freon-114)	$C_2Cl_2F_4$	80	50
Dichloromethane (Carrene No. 1) (Methylene chloride)	CH_2Cl_2	30	30
Dichloromonofluoromethane (Freon-21)	$CHCl_2F$	70	50
Dichloroethylene	$C_2H_2Cl_2$	30	30
Ethane	C_2H_6	1,100	600
Ethyl chloride	C_2H_5Cl	60	50
Isobutane	$(CH_3)_3CH$	130	75
Methyl chloride	CH_3Cl	215	125
Methyl formate	$HCOOCH_3$	50	50
Propane	C_3H_8	325	210
Sulfur dioxide	SO_2	170	95
Trichloromonofluoromethane (Freon-11)	CCl_3F	50	30

Note: For refrigerants not listed in table, high-side test pressure is saturation pressure at 150 F, low-side test pressure is saturation pressure at 115 F, in no case less than 30 psi.

dimensions of which are given in **Fig. 2** and **Table 11.** The 150 and 300 psi steel standards conform to the same dimensions respectively.

Refrigerant flanged fittings and flanges were standardized in 1948. Standard dimensions are given in Tables 12 and 12A. These may be expected, during the next few years, to supersede manufacturers' standards which differ somewhat among

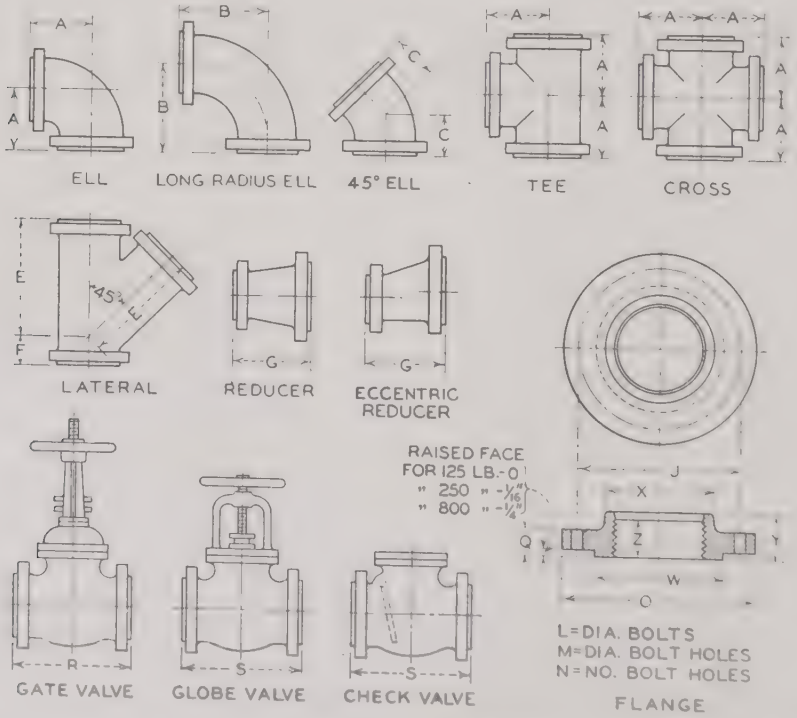
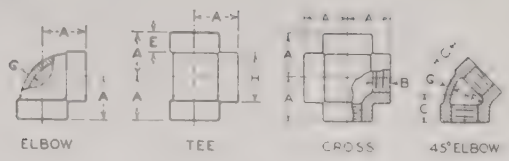


Fig. 2. Dimensions of Flanged Cast Iron Fittings, Flanges and Valves (see Table 11)

Table 10. Dimensions of Screwed Cast Iron Fittings, Straight Sizes

ASA B16d-1941



	Nominal pipe size, in.	A Center to end, elbows, tees and crosses	C Center to end, 45-deg elbows	B Length of thread, min in.	E Width of band, min in.	F Inside diameter of fitting, in. Min	F Inside diameter of fitting, in. Max	G Metal thickness,* in.	H Outside diameter of band, min in.
125-psi	¼	0.81	0.73	0.32	0.38	0.540	0.584	0.110	0.93
	⅜	0.95	0.80	0.36	0.44	0.675	0.719	0.120	1.12
	½	1.12	0.88	0.43	0.50	0.840	0.897	0.130	1.34
	¾	1.31	0.98	0.50	0.56	1.050	1.107	0.155	1.63
	1	1.50	1.12	0.58	0.62	1.315	1.385	0.170	1.95
	1¼	1.75	1.29	0.67	0.69	1.660	1.730	0.185	2.39
	1½	1.94	1.43	0.70	0.75	1.900	1.970	0.200	2.68
	2	2.25	1.68	0.75	0.84	2.375	2.445	0.220	3.28
	2½	2.70	1.95	0.92	0.94	2.875	2.975	0.240	3.86
	3	3.08	2.17	0.98	1.00	3.500	3.600	0.260	4.62
	3½	3.42	2.39	1.03	1.06	4.000	4.100	0.280	5.20
	4	3.79	2.61	1.08	1.12	4.500	4.600	0.310	5.79
250-psi	¼	0.94	0.81	0.43	0.49	0.540	0.584	0.18	1.17
	⅜	1.06	0.88	0.47	0.55	0.675	0.719	0.18	1.36
	½	1.25	1.00	0.57	0.60	0.840	0.897	0.20	1.59
	¾	1.44	1.13	0.64	0.68	1.050	1.107	0.23	1.88
	1	1.63	1.31	0.75	0.76	1.315	1.385	0.28	2.24
	1¼	1.94	1.50	0.84	0.88	1.660	1.730	0.33	2.73
	1½	2.13	1.69	0.87	0.97	1.900	1.970	0.35	3.07
	2	2.50	2.00	1.00	1.12	2.375	2.445	0.39	3.74
	2½	2.94	2.25	1.17	1.30	2.875	2.975	0.43	4.60
	3	3.38	2.50	1.23	1.40	3.500	3.600	0.48	5.36
	3½	3.75	2.63	1.28	1.49	4.000	4.100	0.52	5.98
	4	4.13	2.81	1.33	1.57	4.500	4.600	0.56	6.61

Reducing sizes for 125 psi vary slightly and are given in the B 16d Standard.
Reducing sizes for 250 psi are not covered by the Standard.
* Minimum at any point, 90% of value given.

themselves but conform more or less closely to the standard. The Piping Code provides that facing of refrigerant flanges shall be of the retained gasket type (male and female, tongue-and-groove, or ring) and all the manufacturers' standards are of the tongue-and-groove type. It is necessary to match flanges to be sure they come from the same manufacturer and to be sure that one of the pair is tongued and one grooved, and this frequently leads to annoying delays and extra joints in lines unless it is watched carefully. It also requires the carrying of larger inventories in plants or shops using equipment from more than one manufacturer.

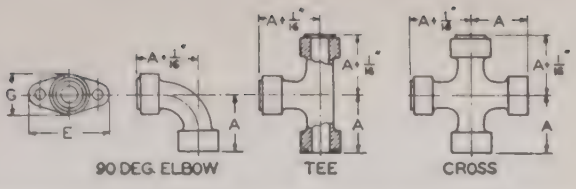
c. Welding is practically universally accepted as the standard method of assembly of ferrous piping systems today, except valves are flanged and flanged unions are used where disassembly is frequent. Rules for qualification of welding procedures and operators are given at length in the Piping Code. This is a complex subject in itself and no brief description here would be other than misleading.

Table 11. Dimensions of Flanged Cast Iron Fittings and Valves (see Fig. 2)

	Nominal pipe size, in.	1	1¼	1½	2	2½	3	3½	4	5	6	8	10	12
125-psi Standard ASA B16a-1939	A	3½	3¾	4	4½	5	5½	6	6½	7½	8	9	11	12
	B	5	5½	6	6½	7	7¾	8½	9	10¼	11½	14	16½	19
	C	1¾	2	2½	2¼	3	3	3½	4*	4½	5	5½	6½	7½
	E	5¾	6¼	7	8	9½	10	11½	12	13½	14½	17½	20½	24½
	F	1¾	1¾	2	2½	2½	3	3	3	3½	3½	4½	5	5½
	G	—	—	—	5	5½	6	6½	7	8	9	11	12	14
	J	3 1/8	3½	3 7/8	4¾	5½	6	7	7½	8½	9½	11¾	14¼	17
	L	½	½	½	⅝	⅝	⅝	⅝	⅝	¾	¾	¾	⅞	⅞
	M	⅝	⅝	⅝	¾	¾	¾	¾	¾	⅞	⅞	⅞	1	1
	N	4	4	4	4	4	4	8	8	8	8	8	12	12
	O	4¼	4⅝	5	6	7	7½	8½	9	10	11	13½	16	19
	Q	7/16	½	9/16	⅝	11/16	¾	13/16	15/16	15/16	1	1⅛	1 3/16	1¼
	R*	—	—	—	7	7½	8	8½	9	10	10½	11½	13	14
	S*	—	—	—	8	8½	9½	10½	11½	13	14	19½	—	—
	X	1 15/16	2 5/16	2 9/16	3 1/16	3 9/16	4¼	4 13/16	5 5/16	6 7/16	7 9/16	9 11/16	11 15/16	14 1/16
	Y	11/16	13/16	⅞	1	1⅛	1 3/16	1¼	1 5/16	1 7/16	1 9/16	1¾	1 15/16	2 3/16
	Z	Same as Y												
250-psi Standard ASA B16b-1944	A	—	—	—	5	5½	6	6½	7	8	8½	10	11½	13
	B	—	—	—	6½	7	7¾	8½	9	10¼	11½	14	16½	19
	C	—	—	—	3	3½	3½	4	4½	5	5½	6	7	8
	E	6½	7¼	8½	9	10½	11	12½	13½	15	17½	20½	24	27½
	F	2	2¼	2½	2½	2½	3	3	3	3½	4	5	5½	6
	G	—	—	—	5	5½	6	6½	7	8	9	11	12	14
	J	3½	3 7/8	4½	5	5 7/8	6 5/8	7¼	7 7/8	9¼	10 5/8	13	15¼	17¾
	L	⅝	⅝	¾	⅝	¾	¾	¾	¾	¾	¾	⅞	1	1⅛
	M	¾	¾	⅞	¾	⅞	⅞	⅞	⅞	⅞	⅞	1	1⅛	1¼
	N	4	4	4	8	8	8	8	8	8	12	12	16	16
	O	4⅞	5¼	6⅛	6½	7½	8¼	9	10	11	12½	15	17½	20½
	Q	11/16	¾	13/16	⅞	1	1⅛	1 3/16	1¼	1⅜	1 7/16	1⅝	1⅞	2
	R*	—	—	—	8½	9½	11⅛	11⅞	12	15	15⅞	16½	18	19¾
	S*	—	—	—	10½	11½	12½	13¼	14	15¾	17½	21	24½†	28†
	W	2 11/16	3 1/16	3 9/16	4 3/16	4 15/16	5 11/16	6 5/16	6 15/16	8 5/16	9 11/16	11 15/16	14 1/16	16 7/16
	X	2 1/16	2½	2¾	3 5/16	3 15/16	4⅝	5¼	5⅜	7	8⅛	10¼	12⅝	14¾
	Y	⅞	1	1⅛	1¼	1 7/16	1 9/16	1⅝	1¾	1⅞	1 15/16	2 3/16	2⅜	2 9/16
	Z	0.68	0.76	0.87	1.00	1.14	1.20	1.25	1.30	1.41	1.51	1.71	1.92	2.12
800-psi Standard ASA B16b1-1931	A	—	—	—	5¾	6½	7	7½	8½	10	11	13	15½	16½
	C	—	—	—	4¼	4½	5	5½	6	7	7½	8½	9½	10
	E	—	—	—	10¼	11½	12¾	14	16½	19½	21	24½	29½	31½
	F	—	—	—	3½	3½	4	4½	4½	6	6½	7	8	8½
	G	—	—	—	6	6¾	7¼	7¾	8¾	10¼	11¼	13¼	15¾	16¾
	J	—	—	—	5	5 7/8	6 5/8	7¼	8½	10½	11½	13¾	17	19¼
	L	—	—	—	⅝	¾	¾	⅞	⅞	1	1	1⅛	1¼	1⅜
	M	—	—	—	¾	⅞	⅞	1	1	1⅛	1⅛	1¼	1⅜	1⅝
	N	—	—	—	8	8	8	8	8	8	12	12	16	20
	O	—	—	—	6½	7½	8¼	9	10¾	13	14	16½	20	22
	Q	—	—	—	1½	1⅝	1¾	1⅞	2⅛	2⅜	2½	2¼	3⅛	3¼
	R*	—	—	—	11½	13	14	—	17	—	22	26	31	33
	S*	This dimension has not been standardized, except for check valves same as R												
	W	—	—	—	3⅝	4⅛	5	5½	6 3/16	7 1/16	8½	10⅜	12¾	15

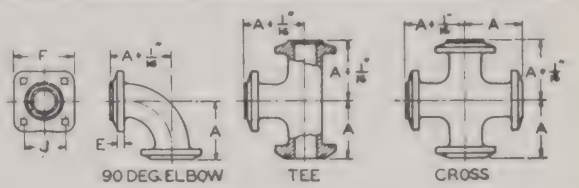
* ASA B16.10.
† Check valves only.

Table 12. ASA B16.16 Standard Cast Iron Refrigerant Fittings



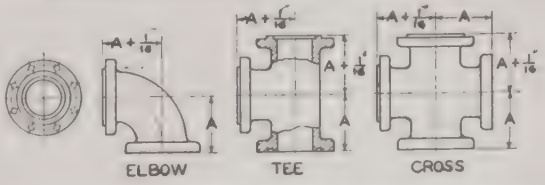
Oval Flanged Fittings

Nominal pipe size, in.	Inside diameter of fitting	Metal thickness of fitting, min	Long diameter of flange E	Short diameter of flange G	Thickness of flange, min B	Thickness† of flange at bolt circle D	Center* to female flange edge, elbow tee and cross A	Diameter of bolt circle	Number of bolts	Size of bolts
$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{4}$	$3\frac{7}{16}$	$1\frac{13}{16}$	$\frac{13}{16}$	$\frac{1}{2}$	$2\frac{3}{8}$	$2\frac{1}{4}$	2	$\frac{1}{2}$
$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{4}$	$3\frac{11}{16}$	$1\frac{13}{16}$	$\frac{15}{16}$	$\frac{9}{16}$	$2\frac{1}{2}$	$2\frac{1}{2}$	2	$\frac{1}{2}$
$\frac{1}{2}$	$\frac{5}{8}$	$\frac{1}{4}$	$3\frac{7}{8}$	$1\frac{13}{16}$	1	$\frac{5}{8}$	$2\frac{1}{4}$	$2\frac{3}{4}$	2	$\frac{1}{2}$
$\frac{3}{4}$	$\frac{13}{16}$	$\frac{1}{4}$	$4\frac{1}{8}$	$2\frac{1}{8}$	$1\frac{1}{8}$	$\frac{11}{16}$	$3\frac{1}{4}$	3	2	$\frac{1}{2}$



Square Flanged Fittings

Nominal pipe size, in.	Inside diameter of fitting	Metal thickness of fitting, min	Length and width of flange F	Thickness† of flange, min B	Thickness of flanges at edge E	Center* to flange edge, elbow tee and cross A	Bolt spacing J	Number of bolts	Size of bolts
1	$1\frac{1}{16}$	$\frac{1}{4}$	$3\frac{9}{16}$	$\frac{15}{16}$	$\frac{9}{16}$	$3\frac{3}{8}$	$2\frac{5}{16}$	4	$\frac{1}{2}$
$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{5}{16}$	$3\frac{7}{8}$	$1\frac{1}{16}$	$\frac{5}{8}$	$3\frac{7}{8}$	$2\frac{5}{8}$	4	$\frac{1}{2}$
$1\frac{1}{2}$	$1\frac{9}{16}$	$\frac{5}{16}$	$4\frac{1}{8}$	$1\frac{1}{8}$	$\frac{11}{16}$	$4\frac{1}{4}$	$2\frac{13}{16}$	4	$\frac{1}{2}$
2	$2\frac{1}{16}$	$\frac{3}{8}$	5	$1\frac{1}{4}$	$\frac{13}{16}$	$4\frac{3}{4}$	$3\frac{7}{16}$	4	$\frac{5}{8}$
$2\frac{1}{2}$	$2\frac{1}{2}$	$\frac{3}{8}$	$5\frac{1}{2}$	$1\frac{3}{8}$	$\frac{7}{8}$	$5\frac{3}{8}$	$3\frac{7}{8}$	4	$\frac{5}{8}$
3	3	$\frac{7}{16}$	$6\frac{1}{8}$	$1\frac{1}{2}$	$1\frac{1}{16}$	$5\frac{7}{8}$	$4\frac{5}{16}$	4	$\frac{3}{4}$
$3\frac{1}{2}$	$3\frac{1}{2}$	$\frac{7}{16}$	$6\frac{1}{2}$	$1\frac{5}{8}$	1	$6\frac{3}{8}$	$4\frac{3}{4}$	4	$\frac{3}{4}$
4	4	$\frac{7}{16}$	$7\frac{1}{4}$	$1\frac{3}{4}$	$1\frac{1}{8}$	$6\frac{7}{8}$	$5\frac{1}{8}$	4	$\frac{7}{8}$



Round Flanged Fittings

Nominal pipe size	Inside diameter of fitting	Wall thickness of fitting	Outside diameter of flange	Thickness† of flange,[1,2] min		Center* to groove flange edge elbow, tee and cross A	Diameter of bolt circle	Number of bolts	Size of bolts
				Tongue	Groove				
5	5	$\frac{11}{16}$	11	$1\frac{5}{8}$	$1\frac{9}{16}$	$8\frac{3}{16}$	$9\frac{1}{4}$	8	$\frac{3}{4}$
6	6	$\frac{3}{4}$	$12\frac{1}{2}$	$1\frac{11}{16}$	$1\frac{5}{8}$	$8\frac{11}{16}$	$10\frac{5}{8}$	12	$\frac{3}{4}$
8	8	$\frac{13}{16}$	15	$1\frac{7}{8}$	$1\frac{13}{16}$	$10\frac{3}{16}$	13	12	$\frac{7}{8}$
10	10	$\frac{15}{16}$	$17\frac{1}{2}$	$2\frac{1}{8}$	$2\frac{1}{16}$	$11\frac{11}{16}$	$15\frac{1}{4}$	16	1
12	12	1	$20\frac{1}{2}$	$2\frac{1}{4}$	$2\frac{3}{16}$	$13\frac{3}{16}$	$17\frac{3}{4}$	16	$1\frac{1}{8}$

* The $\frac{5}{32}$-in. tongue of oval and square flanges, and the $\frac{1}{4}$" tongue of round flanges, is included in the "thickness of flange minimum," and all but $\frac{1}{16}$ in. is included in the "center to flange edge" dimensions.

† See Table 12a.

Welded joints are generally either of the butt-welding or socket-welding type. Dimensions of welding fittings have recently been standardized, or standards are in proposed form, and dimensions are given in **Tables 13, 14** and **15.** Branch connections require suitable reinforcing. Butt-welding is commonly used for 2-in. nominal size and larger, socket-welding for $1\frac{1}{2}$-in. and smaller.

d. Soldering for non-ferrous materials corresponds in function, although not in method, with welding for ferrous materials. Types and dimensions of solder-joint fittings are given in **Table 16.** The clearance between the two surfaces is maintained within narrow limits. The solder, with suitable flux, is run into this narrow annulus in a molten condition. The joint depends on the large area of contact, and of shear. **Hard solder** is ordinarily a silver base alloy, melting above 1,000 F, whereas **soft solder** is ordinarily a tin base alloy, melting below 500 F. Hard soldering is also called **silver brazing,** under which name current practice was reviewed by Edelson.[1] Most shop soldered joints in refrigerant piping are hard soldered. The Piping Code requires the use of hard solder for field joints in refrigerant lines, except for tubes $\frac{1}{2}$-in. nominal size and less in systems containing less than 20 lb of refrigerant.

e. Flared compression fittings are permitted by the Piping Code for refrigerant lines not over $\frac{3}{4}$ in. outside diameter, using annealed copper tubing, providing all such joints are exposed for visual inspection.

[1] Edelson, *Refrig. Eng.*, Jan. 1941.

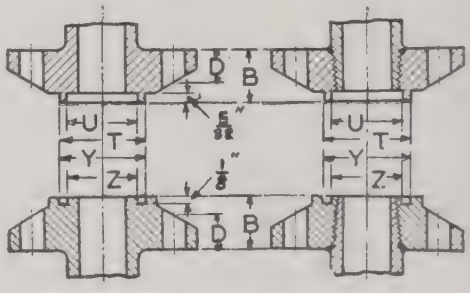


Table 12a. Standard Facing Dimensions

Nominal pipe size	Tongue			Groove			Flange thickness, min		Bolt
	Height	Outside diameter T	Inside diameter U	Depth	Outside diameter Y	Inside diameter Z	D	B	Spacing
Oval									
$\frac{1}{4}$	$\frac{5}{32}$	$1\frac{3}{8}$	1	$\frac{1}{8}$	$1\frac{7}{16}$	$\frac{15}{16}$	$\frac{1}{2}$	$\frac{13}{16}$	$2\frac{1}{4}$
$\frac{3}{8}$	$\frac{5}{32}$	$1\frac{3}{8}$	1	$\frac{1}{8}$	$1\frac{7}{16}$	$\frac{15}{16}$	$\frac{9}{16}$	$\frac{15}{16}$	$2\frac{1}{2}$
$\frac{1}{2}$	$\frac{5}{32}$	$1\frac{3}{8}$	1	$\frac{1}{8}$	$1\frac{7}{16}$	$\frac{15}{16}$	$\frac{5}{8}$	1	$2\frac{3}{4}$
$\frac{3}{4}$	$\frac{5}{32}$	$1\frac{11}{16}$	$1\frac{5}{16}$	$\frac{1}{8}$	$1\frac{3}{4}$	$1\frac{1}{4}$	$\frac{11}{16}$	$1\frac{1}{8}$	3
Square									
1	$\frac{5}{32}$	$1\frac{7}{8}$	$1\frac{1}{2}$	$\frac{1}{8}$	$1\frac{15}{16}$	$1\frac{7}{16}$	—	$\frac{15}{16}$	$2\frac{5}{16}$
$1\frac{1}{4}$	$\frac{5}{32}$	$2\frac{1}{4}$	$1\frac{7}{8}$	$\frac{1}{8}$	$2\frac{5}{16}$	$1\frac{13}{16}$	—	$1\frac{1}{16}$	$2\frac{5}{8}$
$1\frac{1}{2}$	$\frac{5}{32}$	$2\frac{1}{2}$	$2\frac{1}{8}$	$\frac{1}{8}$	$2\frac{9}{16}$	$2\frac{1}{16}$	—	$1\frac{1}{8}$	$2\frac{13}{16}$
2	$\frac{5}{32}$	$3\frac{1}{4}$	$2\frac{7}{8}$	$\frac{1}{8}$	$3\frac{5}{16}$	$2\frac{13}{16}$	—	$1\frac{1}{4}$	$3\frac{7}{16}$
$2\frac{1}{2}$	$\frac{5}{32}$	$3\frac{3}{4}$	$3\frac{3}{8}$	$\frac{1}{8}$	$3\frac{13}{16}$	$3\frac{5}{16}$	—	$1\frac{3}{8}$	$3\frac{7}{8}$
3	$\frac{5}{32}$	$4\frac{5}{8}$	$4\frac{1}{4}$	$\frac{1}{8}$	$4\frac{11}{16}$	$4\frac{3}{16}$	—	$1\frac{1}{2}$	$4\frac{5}{16}$
$3\frac{1}{2}$	$\frac{5}{32}$	$5\frac{1}{8}$	$4\frac{3}{4}$	$\frac{1}{8}$	$5\frac{3}{16}$	$4\frac{11}{16}$	—	$1\frac{5}{8}$	$4\frac{3}{4}$
4	$\frac{5}{32}$	$5\frac{11}{16}$	$5\frac{3}{16}$	$\frac{1}{8}$	$5\frac{3}{4}$	$5\frac{1}{8}$	—	$1\frac{3}{4}$	$5\frac{1}{8}$

All dimensions are given in inches.

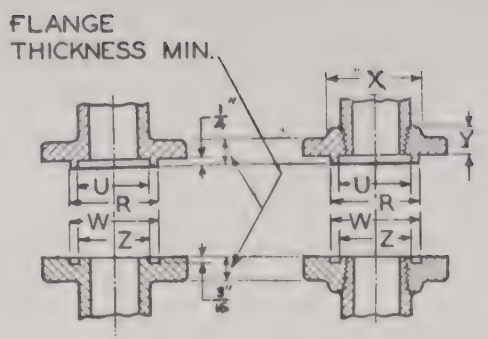
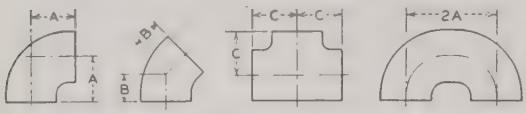


Table 12b. Standard Facing Dimensions (Round Flanges)

Nominal pipe size	Tongue			Groove			Flange thickness, min		Diameter of bolt circle	Diameter hub, min X	Length through hub, min Y
	Height	Outside diameter R	Inside diameter U	Depth	Outside diameter W	Inside diameter Z	Tongue	Groove			
5	$\frac{1}{4}$	$7\frac{5}{16}$	$6\frac{5}{16}$	$\frac{3}{16}$	$7\frac{1}{8}$	$6\frac{1}{4}$	$1\frac{5}{8}$	$1\frac{9}{16}$	$9\frac{1}{4}$	7	$1\frac{7}{8}$
6	$\frac{1}{4}$	$8\frac{1}{2}$	$7\frac{1}{2}$	$\frac{3}{16}$	$8\frac{9}{16}$	$7\frac{7}{16}$	$1\frac{11}{16}$	$1\frac{5}{8}$	$10\frac{5}{8}$	$8\frac{1}{8}$	$1\frac{5}{16}$
8	$\frac{1}{4}$	$10\frac{5}{8}$	$9\frac{3}{8}$	$\frac{3}{16}$	$10\frac{11}{16}$	$9\frac{5}{16}$	$1\frac{7}{8}$	$1\frac{13}{16}$	13	$10\frac{1}{4}$	$2\frac{3}{16}$
10	$\frac{1}{4}$	$12\frac{3}{4}$	$11\frac{1}{4}$	$\frac{3}{16}$	$12\frac{13}{16}$	$11\frac{3}{16}$	$2\frac{1}{8}$	$2\frac{1}{16}$	$15\frac{1}{4}$	$12\frac{5}{8}$	$2\frac{3}{8}$
12	$\frac{1}{4}$	15	$13\frac{1}{2}$	$\frac{3}{16}$	$15\frac{1}{16}$	$13\frac{7}{16}$	$2\frac{1}{4}$	$2\frac{3}{16}$	$17\frac{3}{4}$	$14\frac{3}{4}$	$2\frac{9}{16}$

All dimensions are given in inches.
[1] See Introductory Note 10.

Table 13. Dimensions of Butt Welding Fittings (ASA B16.9-1940), inches



Nominal size	A	B	C
¾*	1⅛	—	—
1	1½	⅞	1½
1¼	1⅞	1	1⅞
1½	2¼	1⅛	2¼
2	3	1⅜	2½
2½	3¾	1¾	3
3	4½	2	3⅜
3½	5¼	2¼	3¾
4	6	2½	4⅛
5	7½	3⅛	4⅞
6	9	3¾	5⅝
8	12	5	7
10	15	6¼	8½
12	18	7½	10

Thickness of metal conforms to either Schedule-40 or Schedule-80 pipe.
* Not covered in ASA 16.9-1940.

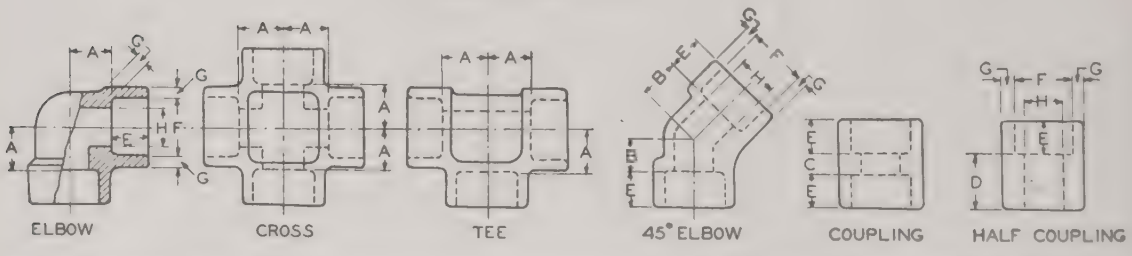
Dimensions for these fittings are given in **Table 17.**

5. Expansion and contraction of pipes take place with changes of temperature, and the piping system must be physically flexible enough to take up these changes in length by deformation, without being unduly stressed. The amount of such change in length, in inches per 100 linear feet, is given in **Table 18.** The Piping Code provides that slip-type expansion joints shall not be used with refrigerants, but that flexibility shall be provided by pipe bends, offsets, or changes in direction of the pipe line itself, also that supports shall be so arranged that they do not unduly restrain the thermal expansion or contraction of the pipe.

So long as the piping system is in one plane from anchor point to anchor point, changes of length must be taken up in bending the pipe, but if, as is usually the case, it is possible to run the pipe in three dimensions, a large part of the deformation can be taken up in torsion. The force required to deform the pipe must be supplied by the anchor points, which must be amply strong for this. These anchor points are ordinarily the equipment to which the pipe is attached. The more flexible the system,

Table 14. Dimensions of Steel Socket Welding Fittings

(Proposed ASA B16.11)*



Nominal pipe size	A	B	C	D	E	F	G		H	
							Sch. 40	Sch. 80	Sch. 40	Sch. 80
⅛	7/16	5/16	¼	5/8	3/8	0.420	0.125	0.125	0.269	0.215
¼	7/16	5/16	¼	5/8	3/8	0.555	0.125	0.139	0.364	0.302
⅜	17/32	5/16	¼	1 1/16	3/8	0.690	0.125	0.158	0.493	0.432
½	5/8	7/16	⅜	⅞	½	0.855	0.136	0.184	0.622	0.546
¾	¾	½	⅜	1 3/16	½	1.065	0.141	0.193	0.824	0.742
1	⅞	9/16	½	1⅛	½	1.330	0.166	0.224	1.049	0.957
1¼	1 1/16	11/16	½	1 3/16	½	1.675	0.175	0.239	1.380	1.278
1½	1¼	13/16	½	1¼	½	1.915	0.181	0.250	1.610	1.500
2	1½	1	¾	1⅝	⅝	2.406	0.193	0.273	2.067	1.939
2½	1⅝	1⅛	¾	1 11/16	⅝	2.906	0.254	0.345	2.469	2.323
3	2¼	1¼	¾	1¾	⅝	3.535	0.270	0.375	3.068	2.900

* Standard also covers Sch.-160 fittings.

the smaller the reaction on the equipment.

Since the advent of high temperatures in steam practice in the 1920's, an extensive literature on the analysis of stresses due to expansion of piping has been built up, largely in the Transactions of the ASME. However, for the relatively small changes in temperature ordinarily encountered in refrigeration, which seldom are as great as 200 F between erection and operation, the adequacy of provisions for expansion or contraction can usually be determined by inspection.

Supports for piping should be provided at intervals not exceeding 40 times the outside diameter of the pipe. In addition to supporting the weight of the pipe, including contents and insulation, it is necessary to provide **sway-bracing** to minimize vibration. If the pipe is pitched to drain, the pitch is determined by the frequency of supports, so that the deflection between supports does not introduce pockets which will not drain. Care must be taken that the supports do not unduly restrain the expansion or contraction, or introduce excessive stresses at joints or equipment, and where expansion stresses are at all critical, design of supports is equally important with providing the proper path. The Piping Code provides for certain minimum dimensions for hangers to guard against mechanical damage, corrosion, etc.

6. Protection against damage is necessary, particularly for small lines. Small insulated lines have a false appearance of strength. Where considerable traffic is adjacent, it is necessary to provide **protection** from **impact** from carelessly handled handtrucks, overhanging loads, ladders and the like. The Piping Code provides that refrigerant piping crossing an open space which affords passageway shall be not less than 7½ ft above the floor unless against the ceiling, and that it shall not be placed in public hallways, lobbies, stairways, elevator or dumbwaiter shafts, except that it may pass across a public hallway if there are no joints in the section in the hallway, and if non-ferrous tubing 1 in. nominal size and less is contained in a rigid metal pipe. It also provides that all refrigerant pipe, with or without insulation, be exposed to view, except for mechanical protection specified in the Code, or when located in a cabinet of a unit system.

7. Control of flow may be either the regulation of the amount of flow, as by an expansion valve, or the shutting off of flow, as by a stop valve. Either may be hand-controlled or automatic. The latter may be controlled by liquid level, temperature or pressure, and is treated in Chapter 22. In any event, control is accomplished by restricting or closing the opening through which the fluid passes.

The Piping Code provides that refrigerant valves shall be of the globe, angle or needle type, gate valves and cocks being barred for refrigerants except in industrial plants having competent operating men constantly in charge. Gate valves and cocks are, however, in common use as stop valves for brine, water and other fluids. Refrigerant valves having stuffing boxes or valve

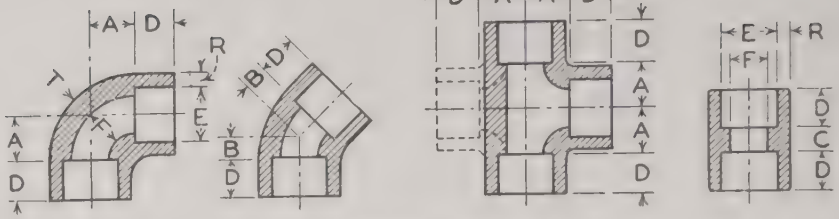
Table 15. Dimensions of Welding Flanges (ASA B16e-1939) inches

Nominal size	Y	
	150 psi	300 psi
½	1 7/8	2 1/16
¾	2 1/16	2 1/4
1	2 3/16	2 7/16
1¼	2 1/4	2 9/16
1½	2 7/16	2 11/16
2	2½	2¾
2½	2¾	3
3	2¾	3⅛
3½	2 13/16	3 3/16
4	3	3⅜
5	3½	3⅞
6	3½	3⅞
8	4	4⅜
10	4	4⅝
12	4½	5⅛

Outside diameter, facing, drilling and bolting of 150-psi and 300-psi steel flanges correspond respectively to 125-psi cast iron (except steel has 1/16-in. raised face) and 250-psi cast iron flanges, given in Fig. 2 and Table 11.

Welding neck refrigerant flanges to match manufacturers' standards are generally available.

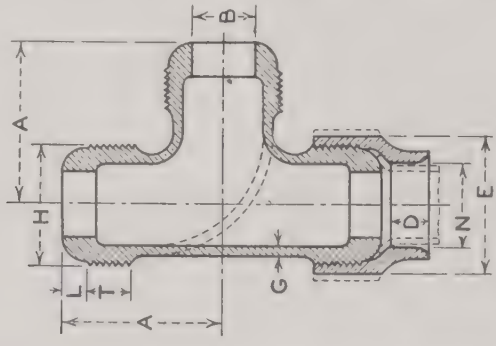
Table 16. Dimensions of Cast Brass Soldered Joint Fittings (ASA A40.3 - 1941), inches

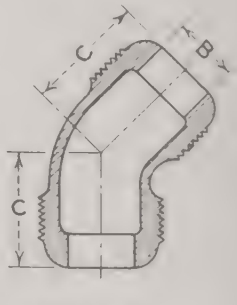


Nominal size	A	B	C	D	E (min)	F	T	R
$\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{1}{16}$	$\frac{5}{16}$	0.378	0.31	0.08	0.048
$\frac{3}{8}$	$\frac{5}{16}$	$\frac{3}{16}$	$\frac{1}{16}$	$\frac{3}{8}$	0.503	0.43	0.08	0.048
$\frac{1}{2}$	$\frac{7}{16}$	$\frac{3}{16}$	$\frac{1}{8}$	$\frac{1}{2}$	0.628	0.54	0.09	0.054
$\frac{3}{4}$	$\frac{9}{16}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{3}{4}$	0.878	0.78	0.10	0.060
1	$\frac{3}{4}$	$\frac{5}{16}$	$\frac{1}{8}$	$\frac{15}{16}$	1.1285	1.02	0.11	0.066
$1\frac{1}{4}$	$\frac{7}{8}$	$\frac{7}{16}$	$\frac{1}{8}$	1	1.3785	1.26	0.12	0.072
$1\frac{1}{2}$	1	$\frac{1}{2}$	$\frac{1}{8}$	$1\frac{1}{8}$	1.629	1.50	0.13	0.078
2	$1\frac{1}{4}$	$\frac{9}{16}$	$\frac{3}{16}$	$1\frac{3}{8}$	2.129	1.98	0.15	0.090
$2\frac{1}{2}$	$1\frac{1}{2}$		$\frac{3}{16}$	$1\frac{1}{2}$	2.629	2.46	0.17	0.102
3	$1\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{16}$	$1\frac{11}{16}$	3.129	2.94	0.19	0.114
$3\frac{1}{2}$	2	$\frac{7}{8}$	$\frac{3}{16}$	$1\frac{15}{16}$	3.629	3.42	0.20	0.120
4	$2\frac{1}{4}$	$\frac{15}{16}$	$\frac{1}{4}$	$2\frac{3}{16}$	4.129	3.90	0.22	0.132
5	$3\frac{1}{8}$	$1\frac{7}{16}$	$\frac{1}{4}$	$2\frac{11}{16}$	5.129	4.87	0.28	0.168
6	$3\frac{5}{8}$	1	$\frac{1}{4}$	$3\frac{1}{8}$	6.129	5.84	0.34	0.204

Standard also includes external shoulder fittings, reducing fittings, caps, plus bushings and fittings with pipe thread end; wrought fittings, clearances, tolerances; ratings, markings and materials.

Table 17. Dimensions of Brass Fittings for Flared Copper Tubes (ASA A40.2-1936), inches





45° ELBOW

Nominal size O.D. of tube	$\frac{3}{8}$ $\frac{1}{2}$	$\frac{1}{2}$ $\frac{5}{8}$	$\frac{3}{4}$ $\frac{7}{8}$	1 $1\frac{1}{8}$	$1\frac{1}{4}$ $1\frac{3}{8}$	$1\frac{1}{2}$ $1\frac{3}{8}$	2 $2\frac{1}{8}$
A	1.42	1.53	1.78	2.09	2.28	2.56	3.06
B	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	2
C	1.06	1.12	1.28	1.44	1.58	1.75	2.06
D	0.19	0.25	0.38	0.50	0.63	0.75	1.00
E	1.08	1.22	1.48	1.90	2.16	2.56	3.26
G	0.09	0.09	0.10	0.11	0.12	0.13	0.15
H	0.875	1.000	1.250	1.625	1.875	2.250	2.875
L	0.25	0.26	0.28	0.30	0.32	0.34	0.38
N	0.52	0.65	0.90	1.15	1.41	1.66	2.16
T	0.43	0.43	0.50	0.56	0.62	0.69	0.81
Threads per inch	14	14	14	12	12	12	12

Standard also gives rating, material, marking, details of threading, and tolerances.

stem packing must be of the back-seating type, to permit repacking under pressure. Stop valves are required by the Code as follows:

In systems having more than 20 but less than 100 lb of refrigerant:

a. At each inlet and each outlet pipe of each compressor.
b. At each outlet of each liquid receiver.

In systems having 100 lb or more of refrigerant:

a. At each inlet and each outlet pipe of each compressor.
b. At each inlet and each outlet pipe of each liquid receiver.
c. At each liquid and each suction branch header.

Stop valves used with soft annealed copper tubing or hard drawn copper tubing $\frac{3}{4}$ in. nominal size or smaller must be securely mounted independent of tubing fastening or supports. Stop valves placed where it is not obvious what they control must be suitably labelled.

Expansion valves are ordinarily needle valves, and the bore should be small enough to require a considerable opening of the valve for normal flow, so that adjustment is not too delicate.

Convenience for operation and maintenance is a criterion of design, and will tell the experienced engineer at a glance the competence of the designer. Valves should be so located that they are convenient to operate, joints should be easy of access for maintenance, location of lines should be such that they are easy to erect, easy to trace and present a neat appearance, and still do not obstruct or otherwise interfere with the use of adjacent space or present hazards such as low head room, projecting valve stems or the like.

Table 18. Expansion (or Contraction) of Piping in inches/100 ft from 50 F

Temperature, F	Cast iron	Wrought iron	Steel	Brass and copper
−50	−0.70	−0.76	−0.73	−1.10
−25	−0.53	−0.58	−0.56	−0.85
0	−0.36	−0.40	−0.38	−0.57
25	−0.18	−0.20	−0.19	−0.29
50	0.00	0.00	0.00	0.00
100	+0.36	+0.39	+0.38	+0.58
150	+0.74	+0.81	+0.77	+1.18
200	+1.14	+1.25	+1.19	+1.81
250	+1.56	+1.71	+1.63	+2.45
300	+1.99	+2.18	+2.09	+3.10

8. Restrictions imposed by **codes, laws and regulations** applying at the site of an installation must be known to the designer and obeyed. The Safety Code for Mechanical Refrigeration and the Code for Pressure Piping are primarily drawn up as guides for safe practice, and should be followed whether they have been given the force of law in a particular community or not. Their principal requirements as they apply to refrigeration piping are practically identical and are outlined in this chapter. They are the basis of most Municipal and State Codes, and will become more so as time goes on, but some ordinances go beyond them in requiring heavier piping and in other special features, and it is necessary for the designer to become familiar with the specific requirements applying at the particular place where the piping is to be installed.

17. OILS AND LUBRICATION

ORDINARILY, the selection of the proper lubricating oil for a given type of refrigeration equipment will be the responsibility of the manufacturer. Only the manufacturer has the necessary laboratory resources and test facilities to make a proper evaluation of all the necessary factors. Servicing of refrigeration equipment should follow manufacturers' recommendations; it is only in extraordinary instances that the individual engineer will find it necessary to make the selection. Therefore, this chapter is written primarily from the standpoint of the manufacturer, giving the basic reasons affecting the original choice.

In general, refrigeration equipment falls into two classes: (1) compressors with shaft seal and external motor drive; and (2) the hermetically sealed unit compressor with direct shaft drive and internal motor operating directly in the oil and refrigerant mixture. The latter type is distinguished by having the motor heat liberated within the compressor shell, whereas in the former type the motor heat is liberated externally. In the shaft seal type, oil temperatures seldom exceed 120 F; in the hermetically sealed type, they may reach 225 F under severe operating conditions. Furthermore, in the shaft seal type of compressor it is relatively easy to pump down the unit, remove the oil charge and replace with fresh dehydrated oil. For systems using Freon refrigerants, the oil charge is usually replaced whenever there is danger of contamination due to opening the system or other cause, whereas on systems using ammonia as a refrigerant, it is customary to replace the oil periodically.

Since the hermetically sealed unit compressor is welded in steel, it is not an easy matter to remove the refrigerant charge and remove the oil; therefore it is the usual practice to consider the original oil charge as the lubricant for the operating life of the machine. These types of machines are usually considered to have a minimum operating life of at least ten years and this consideration, together with the high operating temperatures, has placed entirely new demands upon the chemical stability of the lubricant used.

Furthermore, the design of some types of hermetically sealed units, such as the single vane rotary, requires the lubricant to perform much more efficiently as a sealing agent. In the reciprocating design, the lubricant film is spread over the entire area of contact between the piston and the cylinder wall, so that there is a very large sealing area to resist leakage of refrigerant from the high to the low-pressure side. In the single vane rotary type, however, the only sealing area is a line contact between a vane and a roller, yet this oil film is expected to prevent leakage just as efficiently as in the older type of reciprocating compressors. When it is borne in mind that in all instances, the viscosity and sealing power of oil are diluted to some extent by solution of the refrigerant in the oil, it can readily be seen that these requirements are very stringent.

In the past, refrigeration grade lubricating oils have been almost universally derived from naphthenic base crudes. They have been well refined pale oils characterized by good stability, low carbon residue, low pour point, medium flash and fire, and low viscosity index. They have been satisfactory with many millions of mechanical refrigeration units. In recent years, however, the extreme stability demands associated with the design of the hermetically sealed compressor unit, as outlined above, have required improvements in certain characteristics which were not inherently available in these naphthenic oils. Specifically, the demand has arisen for a higher viscosity index, greater chemical stability, and lower solvent power for the halogenated type of refrigerant. These properties, and particularly the property of high vis-

cosity index, could only be obtained from crudes having a high paraffin content. Oils derived from such crudes, when compared to the refrigeration oils derived from the naphthenic base crudes, have always been characterized by relatively high wax content and a high pour point. However, it has been found possible to overcome in great measure the handicap of a high pour point and to develop an oil for refrigeration purposes which has had very successful use in the refrigeration field.

1. Selection of Type of Oil

Table 1 shows a complete summary of the properties of the two types of oils previously discussed. Those with long experience will recognize, however, that it is impossible to select the proper lubricating oil for a refrigeration system on the basis of laboratory data. Successful refrigeration engineers insist on a complete series of **performance tests,** since changes in mechanical design often have an unsuspected influence on the behavior of the lubricant. It is also advisable to conduct a long term life test on the oil, in the unit in which it is to be used, under severe operating conditions. A representative test of this kind is a life test at 110 F ambient temperature for a period of two years. During this time, the unit is carefully checked at frequent intervals for efficiency of operation, and at the conclusion of the test, it is torn down and carefully inspected for gum, varnish, sludge, corrosion and copper plating, and wax deposits. With this information available, the final selection can then be made with a considerable degree of confidence.

In this final selection, **cost** will be an important item. The high viscosity index oils may be somewhat more expensive than those of low viscosity index derived from naphthenic crudes. It is unlikely that they

Table 1. Refrigeration Oil Properties of Naphthenic Base Oils

Manufacturers' brand no	1	2	3	4	5	6
Specific gravity at 60 F	.902 .913	.904 .916	.913–.925	.919–.932	.923 .937	.929–.914
Flash point, ASTM open cup, F	275-285	300-310	330–340	350–360	370 380	380–395
Fire point, F	315–325	345–355	380–390	410–420	430–445	440–460
S.U. viscosity at 100 F, sec	75–80	100–115	150–160	280–300	500–600	750–800
Color, NPA	1–1½	1–1½	1½–2	1½–2	1½–2	1¾–2¼
Neut. no ASTM Mg KOH per gm	0.00	0.00	0.00	0.00	0.00	0.00
Pour test, ASTM F, max	−45	−40	−35	−25	−10	−5
Sligh oxidation	10–20	10–20	10–20	10–20	5–15	5–15
Steam emulsion, sec	20–40	20–40	30–60	40–80	60–100	70–110
Dielectric strength, kv min	25	25	25	25	25	25

High Viscosity Index Refrigeration Oils

Manufacturers' brand no	27	34	41
Gravity, API (ASTM D287-39)	31.9	30.5	29.2
Flash, C.O.C., F (ASTM D92-45)	385	450	470
Fire, C.O.C., F (ASTM D92-45)	445	515	545
Color, Saybolt (ASTM D156-38)	16	—	—
Color, NPA (ASTM D145–45T)	—	1½–	1½–
Pour point, F (ASTM D97-39)	5	5	5
Viscosity, S.S.U. @ 100 F (ASTM D88-44)	156	310	504
Acid no (ASTM D663-44T)	L.T. 0.05	L.T. 0.05	L.T. 0.05
Steam emulsion no (ASTM D157-36)	60	100	150
Oxidation stability, hr*	1000 plus	1000 plus	1000 plus

* Proposed ASTM Method for Turbine Oils (turbine oil stability test).

will displace those oils from applications where they have performed with entire satisfaction in the past. Under severe operating conditions, the higher cost high viscosity index oils may show an actual saving when all factors are taken into consideration.

2. Selection of Oil Viscosity

The selection of the proper oil viscosity for use in a refrigerating compressor depends upon a number of factors, such as the mechanical **design of the compressor,** the **type of refrigerant** used, whether water-cooled or air-cooled, as well as the **type of service.** The lubricating oil in a refrigerating compressor performs the functions of sealing and lubrication. In most instances, bearing loads in refrigerating compressors are not high and the lubrication function is relatively easy to perform. The problem of sealing is not always so easy to accomplish. It depends to a considerable extent on the pressure differences involved and the solubility of the refrigerant in the lubricating oil.

Some refrigerants, particularly the halogenated hydrocarbon derivatives, are completely miscible with oil in all proportions. Others, such as sulfur dioxide, Freon-114 and Freon-22, are completely miscible above critical solution temperatures but are only partially miscible at lower temperatures. Since the tendency of the refrigerant to dissolve in the oil determines the viscosity of the resultant solution, the choice of the initial viscosity of the lubricating oil must be governed accordingly.

As a general rule, it can be stated that an oil should be chosen of the **lowest viscosity** which will give the necessary **sealing properties** with the refrigerant used and over the range of temperatures and condensing pressures encountered. For these reasons, it is desirable to know the actual viscosity of the oil-refrigerant mixture at operating temperatures. Studies have been made by Rutledge[1] on several oils and refrigerants, and the data for oil and Freon-12 are shown in the curves in **Figs. 10, 11** and **12.** Additional data on oil-Freon-12 solubility

Table 2. Absorption of Ammonia in Oil

(Volume of gas absorbed by unit volume of oil)

Pressure, psi	Temperature		
	32 F	70 F	150 F
14.7	3.14	2.20	1.24
29.3	6.34	4.44	2.39
44.0	10.05	6.60	3.56
58.7	15.66	8.53	4.69
146.7	—	—	12.43

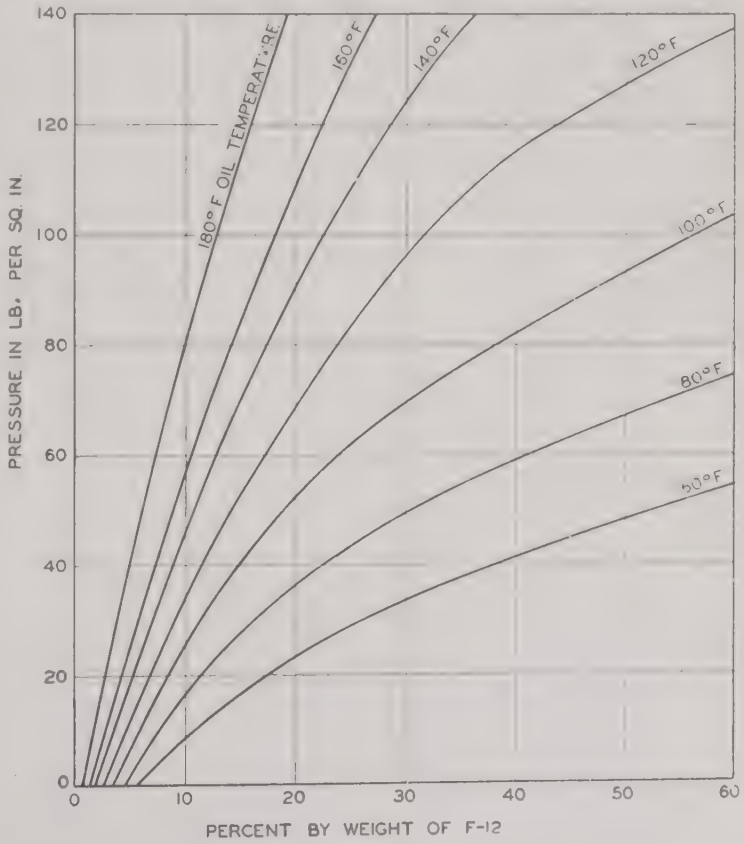


Fig. 1. Solubility of Refrigerant F-12 in Oil as a Function of Oil Temperature and Refrigerant Pressure

Table 3. Lubrication Recommendations

A. Small Systems

Refrigerant	Type of compressor	S.U. at 100 F oil viscosity	Typical brand (Table 1)
Ammonia	Reciprocating	150–300	3, 4, 27, 34
Carbon dioxide	Reciprocating	280–300	4, 34
Sulfur dioxide	Reciprocating	70–200	1, 2, 3, 27, 34
Sulfur dioxide	Rotary	280–300	4, 34
Methyl chloride	Reciprocating	280–300	4, 34
Methylene chloride (Carrene No. 1)	Centrifugal	280–300	4, 34
Methylene chloride (Carrene No. 1)	Rotary	150–300	3, 4, 27, 34
Freon-11 (Carrene No. 2)	Centrifugal	280–300	4, 34
Freon-12	Centrifugal and reciprocating	280–300	4, 34
Freon-21	Reciprocating	280–300	4, 34
Freon-113	Centrifugal	280–300	4, 34
Freon-114	Rotary	280–300	4, 34

B. Industrial Refrigeration

(Ammonia and carbon dioxide compressors with splash, force-feed or gravity circulating systems)

Type of compressor	S.U. viscosity range	Typical brand (Table 1)
Where oil may enter refrigeration system or compressor cylinders	150–300 at 100 F	3, 4, 27, 34
Where oil is prevented from entering system or cylinders:		
In force feed or gravity systems	500–600 at 100 F	5, 41
In splash systems	150–160 at 100 F	3, 27
Steam-driven compressor cylinders when condensate is reclaimed for ice making	140–165 at 210 F	Pittsburgh cylinder oil

C. Miscellaneous Equipment

Type of requirement	S.U. viscosity at 100 F range	Typical brand (Table 1)
Bearings:		
Ring oiled, normal temperature	280–300	4, 34
Ring oiled, low temperature	100–115	2
Chain oiled	280–300	4, 34
Ball and roller bearings:		
Oil lubricated	280–300	4, 34
Grease lubricated	—	No 843 roller-bearing grease
Wick oilers	280–600	4, 5, 34, 41

and viscosity are shown in **Figs. 1, 8,** and **9.**

A practical method for determining the minimum safe viscosity is to determine the total volumetric efficiency of a given compressor system, using several oils of widely varying viscosity. Choose the oil of lowest viscosity giving a satisfactory volumetric efficiency. It is necessary to cite certain precautions in making these tests. They should be run at various ambient temperatures; for example, 70, 90, and 110 F. It has been observed that high viscosity index oils sometimes give as much as 5% more volumetric capacity at 110 F ambient conditions than do low viscosity index oils having a similar viscosity rating at 100 F.

The viscosity index of an oil is an indication of its tendency to reduce viscosity with increasing temperature. A low viscosity index oil may have an index as low

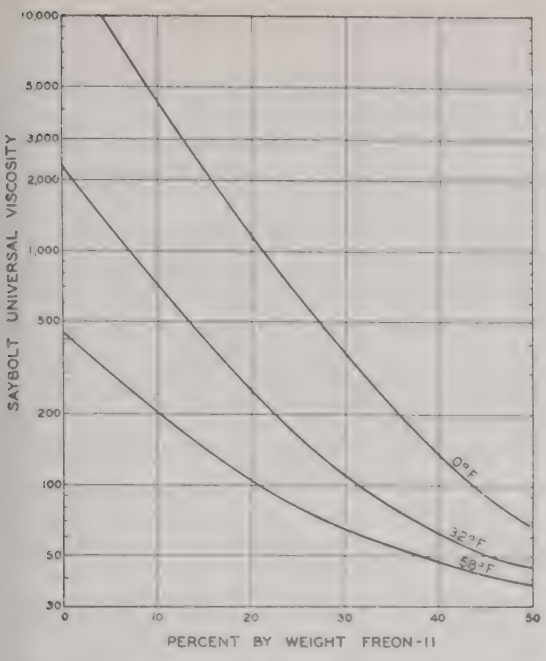


Fig. 2. Viscosity of Oils of Fig. 1 with Various Percentages of Freon-11

as 0, a high viscosity index may be 100. Fig. 16 shows the change of viscosity with temperature of two oils having viscosity indices of 0 and 92, but identical viscosity of 500 seconds S.U.V. at 100 F. Once the viscosity of the oil has been chosen, many other considerations become important, some of which are not covered by the usual lubricating oil specifications. These other considerations may be considered point by point and the governing factors outlined.

Figs. 10, 11, and 12 give the **viscosity** and **dilution ratio characteristics** of oils with Freon-12. If the temperature and pressure of the system are known, Fig. 11 will give the composition by volume of the liquid phase. Using this figure and the value of the viscosity of the pure oil at the given temperature, Fig. 10 gives the viscosity of the mixture and an approximate value for the **specific gravity.** The **absolute viscosity** in centipoises may then be obtained by multiplying the kinematic viscosity in centistokes by the specific gravity.

A numerical example follows: Given an oil whose viscosity at 80 F is 450 Saybolt Seconds, and operating conditions of 40 psi Freon pressure at this same temperature. On Fig. 11 the volume ratio is given as 0.12. Fig. 10 shows that when the oil is diluted to this extent its viscosity is reduced to 105 Saybolt Seconds or 22 centistokes. Since the specific gravity is increased from 0.88 to 0.92, this is equivalent to 20.2 centiposes absolute viscosity.

3. Pour Test and Cloud Point

It has been generally considered in the past that oils for refrigeration application require low pour test and low cloud point. Recent developments have indicated that this is not categorically true. In the past, most refrigeration oils have been refined naphthenic base stocks. These naphthenic base oils have lower cloud and pour points than paraffinic or Mid-continent base oils and usually do not require dewaxing for lubrication applications. Paraffinic or Mid-continent base oils have high wax contents and require considerable dewaxing in order to produce lubricants that are satisfactory

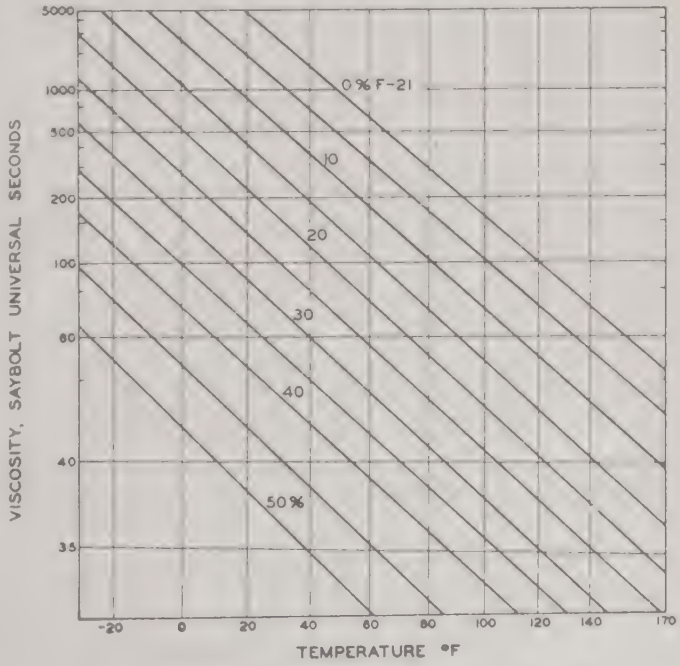


Fig. 3. Viscosity Temperature Curves of Solution of Freon-21 in Oil

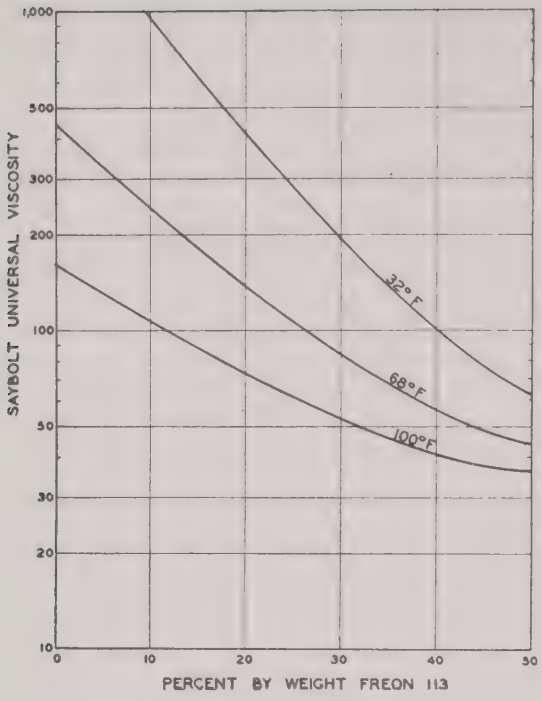


Fig. 4. Viscosity with Various Percentages of Freon-113

for refrigeration use. The advent of the modern dewaxing processes, however, has made it practical to dewax such oils very satisfactorily. However, the suitability of such oils for refrigeration applications cannot be judged by the usual criteria of cloud point, pour point and so-called floc point.

It should be pointed out that in a refrigerating compressor some oil is always circulated with the refrigerant, but in a well designed compressor this quantity usually does not exceed 1 or 2% of the refrigerant circulated, and in almost no instance does it exceed 10%. Consequently, it is the pour point or the cloud point of the lubricating oil mixture corresponding to that actually circulated that becomes the important criterion. It has even been found in some instances that a cloud point or a floc point of the mixture does not have too much bearing on the actual performance in the compressor. It has been observed that some dewaxed oils show a relatively high cloud point in the presence of refrigerants, but that the material in the oil that formed the cloudiness is of such a nature that it redissolves in the refrigerant very readily on a slight increase of temperature. On the other hand, some oils which show a low cloud point or floc point, separate a hard wax which is very difficult to redissolve.

4. Volatility

Volatility of lubrication oils should be such that they will not vaporize at pres-

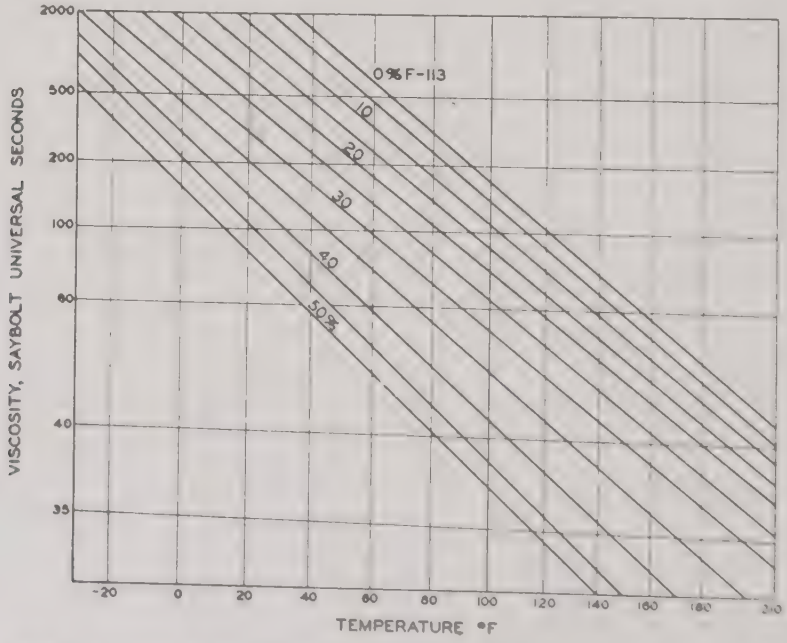


Fig. 5. Viscosity Temperature Curves of Solution of Freon-113 in Oil

sures and discharge temperatures involved in the high side of the system. Since many compressor systems are dehydrated under vacuum while containing small quantities of oil, it is also important that the oils should not be too volatile under these conditions.

Flash and **fire point** of oils used in refrigerating lubricants are not too important with most refrigerants. However, some refrigerants, such as sulfur dioxide, ammonia, and methyl chloride, have a high ratio of specific heats $(C_p)/(C_v)$ and consequently have a high temperature of adiabatic compression. Such refrigerants frequently produce carbonization of oils with low flash and fire point when operating in high ambient temperatures.

5. Moisture

It is important that the moisture content of refrigeration oils be low. Moisture in refrigeration systems cannot be tolerated, since it not only produces freeze-ups at the point of expansion but also accelerates destructive chemical changes in the refrigeration system. It is therefore necessary to require extremely low moisture contents in all materials entering into the refrigeration system. The moisture content for the specification for refrigeration oil usually requires resistance to a 25,000-volt breakdown test with electrodes 1 cm apart. When oils have once been dehydrated so that they will meet this specification requirement, it is necessary to protect them from humid atmospheres and to handle them in sealed containers.

In refrigeration manufacturing plants,

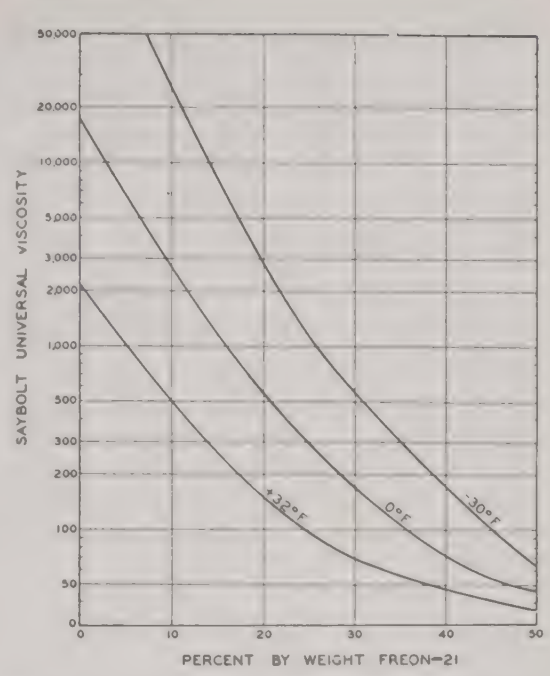


Fig. 6. Viscosity of Oil with Various Percentages of Freon-21

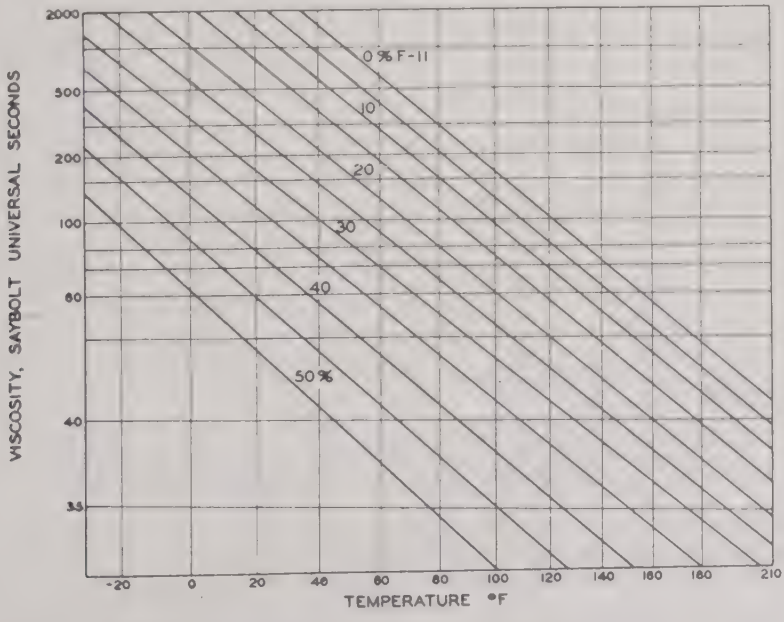


Fig. 7. Viscosity Temperature Curves of Solution of Freon-11 in Oil

oils can be dehydrated to this low level by several methods. The most commonly used method, known as **"blotter pressing,"** involves filtering the oil through carefully dehydrated filter paper. Some manufacturers also use **centrifuging** as a means of removing water. In many instances, however, where refrigeration repair work is done in the field, it is not always possible to protect the oil from moisture contamination; in such cases it is advisable to dehydrate the system by the use of a satisfactory refrigeration **dehydrating cartridge.** Such cartridges contain efficient desiccating materials and will lower the moisture content of both refrigerant and oil to a level of about ten parts of water per million parts of oil and refrigerant.

6. Stability

Refrigeration oils must be stable to **oxidation** and **heat.** The most severe oxidation exposures are found in the processing of the original equipment in the manufacturing plant. Under actual service conditions, oxidation is not a great factor in the deterioration of refrigeration oils. In a well designed and assembled refrigeration system, oxygen will be almost completely absent due to the careful evacuation of the system before charging oil and refrigerant. Since most refrigerants are inert so far as oxidation is concerned and some, such as sulfur dioxide, are definitely reducing in character, oxidation should not take place once the system is in operation. However, with the advent of sealed compressors, an increasing degree of heat stability is required in satisfactory refrigeration oils.

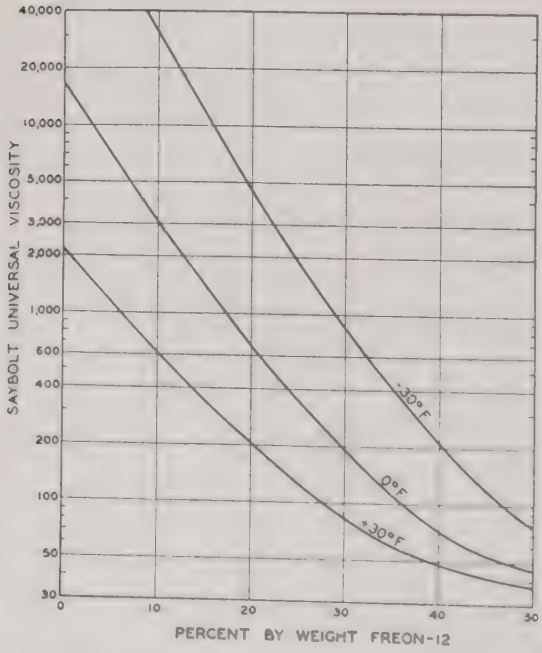


Fig. 8. Viscosity of Oil with Various Percentages of Freon-12

The average operating temperature in refrigeration systems has shown a steady increase over the years. This trend is expected to continue, especially in air conditioning systems. Many cities have already placed stringent restrictions on using water for cooling an air conditioning system. Since the design trend is more and more toward packaged units using sealed compressor systems, it is believed that the temperature requirements may maintain this upward trend. It has been observed that these high operating temperatures produce gum and varnish formations in some types of refrigeration oils. The mechanism of such varnish formation is not clearly understood since oxidizing conditions are almost completely absent. A frequently advanced explanation is that it is caused by polymerization of unsaturates catalyzed by the presence of certain metals, particularly copper.

Much work has been done in attempting to develop chemical additives which will stabilize these oils toward gum and varnish formations, but the usual type of oxidation inhibitor has not been too effective.

7. Copper Plating

Where halogenated refrigerants have been used, a phenomenon known as "copper plating" has been observed. Every refrigeration system contains some copper. In ordinary reciprocating compressor systems, the quantity may be confined very largely to the connecting lines. In sealed rotary compressors, however, the motor windings are in direct contact with the oil and refrigerant of the system. A chemical reaction takes place whereby the copper in contact with the oil and refrigerant is dissolved and goes into solution in the oil and refrigerant where it can be detected by sensitive chemical analysis. Under suitable conditions, it then plates out on the sur-

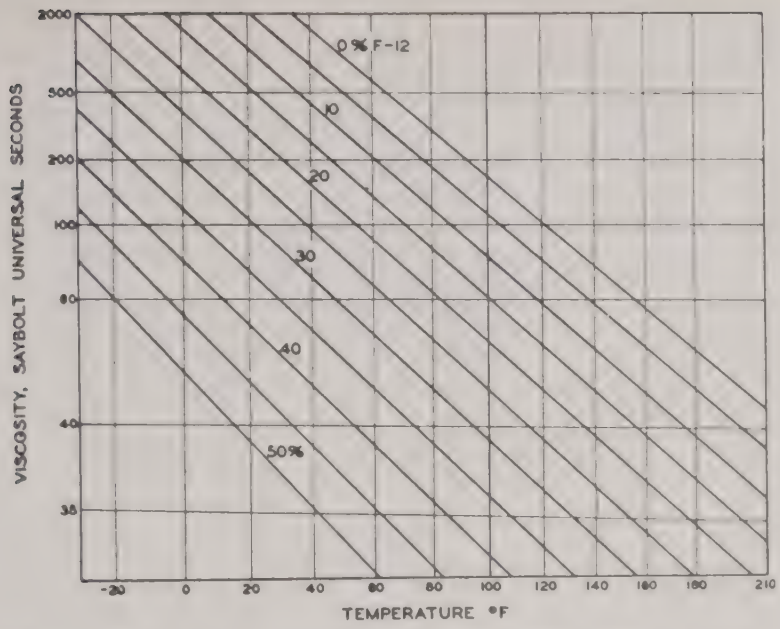


Fig. 9. Viscosity Temperature Curves of Solution of Freon-12 in Oil

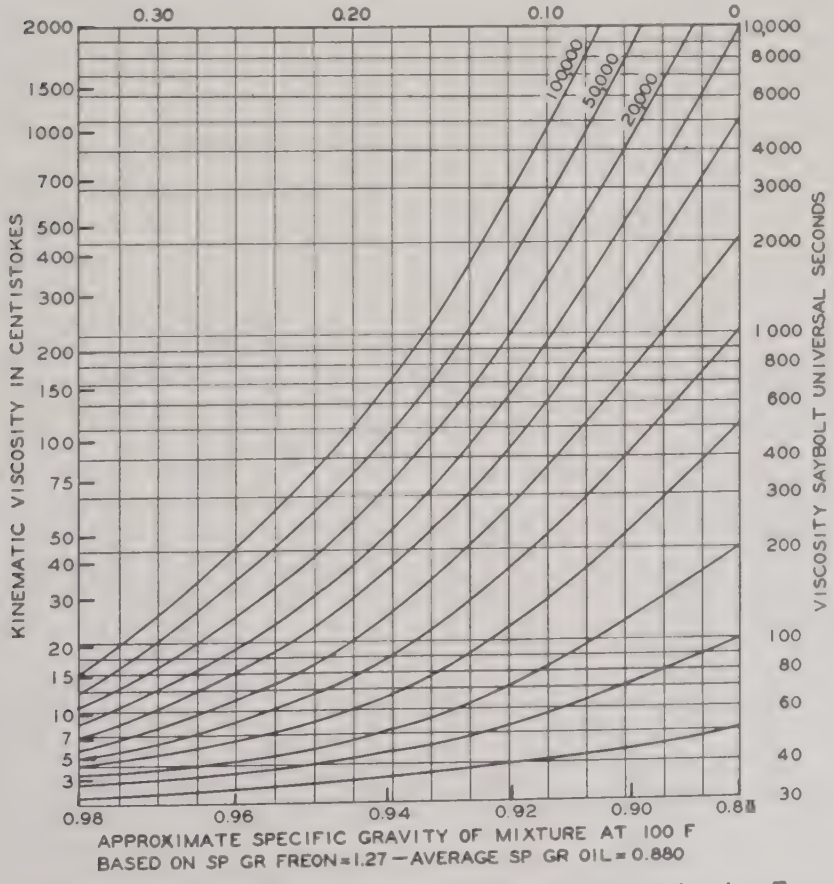


Fig. 10. Viscosity-Dilution Ratio Characteristics of Freon and Any Oil at Any Temperature—Upper Scale Shows Ratio of Freon to Oil by Volume

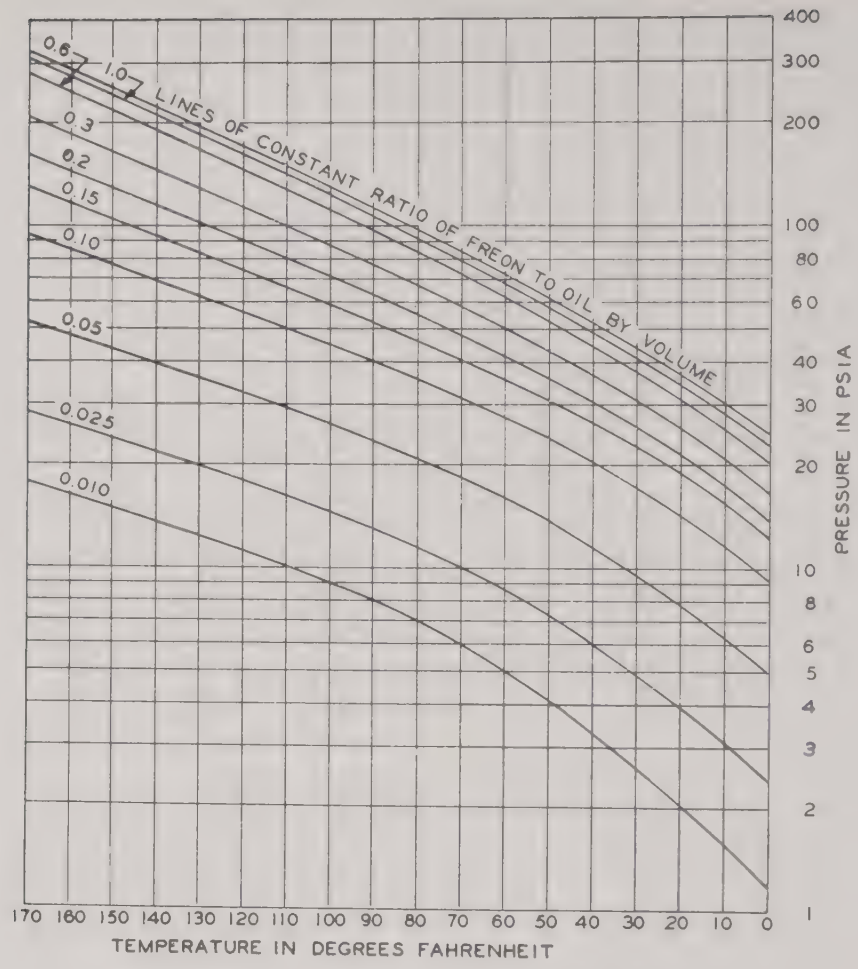


Fig. 11. Dilution Ratio Characteristics of Oil and Freon—Value of Top Line Is Infinity

faces of other metals, but most usually on steel surfaces. Furthermore, since bearing surfaces are usually the brightest, cleanest and most highly polished in the system, the deposition occurs preferentially at these points. Where tolerances are very close, the deposited copper may increase to such a thickness that binding and galling of such a surface may occur. This situation is usually not serious in the older type, slow-speed reciprocating compressors. In certain types of rotary compressors, however, where tolerances are of the order of one-tenth thousandth (0.0001) of an inch, this deposition may eventually result in the failure of the system.

Much study has been devoted to the causes and cure of this difficulty. It has been thought for some time that the lubricating oil plays an important role, and it has been demonstrated that different types of oil differ widely in their ability to dissolve and hold copper in solution. It has been a widely held theory that the copper so dissolved precipitates out in contact with steel surfaces by straight electromotive displacement. However, it has been observed in laboratory experiments that such deposition can actually take place on glass, and it is therefore believed that electromotive displacement is not always involved.

Recent work has also established the fact that copper plating can occur in the presence of the refrigerant alone without the presence of any lubricating oil. However, in oil and refrigerant mixtures, it is certain that the reaction takes place in the presence of some types of oil much more readily than it does in others, and there is no doubt that the lubricating oil is in many instances a contributing factor. It has also been established by much careful laboratory work that the reaction does not

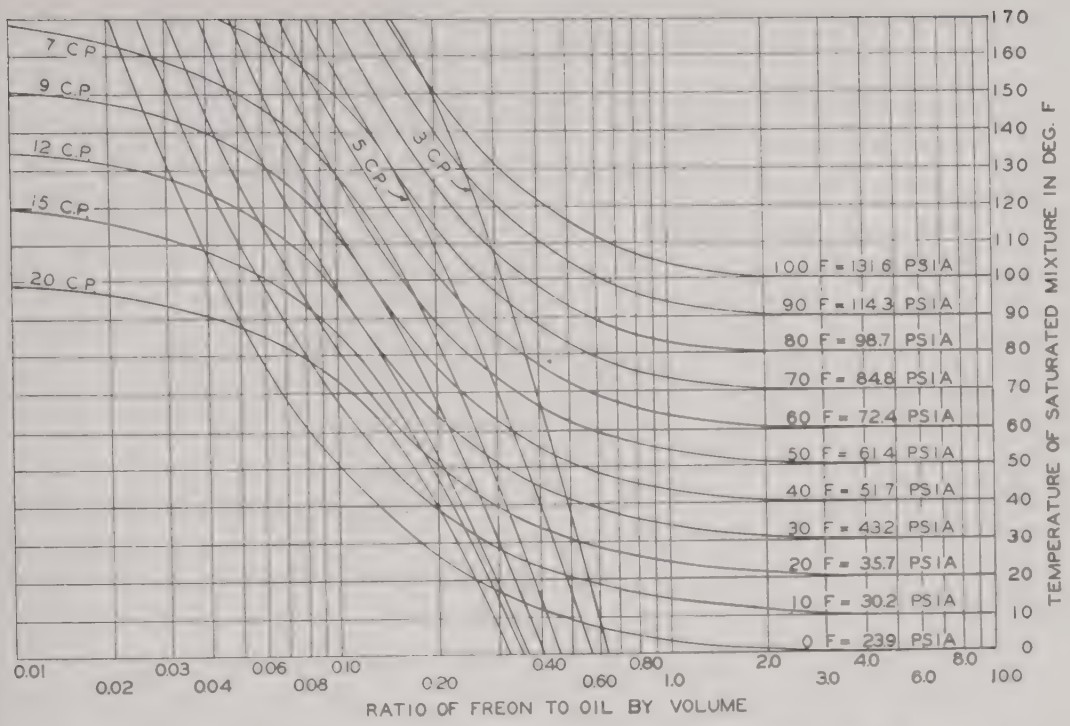


Fig. 12. Viscosity and Dilution Ratio Characteristics of Freon and a 500 Viscosity Oil in a Refrigerating System

take place readily in completely dry systems, and that most difficulties due to copper plating can be largely eliminated by **adequate drying** during the **original manufacture** of refrigerating equipment and by the use of built-in **chemical dehydrators** as an integral part of the equipment in service.

Due to observations with respect to copper plating in the presence of refrigerants alone, competent technical opinion is now turning to the belief that the refrigerant may be the dominant factor in the plating reaction. The theory that has been advanced is that small quantities of moisture present in the system produce a slow chemical hydrolysis of even the most stable halogenated refrigerant—even those of the well-known Freon type. In fact, microanalytical methods have demonstrated the presence of chloride and fluoride ions in corrosion deposits found in refrigeration systems. It is now believed that the ultimate solution of this problem may require even more stable halogenated refrigerants than those currently in use.

8. Mutual Solubility of Oil and Refrigerant

Refrigerants fall into two classifications under this heading: (1) those completely miscible with oil in the range of operating temperatures; and (2) those which exhibit a critical solution temperature in the operating range, i.e.—separate into two immiscible phases. Freon-12 is an example of the

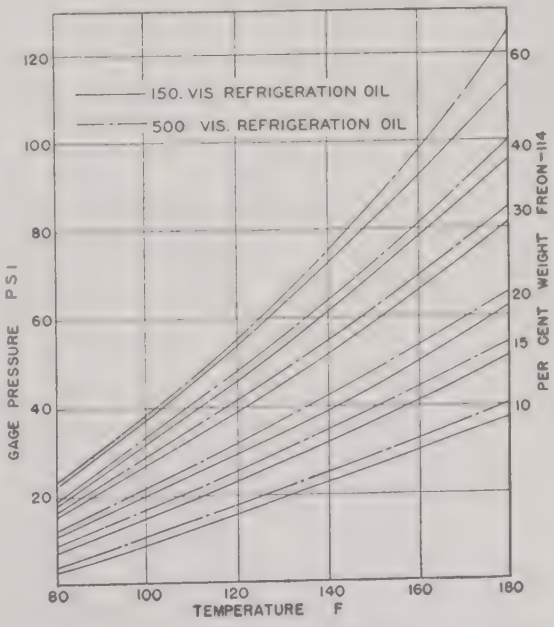


Fig. 13. Temperature Pressure Relationship of Freon-114-HVI Oil Mixtures

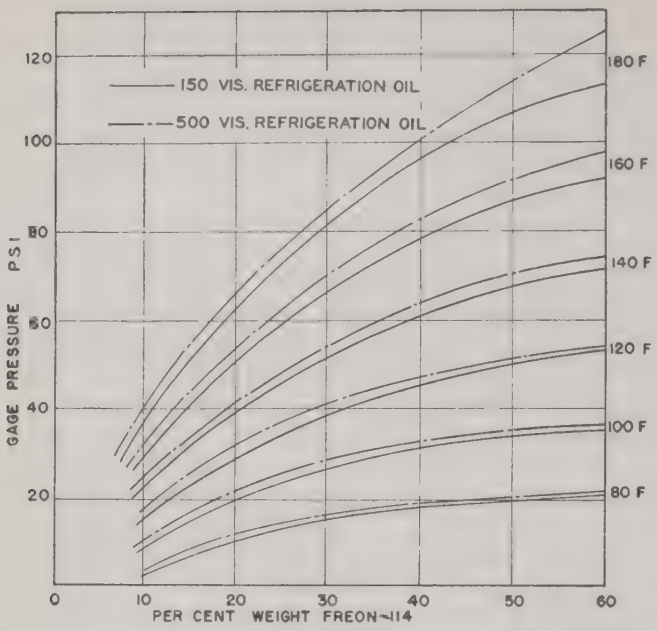


Fig. 14. Temperature-Pressure Relationship of Freon-114-HVI Oil Mixture

first class, and sulfur dioxide an example of the second. Freon-114 (tetrafluorodichloroethane) shows critical solution temperatures as high as 60 F with some oils; hence oil-F-114 mixtures are usually single-phase on the high side, and two-phase on the low side. Typical curves for these relationships are shown in **Fig. 15.**

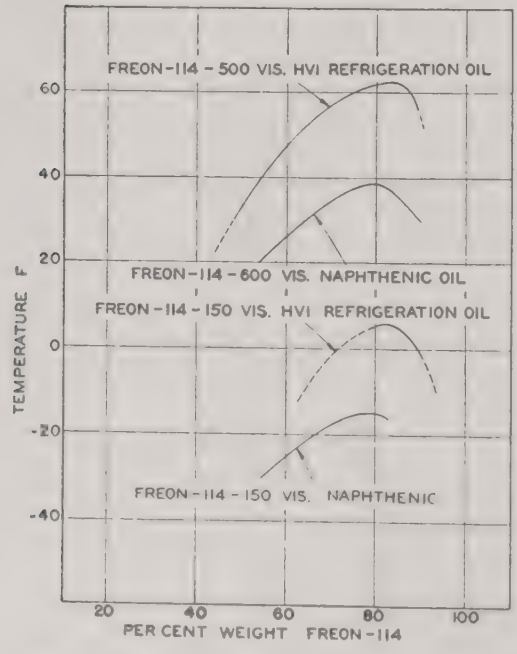


Fig. 15. Critical Solution Temperatures of Freon-114-Oil Mixtures

The amount of refrigerant which an oil will absorb from the vapor phase at a given oil temperature and refrigerant pressure is also an important criterion of refrigerant solubility. Typical curves are shown in Figs. 1 and **16.**

These solubility characteristics have an important bearing on lubricant performance. The extent to which an oil dissolves refrigerant vapor at a given temperature and back pressure determines the effective viscosity of the oil-refrigerant mixture in the machine, and hence affects its sealing power. If the oil sump is located on the low-pressure side of the machine, this characteristic also affects the amount of foaming of the oil-refrigerant mixture when the compressor starts after an off cycle.

As previously pointed out, if the critical solution temperature of the oil-refrigerant mixture is appreciably higher than the evaporator operating temperature, a two-phase system may exist in the evaporator. This condition affects "oil-logging" of the evaporator, rate of oil return, and, in some instances, may affect evaporator temperatures, particularly if it is of the flooded type. In general, it may be said that high viscosity index oils show less tendency to dissolve refrigerant from the vapor phase, and exhibit higher critical solution temperatures in the liquid phase than do the low viscosity index naphthenic base oils. It must be reiterated, however, that the effect of these characteristics can only be evaluated by actual performance tests.

9. Lubrication

There are two general **types** of lubricating ststems for refrigerator compressors—those used with open type compressors and those used with closed type compressors.

a. In open type compressors (with open crankcases), the oil is withdrawn from the refrigerant system. This method of lubrication is usually found in large horizontal industrial compressors used in ice plants

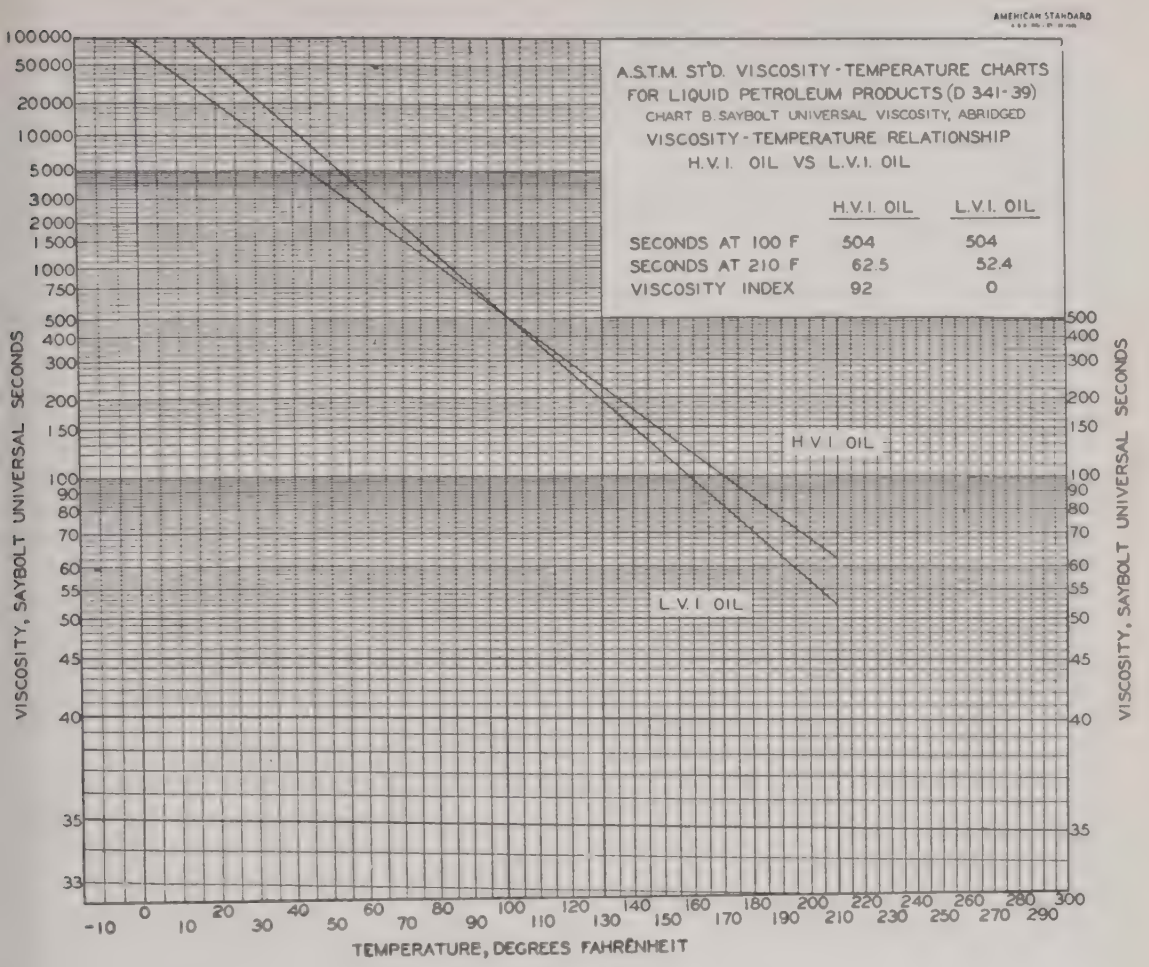


Fig. 16

or industrial processes in which separate systems for the lubrication of cylinders and bearings are used. In such cases, piston and piston rod lubrication is provided by a pressure system which injects predetermined amounts of oil into the stuffing box gland and cylinder, while bearing lubrication is provided by force feed, gravity, or splash systems. A lighter oil is used for cylinders than for the lubrication of bearings where the oil is exposed to greater pressure.

Any excess of oil for cylinder lubrication should be avoided, otherwise it may form a coating in the expansion coils and decrease the efficiency of the system. To prevent this, well designed systems are provided with an oil separator on the discharge line which separates the oil entrained in the high pressure gas as it travels to the condenser.

b. In closed type compressors, the same oil serves for the lubrication of bearings and cylinders and remains in the system. A heavier grade of refrigerating oil must be used in such machines.

All industrial compressors which have been in service for some length of time should be carefully, inspected for worn parts and, if advisable, heavier oils should be recommended to avoid an excess of oil being pumped into the expansion coils.

10. Industrial Compressors

Pressure lubrication is extensively used in large horizontal refrigerator compressors, while splash lubrication is better adapted to vertical compressors.

Pressure lubrication permits more accurate control of the amount of oil delivered to cylinder walls and bearings, but requires considerable piping and frequent

attention by the operator, who has to refill the reservoir and regulate the flow of oil.

a. Mechanical force feed lubricators. Cylinders and bearings are extensively lubricated by mechanical lubricators consisting of a small rectangular tank, divided into several compartments and located above the parts to be lubricated. A plunger mechanism is operated from cams or levers which are connected with some rotating part of the compressor. When the compressor is in operation, the plungers pump the oil through check valves to certain points in the cylinder walls, bearings and other moving parts. The amount of oil can be regulated by adjusting the plunger stroke. Frequently different grades of oil are contained in the compartments of the tank for lubricating the compressor cylinders.

Force feed lubricators for refrigerator compressors should not be open to the air or moisture will enter the system. The function of these lubricators is to meter out the lubricant and deliver it to all points to be lubricated in minute predetermined quantities. The amount supplied by each unit can be seen in the sight-glass, and any excess tends to cause carbonization at some of the hot spots frequently occurring in large industrial compressors. Care must be taken that the oil feed lines are filled before operation of the compressor begins. For this purpose a manual handcrank, which can also be operated when it is found necessary to increase the supply of oil momentarily, is usually part of the lubricator.

In large horizontal and vertical compressors with crossheads, the internal parts, cylinders, piston and rods are lubricated by mechanical lubricators with low pour test refrigerating oils. External parts such as bearings, pins, cross-head guides, governors, etc., are serviced with machine oils by an independent gravity or pressure circulating system.

In the smaller horizontal compressors, oil is introduced at the top-center of the cylinder and in double-acting compressors it is introduced in the middle of the cylinder. Automatic valves keep the oil in the feed lines by checking back pressure during the compression stroke.

In some systems the "injection method" is used. The oil is injected into the suction line and carried into the cylinders by the flow of gas, forming a film of oil over all exposed surfaces. This method is wasteful and there is always danger that excess oil will cause sticky valves.

In practically all compression machines with reciprocating parts, the oil serves also as a **seal** in the **stuffing boxes.** Hot ammonia gas has a bad effect on the packing of stuffing boxes, and sufficient lubricating oil with a high flash point should be used to prevent overheating, piston rod wear and leakage of gas. Lubrication is generally accomplished in a stuffing box by an "oil lantern" which is located between two sets of packing, and consists of hollow spaces surrounding the rod into which the oil is pumped from a reservoir. Another opening from the lantern is connected with the suction side of the compressor and any gas which leaks by the packing rings is returned to the cylinders.

In some types of double-acting compressors, the oil lantern also serves to feed the lubricating oil to the compressor cylinder. In this method, the oil penetrates past the piston rod packing and at the same time protects the packing from the bad effects of the ammonia.

Ammonia and carbon dioxide compressors are very similar to each other as far as construction and operation are concerned. The working pressure of carbon dioxide compressors is higher (from 900 to 1,000 lb.) and consequently the cylinder temperatures are also higher. In both types the lubricating oil must not carbonize at the high temperatures in the cylinders, must have sufficient body (viscosity) to supply a protective film on all frictional surfaces, and must readily drain into the traps at the low temperatures encountered.

Where CO_2 refrigerating machines are operated in the production of dry ice, a white oil may be required. Dry ice is frequently used in direct contact with food products and for this reason some operators insist on a water-white oil which will have no effect on the odor or color of the dry ice should an appreciable amount of oil pass the traps. To avoid such possible contamination of the dry ice, some operators are willing to sacrifice the slightly better lubricity of the pale oils.

b. Oil separators. Most industrial re-

frigerating systems are equipped with oil separators to aid in the separation of the oil from the refrigerant. They are generally cylindrical tanks, with baffles and a sight gage glass to indicate the level of the oil. Particles of oil mixed with the gas after it has left the compressor, are precipitated in the separator.

c. Lubricating systems. Every compressor manufacturer has his own system of lubrication which serves his equipment best. For instance, in some older types of steam driven horizontal compressors, a splash system for cylinder walls and piston lubrication is used while the external parts are oiled by oscillating mechanical lubricators. In some of the latest models oil is pumped from the crankcase into a sump, circulated under pressure and returned through filters to the crankcase. At the same time a mechanical lubricator forces engine oil to bearings and cylinders, and a third oil system is used to regulate the operation of the governor.

Another modification of the force-feed lubrication on large industrial compressors embodies a chain-driven gear pump which furnishes positive pressure lubrication. Oil from the reservoir enters the pump intake through a strainer and is discharged into a screen type filter which can be cleaned while the compressor is operating. The oil is then passed through an oil cooler and distributed through drilled passages or oil lines to the bearings, cross-head pins, guides, etc. Excess oil from the pump is bypassed to the main reservoir or to the oil sump by a relief valve. Automatic devices stop the compressor in case of oil pressure failure.

d. New compressors. A new compressor requires more oil for cylinders and bearings until the surfaces have become smooth and coated with an oil film. It is customary to break in new units with a refrigeration oil of lighter viscosity than is used in regular operation. This oil is known as "break-in" oil.

After the compressor is broken in, it is advisable to drain the oil, clean the crankcase carefully of all dirt, metallic particles, lint, etc. and refill it with clean fresh oil. After the compressor has been started again, the oil level in the reservoir should be watched and oil added until the oil level is several inches above the intake pipe, which should be marked "low level." When the compressor is shut down and the oil has drained back into the reservoir, the oil level should be marked "low level-idle" and after adding oil another level is established for "high level-idle." After the compressor is started again the resulting oil level is marked as "high level" (running.)

Frequency of oil change depends on operating conditions and time-efficiency of operation. Most manufacturers of large heavy-duty compressors recommend an oil change at least every ninety days. Instruction books furnished by each manufacturer should be followed carefully by the operating engineer.

11. Splash Lubrication

Smaller industrial and commercial vertical refrigerator compressors have enclosed crankcases. Cylinders, piston and bearings are lubricated by a splash or circulating system. In the splash system the oil (for cylinders and bearings) is stored in the crankcase, and as the crank dips into the oil during each revolution it splashes a sufficient amount of oil spray to the cylinder walls, pistons, etc. An excess of oil in the crankcase would cause violent agitation or churning of the oil by the crank, preventing precipitation of any foreign matter which might be in the system. Such excess oil might also enter the refrigerant circuit and prevent efficient operation of the condenser, evaporator and oil separator.

Refrigerating oil for splash systems must have sufficiently high viscosity at compression temperature to prevent excess oil consumption, because too light an oil will pass by the piston rings. Since air can enter commercial refrigerating systems, the oil must be resistant to oxidation and very stable, so as to reduce carbon and gum formation to a minimum.

12. Domestic and Small Systems

During the earlier stages of development of these units splash lubrication was the system most commonly used. Some trouble occurred due to "foaming" in reciprocating compressors installed in these smaller units.

Foaming is invariably caused at the start of the cycle, when return gas has been absorbed in the oil and is suddenly

released as the compressor starts lowering the pressure in the crankcase. There is always danger that foam might be carried over to the high side of the compressor with the refrigerant.

This experience led to efforts to develop several **systems of pressure lubrication** in order to obtain positive circulation of the oil through the compressor. One type of pressure lubrication is similar to the lubrication of an automotive engine by an immersed gear pump in the crankcase. The pump discharges the oil under pressure (regulated by the speed of the gears) through small feed lines to the cylinder walls, or through drilled passages to the bearings. After forming the required protective film, the surplus oil returns to the crankcase by gravity and is recirculated.

Another type of pressure lubrication makes use of a rotary pump. Two blades moving in a slotted rotor, pump the oil from a reservoir through drilled passages in the motor shaft to the bearings and cylinder walls. In some units part of the oil is used to cool the motor winding before returning to the sump.

Another system applies splash lubrication to lubricate the cylinder walls. The bearings are lubricated by a reciprocating pump with a single oscillating cylinder operated from an eccentric on the crankshaft.

As stated previously, lubrication is provided for when the manufacturer charges the unit with the refrigerant and oil before it leaves the factory. Both are expected to last and give satisfactory service for years. If an oil change should be required either the manufacturer or dealer should do it, because they know the best type of refrigerating oil for their particular product.

13. Driers or Cartridge Dehydrators

Driers or cartridge dehydrators are tubular appliances, containing moisture-absorbing material such as silica gel, activated alumina, anhydrous calcium sulfate, calcium oxide or anhydrous calcium chloride, in a sack or in a dispersion tube. They are manufactured with capped or flanged-end shells which can be inserted into the liquid-circulating line. Refrigerator manufacturers and service men insert them in commercial and sealed units to remove moisture from the system. After the liquid refrigerant and the oil circulate through the cartridge for a specified time the cartridge is disconnected.

14. Lubrication of Bearings

On a large heavy-duty refrigerator compressors a gear pump or motor-driven centrifugal pump forces oil to a header supplying oil to the bearings. Floor lubrication (splash system) is also used, as previously described.

Large industrial horizontal or vertical refrigerating compressors use an independent gravity pressure circulation or force feed lubricating system. In either of these systems the oil should go through a separator if it is contaminated with water.

Bearings are frequently lubricated by ring and chain oilers. A ring oiler consists of a machined ring having a larger diameter than the journal. The ring rotates with the journal, picking up oil from a reservoir and carrying it up to the bearing surface. This surface generally contains grooves for the distribution of the oil over the entire surface, after which the excess oil runs back into the reservoir. The level in the oil reservoir must be maintained sufficiently high so that the ring always dips well into the oil. Ring oilers are used for the lubrication of outboard crankshaft bearings of industrial reciprocating compressors and for rotor bearings of centrifugal compressors. Chain oilers operate on the same principle as ring oilers, except an endless chain is substituted for the ring.

Oil also acts as a shaftseal (where the shaft passes through the casing) in combination with the lapped metallic seals. This arrangement prevents air from entering the system and the loss of refrigerant to the atmosphere. In centrifugal compressors oil is supplied to the seal by pressure during operation and the metal seal faces are separated by oil pressure. When the compressor is not operating a metal-to-metal seal is made under oil.

Outboard bearings are also lubricated with sight-feed oilers consisting of a cylindrical cup filled with the oil which flows through a tube into a hole in the bearing. A glass is inserted in the oil tube so that

the flow of oil is visible and can be regulated by a needle valve inserted in the tube.

15. Air Conditioning Equipment

The lubrication of small **unit type** air conditioners is similar to that previously covered under domestic and small systems. The original charge of refrigerating oil and any necessary service should be taken care of by the manufacturer or dealer from whom the equipment was purchased. In railroad, trucking, bus and airplane applications, which place more severe service on these units, regular inspection and overhaul periods are customary.

In vertical reciprocating compressors of small and medium tonnage, splash lubrication is mostly used.

In large centralized systems (large theaters, hotels, department stores, hospitals, industrial plants, office buildings, etc.), pressure lubrication is used in the large type horizontal and vertical compressors for the lubrication of cylinders and piston rods, etc. In some designs an overflow pipe from the stuffing-box oil lantern carries the oil to the suction line and delivers it to the cylinder.

Force-feed lubricators are also used. Some compressors in this service contain an enclosed type of force feed pump located on the bottom of the crankcase and driven from the main shaft. Such a pump always has positive suction and delivers oil through feed lines to all moving parts under positive pressure.

The bearings of **motors and fans** used for the distribution of the cooled air are mostly of the antifriction type, and their lubrication is identical with such bearings in industrial use. The lubricant must protect the highly polished surfaces from pitting and corrosion, must seal the housing against foreign matter (dirt and water). and must support sliding and rolling contact within the housing. Oil or grease serve as the lubricant. The recommendations of the manufacturer of such bearings, which are based on the service, operating conditions, speed and temperature, should be followed.

Before air is conditioned for circulation, dust and dirt should be removed. Various types of **filters** and **cleaners** are used to clean the air thoroughly without offering too much resistance to the air flow. Some of these filters in large air conditioning systems employ a viscous **oil film** to hold dust and dirt. The coating which gradually accumulates renders the filters inefficient unless cleaned and reoiled. Self-cleaning automatic filters are on the market in which screen panels rotate continually through an oil reservoir at the base of the filter, and the dirt is washed off and accumulated in a sludge-pan. The oil in the reservoir is changed whenever it becomes too saturated with dirt.

References

1. Rutledge, *Refrig. Eng.*, no. 1, vol. 35, pages 31–36, Jan. 1938.
2. Private communication, Shell Oil Co.
3. Private communication, Frigidaire Division, General Motors Corporation.

18. MOISTURE AND DRYING METHODS

1. Moisture in refrigerating machines constitutes a most important problem for both the manufacturer and service man. Engineering has solved most of the mechanical problems in refrigeration, while producers have virtually eliminated difficulty due to refrigerants, unless they are mishandled. In the main, satisfactory lubricating oils have been provided, reducing trouble from this source to a minimum. However, moisture is still found in machines. Absence of moisture above an allowable minimum is absolutely essential to satisfactory machine operation. It is therefore imperative that moisture be eliminated during manufacture, that entrance of moisture in a system be guarded against in all field operations and that when it does get in, removal be accomplished as speedily as possible. Moisture may be present in a system as the result of:

1. Faulty drying of equipment at factory and in service operations.
2. Introduction during assembly or service operations in the field.
3. Low-side leaks, resulting in entrance of moisture-laden air.
4. Leakage of water condenser.
5. Oxidation of certain hydrocarbons of oil, to produce moisture.
6. Wet oil and/or refrigerant.
7. Decomposition of motor insulation in hermetically sealed units.

Item 1 will be discussed below, and items 2 to 5 may be corrected by proper servicing. Moisture is present in such small amounts in refrigerants and oils which are bought to specification and properly handled thereafter, that virtually no difficulty arises from these sources.

Effects of Moisture

2. The presence of moisture in a refrigerating system may result in one or all of the following undesirable **effects:**

1. "Freezing up" at expansion valves or capillary tubes, ice in evaporators.
2. Corrosion of metals to form sludge.
3. Flapper-valve failure.

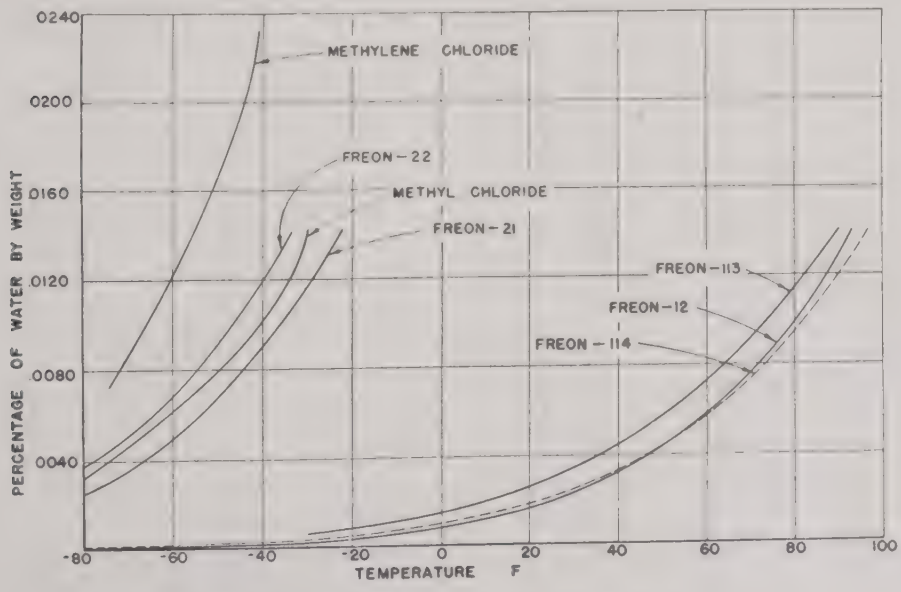


Fig. 1. Solubility of Water in Refrigerants

4. Copper plating.
5. Chemical damage to insulation, etc.

Ice separates from the Freon refrigerants, methyl chloride, methylene chloride, butane, and isobutane, whenever the amount of moisture is sufficiently great and the temperature of the refrigerant low enough. The latter accounts for the formation of ice in expansion valves, capillary tubes and evaporators, but not elsewhere in the machine.

In low-temperature units a frozen expansion valve or capillary tube may result due to the separation of **wax** from oil-methyl chloride or oil-Freon mixtures.[1] The presence of wax may be demonstrated by **washing** the expansion valve with a small amount of carbon tetrachloride, and evaporating the solution so obtained. White wax or an extremely viscous oil-wax mixture proves that wax is the offending element although it does not rule out moisture. The wax separation characteristics of the oil should be determined,[1] and if the oil is not satisfactory another should be used.

The separation of **water,** as ice, is related to the solubility of water in a refrigerant. The solubility of water (ice) varies with the different refrigerants mentioned above, and is great enough in ammonia, carbon dioxide, and sulfur dioxide so that freeze-ups do not occur. The greater the solubility of water in a refrigerant, the less likely that ice will separate. **Fig. 1**[2a] contains curves representing the solubilities of water in Freon-11, Freon-12, Freon-21, Freon-22, Freon-113, Freon-114, methyl chloride and methylene chloride. No data are available for butane and isobutane.

3. The **corrosion** of metals occurs whenever the quantity of water exceeds certain fairly well defined values.[3] **Table 1** contains the results of tests conducted on steel in glass tubes where the quantity of water could be clearly defined.

Limits for copper, brass and aluminum are slightly higher in all three refrigerants. **Aluminum** in methyl chloride is eliminated from moisture corrosion consideration, since a direct action occurs resulting in the formation of spontaneously flammable products.[4] Higher temperatures, encountered in the compressor and condenser, will lower these limits appreciably, so that corrosion occurs with less moisture in a machine at these points than with steel at room temperature.

The **mechanism** of corrosion, which results in the formation of metallic salts, (sludges) has been studied.[5] Water reacts with sulfur dioxide, methyl chloride and Freon-12 (hydrolysis[5,6]) to form sulfurous, hydrochloric, and hydrofluoric acids, respectively. These acids react with iron, copper and aluminum (see also direct action aluminum on methyl chloride) to form metallic salts (sludges) of well defined composition. By chemical analysis of these salts it is possible to demonstrate their origin as being due to water. Chemical analysis thus becomes important.

Table 1

Refrigerant	Water by weight, %	Results
Sulfur dioxide	0.03	Slight discoloration
	0.10	Slight scale
	0.15	Heavy scale
		Presence of air did not affect results
Methyl chloride	0.02	Slight discoloration
	0.03	Marked discoloration
		Very slight scale
	0.05	Moderate to heavy scale
		Presence of air increased corrosion in all cases
Freon-12		Similar to methyl chloride

Approximately 90% of the sludges produced in refrigerating systems are due to moisture;[5] the others are associated with oil, or minor causes.

Corrosion in a sulfur dioxide system proceeds within a few days, or even hours, provided enough water is present, whereas it is much slower for methyl chloride and Freon-12. This is due to the very rapid action of water on sulfur dioxide and the rather slow action of water on the other two refrigerants, to produce the acids responsible for corrosion.

In the presence of air, corrosion is much worse in all refrigerants except sulfur dioxide. Corrosion in a butane or isobutane

system is due to the direct action of water and perhaps air, on metals.

Waring[32] has shown that a relatively large quantity of moisture in a reciprocating system using Freon-12 as refrigerant, results in accelerated flapper-valve breakage. The same system, overhauled completely and equipped with an efficient drier, operated indefinitely without flapper-valve breakage. Incipient corrosion, probably of an intercrystalline nature, was advanced as the cause of fatigue failure of the flapper-valves.

Copper plating may be the result of the action of, or related to, moisture,[7] but a proper relationship seems difficult to establish.

Consideration of the foregoing shows that the amount of water present in a refrigerating system must be small enough to avoid ice separation and corrosion. With sulfur dioxide, corrosion alone must be avoided, which probably limits the maximum water content to below 0.03%.

Some limited information[8] regarding the water content of machines that are operating satisfactorily is given in **Table 2.**

These data indicate that the moisture tolerance of Freon-12 is lower than for methyl chloride. This is quite generally borne out by experience in manufacturing and servicing. Since the quantity of moisture required to produce corrosion is well above that causing a freeze-up, elimination of freeze-ups should exclude corrosion of the type due to water.

Table 2. Moisture Content of Machines Considered Sufficiently Dry for Normal Operation

Company	Refrigerant	Water by weight, %	Type of system
A.			
	Freon-12	.0027	Large commercial
	Freon-12	.0026	Large commercial
B.			
	Freon-12	.0037	Domestic hermetic
	Freon-12	.0037	Domestic hermetic
	Freon-12	.006	Domestic unit returned due to moisture
C.			
	CH_3Cl	.0058	Small commercial
	CH_3Cl	.0058	Small commercial
D.			
	CH_3Cl	.007	Small commercial
	CH_3Cl	.005	Small commercial
E.			
	Freon-12	.002	Hermetic
	Freon-12	.003	Hermetic
	Freon-12	.003	Small commercial
	Freon-12	.002	Hermetic
	Freon-12	.002	Small commercia
	Freon-12	.003	Hermetic

Drying Methods

4. There are three general requisites of a satisfactory process of drying refrigerating equipment, namely, heat, time and a method for the removal of moisture.

a. **Heat,** which is necessary for the evaporation of water, may be applied by placing the equipment in an oven, using hot air or heating the apparatus with a flame, wherever possible.

b. **Time** is required for the evaporation of water, the elimination of water adsorbed on the various surfaces and the removal of the water from the equipment with dry air or by means of a vacuum.

c. In general, higher temperatures cut down, but cannot reduce the time requirement below a minimum which must be set for each process, type of equipment and the degree of drying desired. A **current of dry air** drawn or blown through the equipment removes moisture as it becomes totally or partially saturated. The rate of flow should be high enough to avoid complete saturation of the air in order to minimize the danger of deposition of moisture in some cooler portion of the equipment. On the other hand, there is little to be gained by the use of a very high rate of air flow, since the amount of water removed per unit of air will be small. Since hermetic windings can be damaged by too high temperatures or too long a time in the dehydration oven, caution should be exercised in setting up a drying procedure.

Dry air may be **manufactured** by several methods, among which are:

a. Removing the major part of the moisture with a refrigerating unit, followed by more complete drying in a unit charged with activated alumina, silica gel, or Drierite.

b. The use of a drier unit, charged with activated alumina, silica gel, or Drierite, and equipped with heating units (electrical or steam) for the regeneration of the drier. Normally, such units are provided in pairs, so that continuous operation may be assured. Fenwick[9] describes a unit which, along with others, is commercially available.

Under some circumstances in the field, where the equipment cannot be heated, anhydrous methanol (methyl alcohol) has been used to flush out moisture. The alcohol, having a lower boiling point, may be removed more readily than water with a stream of dry air, carbon dioxide or oil-pumped nitrogen, or with the application of a vacuum. Under no circumstances should methanol in quantity be allowed to remain in the system, since it will produce corrosion.[10] Methanol, in quantities somewhat less than 1%, does not induce rapid corrosion and moreover exerts an antifreeze action.

Vacuum

5. Air is evacuated from equipment by using mechanical **vacuum pumps** or, in some instances, water suction pumps.[11] The latter are not in general desirable since fluctuation of water pressure may cause them to suck water back into the equipment being evacuated. Moreover, the vacuum obtained never exceeds the vapor pressure of the water used to operate the pump.

The boiling point of water is lower in a vacuum and consequently a lower temperature is required to evaporate moisture than at higher pressures. Water boils at approximately 104 F, under 28 in. of vacuum, and at 59 F, under 29.5 in. In general, temperatures much higher than these are employed with the vacuum method of drying.

Whether dry air or a vacuum is employed seems to make little difference, provided proper equipment is used for producing dry air or the desired vacuum, and the time and temperature requirements are met. Goddard[30] has shown that a vacuum treatment, followed by breaking the vacuum with dry air, when the vacuum period is concluded, gives additional drying, since the water remaining as a gas is removed by flushing with air. Carbon dioxide, oil-pumped nitrogen or gaseous refrigerant may be employed in lieu of air. Anderson[12] has shown one very desirable feature associated with the use of dry air, namely: when air is drawn through equipment restrictions and plug-ups are revealed; when a vacuum is employed neither is indicated. Blair and Calhoun[31] have pointed out a fallacy in the assumption that a low-pressure reading of an evacuated system insures the removal of all moisture in excess of that corresponding to this pressure. This is due to failure to reach true equilibrium conditions.

Little has been published concerning the **drying procedures** used by the major refrigerating machine manufacturers or employed by service engineers in the field. A few articles indicate the time, temperature, and moisture removal methods of a few procedures. The Westinghouse process for domestic units is described by Anderson[12] as involving a series of nine units, an oven temperature of 257 F, the use of air dried (not exceeding dew point of −58 F) with calcium chloride and potassium hydroxide and admitted under controlled low pressure, and a time (not revealed) sufficient for complete drying. A moisture test is made on the ninth unit.

Wile[13] describes, in very general terms, the drying method used by Savage Arms Corporation, for ice cream cabinet units. Better than ordinary drying, necessitated by the low temperatures involved, is attained by "dehydrating each piece of equipment . . . at high temperature and a very fine vacuum." This procedure apparently results in systems sufficiently dry, but silica gel drier units are placed on each machine as an additional precaution.

A general article[6] gives a drying procedure for sulfur dioxide machines, which includes heating in an oven at 250–275 F under vacuum (greater than 29.5 in.) for 4 hr. This is claimed to reduce the moisture content of the machine to approximately 0.0095 oz of water per cu ft of internal machine volume. As a means of removing the water vapor remaining in the machine, flushing with dry air is advocated. A discussion of drying by vacuum treatment at room temperature and flush-

ing with dry air is also included in this article.

A number of articles have been written covering drying procedures employed by service men. Newcum[11] describes a method and apparatus for cleaning and dehydrating equipment in the field which involves drawing a vacuum on the system while heating it progressively toward the outlet connected to the vacuum pump. When this process is finished, dry hot air is admitted through a drier unit for some time. Properly carried out, this procedure should result in dry equipment. Fenwick[9] describes a system and apparatus for producing dry air and for drying units in the field as well as in the factory. Details of operation are given. A dehydrating unit which can be built in any well equipped shop is discussed by Dinsmoor.[14]

Control of Drying Operations

6. Proper control of drying operations is necessary if moisture difficulties are to be avoided. How may the manufacturer be certain that all equipment is dry and what routine checks may be used to catch "wet" machines before they reach the customer? Routine checking of the finished machine varies with the type of drying process employed. Where a stream of air is used, a dew-point determination of the outgoing air has been employed. Fenwick[9] advocates the use of a dew point of −40 F (equal to 0.046 gr water per cu ft) for the air leaving the equipment, as proof that it is dry.

Anderson[12] determines the residual moisture in the last machine of a series, which has been dried with air, by drawing off the moisture into a liquid air trap and from this calculating the quantity in the machine.

McGovern[6] recommends drawing a vacuum on the equipment being tested, closing off the system and observing the pressure rise, if any. If no pressure rise occurs after a time, the machine may be considered dry, since moisture remaining in the machine will create its own gas pressure.

Moisture may also be determined by removing a portion of the liquid refrigerant from a machine which has been operated for several hours and until the moisture is in equilibrium throughout. The quantity of moisture in the removed sample is determined by the phosphorous pentoxide method.[15] Recently Walker and Rinelli[16] developed a method applicable to methyl chloride, methylene chloride and the Freon refrigerants which makes possible routine analysis of a very large number of machines.

A standard[17] for refrigerant system tubing and cooling coil has been set at a maximum permissible moisture content of 10.6 milligrams per liter (300 milligrams per cu ft) of internal volume. A method for determining the moisture content was outlined.

Driers

7. After a machine has been installed, overhauled, recharged, etc., moisture is a very frequent cause of trouble. In the absence of other difficulties, a machine may be put into satisfactory operating condition by the removal of moisture with a drier. A drier is a chemical compound capable of adsorbing, or reacting chemically with the moisture contained in the liquid or gaseous refrigerant-oil mixture. The drier is placed in a suitable unit equipped with screens and filter pads to prevent the solid drier from entering the refrigerating system. The drier unit, in addition to holding the drier, functions as a filter and removes any solid found in the liquid or gaseous refrigerant reaching it.

There are several substances which have been used as driers in the chemical laboratory. Information concerning these driers has been published in a series of papers.[18,19,20] The data from these papers are not applicable directly to the problem of drying a refrigerant and consequently more is to be gained from several general articles.[21,22,23,6,24,3]

In order that a chemical substance be acceptable for use as a drier, it must possess, in addition to the property of water removal, characteristics which eliminate any possibility of undesirable reactions with the refrigerant, oil or machine parts.[6,24] Activated alumina, silica gel, Drierite, calcium oxide and calcium chloride have been most widely used, the first three being rated highly acceptable.

Table 3 contains the results of tests[3] and shows the relative efficiency of several driers.

Table 3. Drying Power of Various Materials

Drier	Refrigerant	Liquid or vapor	Maximum residual moisture,* % (Initial water concentration) .25%	.02%
Activated alumina	Sulfur dioxide	L	.15	.005
		V	.01	
	Methyl chloride	L	.02	.006
		V	.01	
Silica gel	Sulfur dioxide	L	.15	.006
		V	.01	
	Methyl chloride	L	.01	.004
		V	.01	
Drierite (Calcium sulfate)	Sulfur dioxide	L	.15	.009
		V	.08	
	Methyl chloride	L	.05	.005
		V	.04	
Calcium chloride $CaCl_2$	Sulfur dioxide	L	.09	.013
		V	.03	
	Methyl chloride	L	.10	.005
		V	.04	
Calcium oxide CaO	Sulfur dioxide	L	.20	—
		V	.15	
	Methyl chloride	L	.15	—
		V	.08	
Barium oxide BaO	Sulfur dioxide	L	.20	.017
		V	.15	
	Methyl chloride	L	.05	.006
		V	.05	
Zinc	Sulfur dioxide	L	.25	—
		V	.25	

* Excluding samples which were of definitely poor grade.

A brief discussion of the various driers which have been and/or are being used in the refrigeration industry is given below.

Activated alumina[25] is a granular aluminum oxide which removes moisture by adsorption. It also adsorbs acids.[24] It dusts slightly on handling but this is not a source of trouble. It has had extensive and satisfactory use with sulfur dioxide, methylene chloride, methyl chloride, Freon-11 and Freon-12, is used with sulfur dioxide in vapor phase (suction line) only, with other refrigerants in either suction or liquid lines, usually the latter. It may be left on the machine indefinitely.

Silica gel[26] is a glass-like silicon dioxide gel which removes moisture by adsorption; does not dust. (See additional remarks under activated alumina.) It also adsorbs acids.[27]

Drierite[28] is anhydrous calcium sulfate prepared as a granular white solid. It removes moisture by chemical action. It dusts somewhat more than activated alumina but this does not cause trouble. Drierite is also cast in sticks. (See additional remarks under activated alumina.)

Calcium oxide removes water and acid by chemical action. It dusts somewhat. It cannot be used with sulfur dioxide but is satisfactory for other refrigerants. It rates as a fair drier with tendency to powder when excess moisture is present.

Calcium chloride removes water by chemical action and is not rated as a drier capable of reducing the moisture content to a low level. It may be used with all refrigerants. Excess moisture produces a solution of calcium chloride which is highly corrosive. It will not remove acid.

Unlike activated alumina, silica gel, Drierite and calcium oxide, it should not be left on a machine longer than a few days, and calcium chloride should be used with caution.

Barium oxide removes moisture by chemical action but should be used with care since it has been known to cause explosions.[24] It powders considerably after reaction with water.

Magnesium perchlorate, barium perchlorate. These are powerful oxidizing agents and have caused serious explosions with oil and methyl chloride.

Phosphorus pentoxide is an excellent drier, but its fine powdery form makes it difficult to handle and produces extensive resistance to flow of gas and/or liquid. A mixture with quartz[29] has been accorded limited use.

Soda-lime is a mixture of calcium oxide and sodium hydroxide. Its use has been limited and it is not recommended as a satisfactory drier.

Factors in the Use of a Drier[3]

8. Since a drier is extremely sensitive to moisture, it must be protected from it at all times until ready for use. Handling a drier in moist air must be avoided, and for this reason it is recommended that the drier be obtained in a factory-packed unit, or that it be prepared for use in the unit under conditions which will preclude contamination by moisture.

The position of a drier unit on a machine is important. A liquid feed line should enter the drier unit at the bottom with the exhaust at the top. This arrangement will insure the most uniform contact between refrigerant and drier. If a drier is placed vertically in the suction line, the feed should be at the top with the exhaust at the bottom in order that the oil may be blown down through and out of the drier unit.

A very important factor bearing on the whole problem of the use of a drier is the question of rate of moisture removal and the time a drier unit must be left on the machine. A critical glance at the problem will show that the time is longer than is often assumed. To take a specific example, suppose that three drops of water get into an otherwise dry system containing 3 lb of refrigerant. These three drops of water would amount to about .02% moisture. Since this amount of water in methyl chloride is sufficient to cause discoloration of steel and may freeze out at temperatures slightly below zero, it would appear advisable to dry the system down at least to .01% moisture. Suppose now that ½ lb of methyl chloride circulates per hr, which would represent a typical normal load in a household machine. Then in the first hour not more than one-sixth of the moisture would be removed, even if the drier did a complete job of moisture removal. Furthermore, since the returning drier refrigerant would dilute the wet refrigerant, the feed liquid would become progressively drier and less and less water could be removed by the drier. Hence, under even these ideal conditions, considerably more than 6 hr would be required. In addition, moisture will tend to collect in the evaporator of a flooded system, because even in relatively small concentrations it vaporizes much less readily than the refrigerant.

Also, in either a flooded or dry type system, it is highly probable that at least part of the water will be dissolved or entrained in the oil entering the compressor and will take a long time to be carried over into the refrigerant cycle where it can be removed.

As a further consideration, it must be recognized that no drier will reduce the moisture content to zero even under ideal conditions, while under practical refrigeration conditions the efficiency of the drier is apt to be considerably less than ideal.

Acrolein is required in some cities as a warning agent for use with methyl chloride. Tests[8] show that activated alumina, silica gel and barium oxide reduce the acrolein content of methyl chloride approximately 80, 40 and 50%, respectively, over a period of 70 days. Calcium chloride, Drierite and calcium oxide do not appreciably reduce the acrolein content in the same time period.

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19. LOAD CALCULATIONS

I. Source of Heat

THE calculation of the loads on refrigerators and refrigerated rooms is based on well-established procedures. Regardless of the size of the refrigerator, there are four general sources of heat:

1. Wall losses. The heat leakage through the walls, floor, and ceiling of a refrigerator is a function of the thickness and type of insulation, the thickness and construction of the walls, the temperature difference between the outside air and the refrigerator air, and the outside surface area of the refrigerator. Mathematically, this may be expressed as

$$Q = AU\Delta T$$

where Q = heat leakage in Btu/hr
A = outside area in sq ft
U = overall conductivity, Btu per hr sq ft deg F
ΔT = difference between design outside temperature and average refrigerator temperature, deg F

2. Air changes, whether controlled as in providing ventilation air or uncontrolled, introduce an additional load. The warm air entering the refrigerator when the door is opened must be cooled to the temperature of the refrigerated space. In addition, there may be air infiltration through joints or cracks in the framing or insulation of the enclosure. The refrigeration requirement to cool this incoming air is dependent on the outside and inside temperatures and humidities and consists of (a) the sensible heat removed while cooling the air, and (b) the latent heat of vaporization of moisture that is condensed from the air when it is cooled.

3. Product load. A product placed in a refrigerator at a temperature higher than the storage temperature will lose heat until it reaches the storage temperature. The quantity of heat to be removed may be calculated from a knowledge of the product, including its state upon entering the refrigerator and final state, its weight, specific heat before and after freezing and its latent heat. In the case of fresh fruits and vegetables, there may also be the heat generated within the product itself due to respiration, known as the heat of evolution.

4. Miscellaneous sources of heat include the heat added through lighting, motors operating, people working within the refrigerated space, etc.

In order to make a complete calculation and obtain the total load, each of the four sources must be treated separately and the total determined. From experience over a period of years, various shortcut methods have been evolved which eliminate the need for some of the detailed calculations indicated above. These methods may be used successfully provided the user is acquainted with their limitations. In the sections following, load calculations are discussed, first with respect to commercial refrigerators and coolers up to 1,500 cu ft in volume, and second with regard to larger storage coolers for which it is necessary to consider separately each of the four components of the total load.

After the total refrigeration load has been determined, the condensing unit and evaporator are selected having a capacity which will meet the load requirements with an operating time less than 24 hr per day. This procedure is necessary for several reasons. With finned type evaporators, either natural convection or forced circulation, the condensing unit must stop periodically to permit defrosting of the evaporator. In refrigerators held at 34 F and higher, defrosting is accomplished by air circulation, and in refrigerators held at lower temperatures by some other means, such as water or brine defrost, hot gas defrost, or electric heat. The additional refrigeration capacity permits these periodic shut-downs without losing control of

Table 1. Load Factors for Refrigerators and Coolers Having Gross Interior Volumes Not in Excess of 1,500 cu ft

(Btu/hr F sq ft of surface area)

Insulating material	Thickness of		Heat leakage only	Total load factor		
	Insulation, in.	Wall, in.		Light usage	Average usage	Heavy usage
Corkboard, balsam wool, celotex, ground cork, insulite, kapok, rock cork, linofelt, mineral wool, palco bark, thermocraft, etc.	2	4	.125	.153	.167	.181
	2½	4½	.112	.140	.154	.168
	3	5	.100	.128	.142	.156
	3½	5½	.091	.119	.133	.147
	4	6	.083	.111	.125	.139
	4½	6½	.077	.105	.119	.133
	5	7	.072	.100	.114	.128
	5½	7½	.067	.095	.109	.123
	6	8	.063	.091	.105	.119
	7	9	.051	.083	.094	.105
	8	10	.046	.078	.089	.099
	10	12	.038	.069	.080	.090
Sawdust, shavings, lithboard	2	4	.165	.193	.207	.221
	2½	4½	.148	.176	.190	.204
	3	5	.132	.160	.174	.188
	3½	5½	.120	.148	.162	.176
	4	6	.110	.138	.152	.166
	4½	6½	.102	.130	.144	.158
	5	7	.095	.123	.137	.151
	5½	7½	.088	.116	.130	.144
	6	8	.083	.111	.125	.139
Wood, cottonseed hulls, rice chaff, strawboard	2	4	.205	.233	.247	.261
	2½	4½	.184	.212	.226	.240
	3	5	.164	.192	.206	.220
	3½	5½	.149	.177	.191	.205
	4	6	.136	.164	.178	.192
	4½	6½	.126	.155	.168	.182
	5	7	.118	.146	.160	.174
	5½	7½	.110	.138	.152	.166
	6	8	.103	.131	.145	.159

Note: These values are for a layer of wood on each side of the insulation. For walls without wood sheathing, use factor from table for ½ in. less insulation. The wood framing of cabinets has been taken into consideration in determining these factors.

the room temperatures which must be maintained.

Refrigeration loads are usually figured on an average of maximum conditions over a 24-hr period rather than on peak conditions. The capacity reserve obtained by sizing the refrigeration system on less than 24-hr operation gives a safety factor which is utilized when design conditions are exceeded due to increase in ambient temperature, increase in usage, etc.

It is usual practice in the commercial refrigeration field to base the equipment selection on approximately 16 hr operating time for systems maintaining room temperatures of 34 F and higher, and 18 to 20 hr operating time for systems maintaining lower room temperatures. The required condensing unit and evaporator capacity is obtained by dividing the 24-hr load by the desired running time.

II. Commercial Cabinets and Coolers (up to 1,500 cu ft volume)

Unless there are unusual conditions of usage or application, the **load calculations** for commercial cabinets and coolers may be based on empirical methods without resorting to detailed calculations. The results thus obtained are within a degree of accuracy adequate for all general purposes. Should unusual conditions exist, the load may be calculated by using the same methods outlined later for larger storage spaces.

Commercial cabinets and coolers considered herein are: (1) the service cabinet or reach-in refrigerator; and (2) the service cooler storage room or walk-in cooler of either the prefabricated or built-on-the-location type.

The various refrigeration loads in the fixtures herein treated are the same as those enumerated above, but in the interests of simplification they may be regrouped as follows:

1. Wall losses. The heat leakage through the walls, floors and ceiling are functions of the outside surface area of the refrigerator, the temperature difference between room air and refrigerator air, and the over-all conductivity.

2. Usage load. Rather than break the other loads into their respective elements, it is common practice to group all other loads into one classification known as the usage load. This procedure eliminates the necessity for determining separately the product load, air change load, and miscellaneous loads which, with a wide variety of products in relatively small quantities and frequent door openings, could become a laborious chore, particularly if repeated many times for similar cabinets or coolers. The usage load includes:

a. Food placed in the refrigerator or cooler
b. Warm receptacles placed in the refrigerator or cooler
c. Infiltration of warm air, including moisture content due to poor openings and leakage
d. Incidental occupancy and lighting.

Types of service are classified as light, average, and heavy. **Light usage** includes installations where the door openings are infrequent, the room temperature is not excessive, the quantity of material to be cooled is not great, and where it is not necessary to maintain a refrigerator temperature below 45 F. A florist's refrigerator is an example of light usage.

Average usage includes installations not subject to extreme temperatures and where the quantity of food handled in the refrigerator is not abnormal. Refrigerators in delicatessens and clubs may generally be classified under this type of usage.

Heavy usage includes installations such as those in busy markets, restaurant and hotel kitchens where the room temperatures are likely to be high, where rush periods place heavy loads on the refrigerator, and where large quantities of warm foods are often placed in it.

3. The **heat leakage** and usage load are combined into "load factors" for the particular type of insulation and service for which the refrigerator is used. **Table 1** gives the overall conductivity for various types of insulation and the total load factor for the three types of usage. The conductivity factors, based on walls having wood framing or studding and wood sheathing on both sides, are applicable to refrigerators having gross interior volumes less than 1,500 cu ft. Load factors for glass (U_g) are given in **Table 2.** The load factors do not include any allowance for loads such as ice freezing or food freezing.

The total load for a refrigerator or cooler exclusive of any ice or food freezing, is

$$Q = (A - A_g)(T_1 - T_2)U + A_g(T_1 - T_2)U$$

where

Q = heat load (Btu/hr)

Table 2. Load Factors for Glass Areas in Refrigerators and Coolers

(Btu/hr F sq ft surface area)

Number of panes	Heat leakage only	Total load factor		
		Light usage	Average usage	Heavy usage
1	1.13	1.16	1.17	1.19
2	0.46	0.49	0.50	0.52
3	0.29	0.32	0.33	0.35
4	0.21	0.24	0.25	0.27

Note: In estimating the leakage through glass surfaces, use the area of the exposed glass only.

A = total outside surface area of refrigerator, sq ft
A_g = exposed area of glass, sq ft
T_1 = maximum average room temperature, deg F
T_2 = average refrigerator temperature, deg F
U = load factor (Btu/hr F sq ft from Table 1)
U_g = load factor (Btu/hr F sq ft from Table 2)

As an example it is required to calculate the load on 8-ft × 10-ft × 10-ft walk-in cooler with three double-pane glass service doors having an exposed glass area of 4 sq ft each. Cooler has 4 in. of ground cork insulation with wood sheathing on both sides. Service is average usage. The maximum average room temperature is 100 F and the average cooler temperature is 40 F.

Total outside area $A = 520$ sq ft
Exposed area of glass $A_g = 3 \times 4 = 12$ sq ft
Load factor $U = 0.125$ Btu/hr F sq ft (from Table 1)
Load factor $U_g = 0.50$ Btu/hr F sq ft (from Table 2)
$Q = (520 - 12)(100° - 40°)\ 0.125 + 12(100° - 40°).50 = 4170$ Btu/hr
Condensing unit and evaporator capacity required on a 16-hr operating time basis is $4,170 \times 24/16 = 6,255$ Btu/hr.

4. Frequently it is desired to **make ice** in a service cabinet or cooler. This load must be figured in addition to the usage loads figured above. The latent heat of fusion of water is 144 Btu/lb. However, the water is usually at some temperature above 32 F, from which it must be cooled to 32 F. After it is frozen, some additional refrigeration is needed to cool the ice to the evaporator temperature, usually about 20 F. As the specific heat of ice is 0.5 Btu/lb, this is 6 Btu/lb. If the initial temperature of the water is 72 F, a total of 190 Btu/lb heat removal would be required to convert that water to ice at 20 F, as follows:

Sensible heat removal (72 − 32)1.0	= 40 Btu/lb
Latent heat removal	= 144 Btu/lb
Sensible heat removal (32 − 20) 0.5	= 6 Btu/lb
Total	= 190 Btu/lb

If, in the example given above, 75 lb of ice are to be frozen twice a day, the calculation would be made as follows (assuming the two freezings are spread over 24 hr)

$75 \text{ lb} \times 2 = 150$ lb ice required for 24 hr

$$\frac{150 \times 190}{24} = 1{,}188 \text{ Btu/hr}$$

The condensing unit and evaporator capacity on a 16-hr operating time basis then becomes $(4{,}170 + 1{,}188) \times 24/16 = 8{,}037$ Btu/hr.

It is sometimes required that the quantity of ice be frozen in a specified period of time; then the condensing unit capacity (and the evaporator capacity) must be adequate to meet this requirement.

Suppose in the foregoing example 100 lb of ice are to be frozen in 8 hr. The percentage of running time may be figured on the basis of the total load per 24-hr day, as was done above. A further check should be made to determine if the heat leakage load, plus the usage load plus the ice freezing load over the period during which the ice is being frozen, does not exceed the capacity of the condensing unit. Using the example of the walk-in cooler above:

Heat leakage and usage load per 24 hr, $4,170 \times 24$	= 100,080 Btu
Ice freezing, 100 lb in 8 hr, 100 lb × 190 Btu/lb	= 19,000 Btu
Total load, Btu/24 hr	= 119,080 Btu

Condensing unit capacity required on 16-hr operating time basis is

$$\frac{119{,}080}{16} = 7{,}443 \text{ Btu/hr}$$

In order to determine that a condensing unit having the foregoing capacity has sufficient capacity to carry the heat leakage and usage loads and to freeze 100 lb of ice during the 8-hr period, the following check calculation is made:

Heat leakage and usage loads	4,170 Btu/hr
Ice freezing load (per hr during 8-hr period) 19,000 ÷ 8 hr	2,377 Btu/hr
Total refrigeration required per hr during 8-hr ice freezing period	6,547 Btu/hr

Since this total load is less than the condensing unit capacity as figured for a 16-hr operating time per day, the condensing unit will handle the ice freezing load during the specified time.

Other types of **additional loads,** such as

food freezing or bottled beverage cooling, which are not included in the load factors given above, may be treated in the same manner as in the preceding ice freezing example.

The freezing of food is a heat transfer problem which involves both the amount of heat to be removed and the rate of removal. Refer to the Applications Volume of the ASRE Data Book (Sec. I) for a discussion of food freezing.

Although it is recognized that interior volume, and hence usage load, may vary somewhat for a given number of square feet of exterior surface, the ease with which loads for smaller refrigerators and coolers may be estimated by the method based on square feet of exterior surface makes it a preferred practice. An understanding of the limitations of this short method can be gained through experience. A comparison of results between the short method and the detailed method for several typical applications will be helpful in gaining this experience.

III. Cold Storage Rooms (larger than 1,500 cu ft volume)

In determining the refrigeration requirements of larger storage spaces it is generally necessary to give more detailed consideration to each of the load components than for refrigerators and the smaller walk-in coolers, since any one of these components may be an appreciable portion of the total load. As stated before, the methods outlined below may also be applied to the smaller coolers if desired.

1. The wall losses of the cooler are calculated by establishing the coefficient of transmittance, U, for each of the wall sections and the floor and ceiling, taking into account the several materials used. This is done through use of the following equation:

$$U = \frac{1}{1/f_i + L/k + 1/f_o}$$

where

U = coefficient of transmittance, Btu/hr F sq ft
L = thickness of the wall, in.
k = conductivity of material in wall, Btu per hr F sq ft per in.
f_i = inside film or surface conductance, Btu/hr F sq ft
f_o = outside film or surface conductance, Btu/hr F sq ft
f_i and f_o are frequently used as 1.65 and 6 respectively.

With thick walls and low conductivity, the resistance L/k makes U so small that $1/f_i$ and $1/f_o$ have little effect. As walls are seldom made of one material, the value L/k represents the composite value of the several materials used in series to the heat flow, and for a wall with flat parallel surfaces of materials 1, 2, 3, etc., is

$$U = \frac{1}{L_1/k_1 + L_2/k_2 + L_3/k_3 +}$$

where L_1, L_2, L_3, etc. are the thicknesses and k_1, k_2, k_3, etc. are the conductivities of the several materials used.

Having established the coefficient of transmittance (U), the heat leakage through the entire area or section of the wall, ceiling or floor, is given by the basic equation

$Q = A U \Delta t$
Q = heat leakage, Btu/hr
A = outside area of section, sq ft
U = coefficient of transmittance, Btu/hr F sq ft for section
Δt = difference between average outside temperature and average refrigerator temperature, deg F

Each section of the cooler in question should be considered separately, in order to determine whether there are any differences in construction which would change the value of U, or whether the location of the section, such as one wall having sun exposure or being a common wall with another refrigerated room, would necessitate a different outside temperature being used.

Example: Calculate the heat leakage through a 100-sq ft section of a wall consisting of 4 in. of corkboard applied in two layers with hot asphalt against a 12-in. brick wall faced with ½-in. mortar and finished with ½ in. of cement. The outside temperature is 100 F and the temperature within the refrigerated space is 40 F.

From Chapter 12, Table 4, the values of k are

Brick, low density	5.0
Cement mortar	12.0
Corkboard	0.27
Cement plaster	12.0

Table 3. Approximate Wall Heat Losses
(Btu/hr sq ft F)

Insulation	Btu/hr sq ft F
Cork or equivalent inches	
3	0.10
4	0.075
5	0.060
6	0.050
7	0.043
8	0.0375
9	0.0333
10	0.0300
11	0.0273
12	0.0250
Glass	
No. of panes	
1	1.13
2	0.46
3	0.29
4	0.21

then

$$U = \frac{1}{12/5 + 0.5/12 + 4/0.27 + 0.5/12}$$

$$= \frac{1}{2.4 + .0417 + 14.81 + .0417}$$

$$= .0575 \text{ Btu/hr F sq ft}$$

$$Q = 100 \times .0575 \times (100 - 40)$$

$$= 345 \text{ Btu/hr}$$

Heat transfer factors for cork or equivalent are given in **Table 3.** The thickness of cork insulation specified is the actual cork or equivalent insulation thickness, not the overall wall thickness. Judgment must be exercised in using this or similar tables, for many commercial fixtures have fill insulation with considerable wood studding and the values given may be low when applied to such construction.

Where the refrigerated space is located remotely from artificial sources of heat that will affect the outside design temperature, and where more specific information is not available, the design temperatures given in **Table 4** may be used. The average ambient temperature, representing the average of the hottest 24 hours over a five-year period, in the absence of better information may be used as the outside temperature on which to base load calculations.

It is to be noted that these design temperatures are somewhat lower than those used for air conditioning design. The use of different design temperatures for refrigeration and air conditioning applications is not incongruous since the procedures for applying the equipment are somewhat different. Equipment for refrigeration applications is usually selected to carry the 24-hr load with less than 24-hr operating time, whereas equipment for air conditioning applications is selected to carry the load at design conditions with 100% running time. In refrigeration applications the peaks are handled by the reserve capacity, while in air conditioning applications the indoor temperatures are allowed to increase when outdoor design conditions are exceeded.

The maximum ambient temperature given in Table 4 is the average of the highest temperature of the hottest day of five consecutive years. These temperatures are particularly useful in selecting an air-cooled condensing unit, as such a unit must be capable of operating at the maximum design ambient temperature without exceeding its motor operating limit. The temperatures given must not be used for air conditioning designs.

The walls or roof of a storage cooler exposed directly to the **sun's rays** will absorb additional heat. The amount of heat thus absorbed depends on:

(a) The **nature** and **type of surface,** a dark surface absorbing more heat per unit area than a light-colored surface.

(b) The **intensity** of the **solar radiation,** which varies for different locations of the country with the seasons of the year, depending on the angle which the sun's rays make with the surface affected. The more nearly the rays strike perpendicularly to the surface, the more heat is absorbed. The calculation of the effect of solar radiation is detailed and involved. For practical purposes it is satisfactory to adjust the normal temperature difference used in making the load calculations to compensate for the effect of the sun's rays.

The number of degrees to add to the normal temperature difference to compensate for the effect of the sun's rays is shown in **Table 5.** The heat leakage through a roof

Table 4. Refrigeration Design Ambient Temperature Guide*

Location	Average ambient temp	Maximum ambient temp
Alabama		
Birmingham	88	99
Mobile	88	97
Arizona		
Flagstaff	75	90
Phoenix	100	113
Tucson	84	98
Arkansas		
Fort Smith	91	103
Little Rock	90	100
California		
Bakersfield	96	114
Fresno	94	111
Los Angeles	83	94
Oakland	75	89
Sacramento	90	108
San Diego	75	80
San Francisco	75	83
Colorado		
Colorado Springs	83	94
Denver	83	98
Grand Junction	88	102
Pueblo	83	100
Connecticut		
Hartford	83	94
New Haven	83	95
New London	83	93
Norwalk	83	96
Delaware		
Dover	87	96
Milford	87	98
Wilmington	87	94
District of Columbia		
Washington	89	98
Florida		
Jacksonville	88	96
Miami	88	90
Orlando	88	97
Tallahassee	88	100
Tampa	88	95
Georgia		
Atlanta	87	95
Savannah	89	99
Idaho		
Boise	89	105
Pocatello	83	100
Illinois		
Cairo	89	101
Chicago	87	98
Peoria	88	100
Quincy	90	103
Rockford	87	101
Springfield	90	102
Indiana		
Evansville	90	100
Fort Wayne	87	100
Indianapolis	89	99
South Bend	87	101
Terre Haute	90	100
Iowa		
Burlington	90	101
Davenport	90	100
Des Moines	90	102
Dubuque	90	99
Keokuk	90	101
Mason City	86	97
Sioux City	90	102
Kansas		
Concordia	93	108
Dodge City	92	106
Hutchinson	92	108
Salina	95	111
Topeka	92	105
Wichita	91	104
Kentucky		
Lexington	86	98
Louisville	88	99
Louisiana		
Baton Rouge	88	98
New Orleans	89	98
Shreveport	92	102
Maine		
Eastport	70	81
Portland	81	93
Maryland		
Baltimore	89	99
Cumberland	87	102
Massachusetts		
Boston	84	94
Fall River	81	90
Lawrence	81	94
Worcester	81	92
Michigan		
Alpena	82	95
Detroit	86	99

* Do not use these temperatures for Air Conditioning Design; for that use Table 2, Chapter 11.

Table 4. Refrigeration Design Ambient Temperature Guide (Continued)

Location	Average ambient temp	Maximum ambient temp
Grand Rapids	86	98
Jackson	86	99
Lansing	86	96
Marquette	81	96
Saginaw	88	101
Minnesota		
Duluth	79	92
Minneapolis	90	102
St. Cloud	88	101
Mississippi		
Jackson	90	99
Vicksburg	90	96
Missouri		
Hannibal	90	102
Kansas City	92	103
St. Joseph	92	103
St. Louis	92	103
Springfield	88	98
Montana		
Billings	85	104
Butte	75	96
Havre	82	99
Helena	82	102
Nebraska		
Lincoln	94	106
North Platte	89	103
Omaha	92	104
Nevada		
Reno	84	101
Tonopah	84	96
New Hampshire		
Concord	81	92
New Jersey		
Atlantic City	83	92
Paterson	85	95
Trenton	85	96
New Mexico		
Albuquerque	83	99
Santa Fe	81	90
New York		
Albany	83	96
Binghamton	83	94
Buffalo	80	89
Elmira	83	97
New York	85	93
Poughkeepsie	83	95
Rochester	83	95
Syracuse	83	96
Watertown	83	93
North Carolina		
Asheville	81	93
Charlotte	86	98
Raleigh	86	98
Wilmington	86	95
Winston-Salem	86	97
North Dakota		
Bismark	87	103
Devils Lake	84	100
Ohio		
Akron	86	98
Canton	86	97
Cincinnati	88	100
Cleveland	83	95
Columbus	88	98
Dayton	88	99
Toledo	87	99
Youngstown	86	97
Oklahoma		
Oklahoma City	92	104
Tulsa	92	105
Oregon		
Portland	81	95
Pennsylvania		
Altoona	82	96
Erie	83	92
Harrisburg	85	97
Philadelphia	87	97
Pittsburgh	85	96
Scranton	82	95
Rhode Island		
Providence	83	94
South Carolina		
Charleston	88	98
Columbia	88	99
South Dakota		
Huron	93	107
Pierre	94	110
Rapid City	87	103
Sioux Falls	88	102
Tennessee		
Chattanooga	87	98
Knoxville	87	98
Memphis	89	99
Nashville	87	98
Texas		
Dallas	92	102
El Paso	92	102
Fort Worth	92	104

Table 4. Refrigeration Design Ambient Temperature Guide (Concluded)

Location	Average ambient temp	Maximum ambient temp
Houston	92	99
San Antonio	92	102
Utah		
Modena	80	97
Salt Lake City	88	101
Vermont		
Burlington	80	91
Virginia		
Lynchburg	87	99
Norfolk	87	95
Richmond	87	98
Washington		
Olympia	75	90
Seattle	75	86
Spokane	75	102
Walla Walla	87	105
West Virginia		
Charleston	87	102
Clarksburg	84	97
Huntington	87	100
Parkersburg	86	98
Wheeling	86	101
Wisconsin		
Green Bay	85	97
La Crosse	87	99
Madison	87	96
Milwaukee	87	99
Wyoming		
Cheyenne	79	94
Lander	80	98
Sheridan	86	102

Table 5. Allowance for Solar Radiation

(Degrees Fahrenheit to be added to the normal temperature difference for heat leakage calculations to compensate for sun effect—not to be used for air conditioning design)

Type of surface	East wall	South wall	West wall	Flat roof
Dark colored surfaces such as Slate roofing Tar roofing Black paints	8	5	8	20
Medium colored surfaces, such as Unpainted wood Brick Red tile Dark cement Red, gray or green paint	6	4	6	15
Light colored surfaces, such as White stone Light colored cement White paint	4	2	4	9

having a tar or asphalt covering would be calculated by adding 20 degrees to the ambient temperature which would be used with no sun exposure. Likewise, the ambient temperature used for sun-exposed side walls should be corrected accordingly. An average of the figures given for cardinal directions may be used for walls facing at angles between those directions. The temperatures given in Table 5 are applicable throughout the 24-hr day and hence are added to the ambient temperature in calculating the 24-hr heat gain. These values are not to be used in air conditioning calculations.

2. Air changes. Door openings through normal usage of the refrigerated space cause an air change which adds to the load, since the warm air which enters must be reduced to the temperature of the refrigerated space. The load thus imposed is very difficult to determine with any great degree of accuracy, for it is dependent on the number of times the door is opened, and the quantity, temperature, and humidity of the entering air.

As the volume of the refrigerated space increases, the quantity of air entering the space during a single door opening becomes a constant; for all practical purposes it may be assumed to be the same for all enclosures of 1,000 cu ft and larger. With a fixed air change quantity based on experience,

the additional load due to air changes is a function of the wet-bulb temperature of the entering air and the final temperature to which this air is reduced. **Table 6** gives values of additional load in Btu/hr which are based on experience and which provide a means of determining this load within reasonable limits.

For convenience, entering air conditions are given in terms of dry-bulb temperature and relative humidity. When the average relative humidity is not known, it is satisfactory to use the values given in the column headed 95 F, 50% relative humidity. The values given for entering air having dry-bulb temperatures of 40 F and 50 F are for refrigerated spaces having a refrigerated ante-room or vestibule between the opening to the main enclosure and the outside.

Should additional ventilation be required, the heat load may be calculated by the following formula:

$$Q = V/v_a \times (h_1 - h_2)$$

where

Q = ventilation load, Btu/hr
V = ventilation requirement, cu ft/hr

Table 6. Air Change Load Due to Door Openings, Btu/hr for Refrigerators 1,000 cu ft and Larger

Storage room temp F	Temperature of outside air, F							
	85		90		95		100	
	Relative humidity, %							
	50	60	50	60	50	60	50	60
65	545	715	780	985	1,040	1,290	1,330	1,640
60	715	865	950	1,150	1,210	1,460	1,500	1,810
55	940	1,130	1,190	1,400	1,450	1,690	1,730	2,050
50	1,110	1,300	1,360	1,570	1,620	1,870	1,920	2,230
45	1,260	1,460	1,510	1,730	1,780	2,040	2,080	2,400
40	1,420	1,610	1,680	1,900	1,960	2,200	2,250	2,580
35	1,560	1,760	1,830	2,040	2,090	2,340	2,400	2,720
30	1,680	1,880	1,900	2,130	2,220	2,470	2,480	2,820

Storage room temp F	Temperature of outside air, F									
	40		50		80		90		100	
	Relative humidity, %									
	70	80	70	80	50	60	50	60	50	60
30	175	210	420	475	1,420	1,570	1,900	2,130	2,480	2,820
20	400	440	655	710	1,720	1,870	2,200	2,440	2,800	3,140
10	610	640	860	920	2,000	2,120	2,460	2,690	3,060	3,400
0	810	840	1,070	1,125	2,250	2,400	2,760	3,000	3,380	3,720
−10	970	1,020	1,250	1,310	2,460	2,630	3,000	3,240	3,620	3,980
−20	1,175	1,210	1,450	1,500	2,720	2,900	3,240	3,520	3,920	4,290
−30	1,370	1,410	1,650	1,720	2,980	3,160	3,540	3,800	4,200	4,560

Notes: Based on one door in use at a time.
For heavy usage, multiply above values by 2.0.
For two doors each in adjacent walls and used simultaneously, multiply values by 1.75.
For two doors, each on opposite walls and used simultaneously, multiply values by 2.5.
For volumes 250–400 cu ft use 50% of values given.
401–600 cu ft, use 70% of values given.
601–999 cu ft, use 85% of values given.

v_a = specific volume of entering air, cu ft/lb
h_1 = enthalpy at entering air conditions, Btu/lb
h_2 = enthalpy at final conditions, Btu/lb

The values for v_a, h_1, and h_2 may be taken from a psychrometric chart, or from a table of properties of air and water vapor mixtures. (See Chapters 4 and 10.)

3. Product load. A product placed in a refrigerator at a temperature higher than the storage temperature will lose heat until it reaches the storage temperature. The quantity of heat to be removed may be calculated from a knowledge of the product, including its state upon entering the refrigerator and final state, its weight, specific heat above and below freezing, its freezing temperature, and latent heat. When a definite weight of product is cooled from one state and temperature to another state and temperature, some or all of the following calculations must be made:

Heat removal from initial temperature to some lower temperature above freezing:

$$Q = W \times c \times (t_1 - t_2)$$

Heat removal from initial temperature to freezing point of product:

$$Q = W \times c \times (t_1 - t_f)$$

Heat removal to freeze product:

$$Q = W \times h_{if}$$

Heat removal from freezing point to final temperature below freezing:

$$Q = W \times c_i \times (t_i - t_3)$$

where

W = weight of product in lb
c = specific heat of product above freezing, Btu/lb
t_1 = initial temperature above freezing, F
t_2 = lower temperature above freezing, F
t_i = freezing temperature of product, F
h_{if} = latent heat of fusion, Btu/lb
c_i = specific heat of product below freezing, Btu/lb
t_3 = final temperature below freezing, F

The product load is represented by the sum of the several loads given above. If it is required to remove the heat from the product in a given number of hours the heat load may be determined by the following formula:

$$\frac{\text{Product load}}{\text{Hours in which product is to be cooled}}$$

Specific heats above and below freezing and **latent heats of fusion** for many products are given in **Table 7.** It should be noted that the latent heat of fusion has a definite relationship to the water content of the product; in the absence of data on the latent heat of a product and with a knowledge of its moisture content, the latent heat may be estimated by multiplying the percentage of water expressed as a decimal by 144, the latent heat of fusion of water. Most food products have a freezing temperature in the range of 26 to 31 F, with an average freezing temperature of about 28 F. If the exact freezing temperature is unknown, it may be assumed to be 28 F. See Applications Volume, ASRE Data Book (Sec. I), for a discussion of food freezing.

To illustrate the foregoing methods, assume that 100 lb of lean beef are to be cooled from 90 to 40 F, after which it is to be frozen and cooled to 0 F.

From Table 7, the specific heat of lean beef before freezing is 0.77 Btu/lb and after freezing is 0.40 Btu/lb. The latent heat of fusion is 100 Btu/lb.

To cool from 90 F to 40 F in chill room:

$$100 \times 0.77 \times (90° - 40°) = 3{,}850 \text{ Btu}$$

To cool from 40 F to freezing point in freezer:

$$100 \times 0.77 \times (40° - 28°) = 924 \text{ Btu}$$

To freeze:

$$100 \times 100 = 10{,}000 \text{ Btu}$$

To cool from freezing point to storage temperature:

$$100 \times 0.40 \times (28° - 0°) = 1{,}120 \text{ Btu}$$

Total	15,894 Btu

Note that in this example the latent heat of fusion represents approximately two-thirds of the total load. In calculations involving freezing of a product the latent heat of fusion will always be a high percentage of that particular product load.

Table 8 gives usage load factors, Btu/hr cu ft F, which may be used for estimating loads which do not involve removal of latent heat of fusion. The values given in-

Table 7. Specific and Latent Heat Data for Foods

Product	Specific heat		Latent heat of fusion h_{if}	Average freezing temp F
	Above freezing	Below freezing		
Vegetables				
Artichokes	0.90	0.46	115	29.1
Asparagus	0.95	0.43	134	29.8
Beans, lima	0.89	0.41	100	30.1
Beans, string	0.92	0.47	128	29.7
Beans, dried	0.30	0.24	18	
Beets	0.86	0.47	129	31.1
Broccoli	0.91	0.47	130	29.2
Brussels sprouts	0.90	0.46	122	
Cabbage	0.93	0.43	129	31.2
Carrots	0.86	0.45	120	29.6
Cauliflower	0.94	0.48	134	30.1
Celery	0.96	0.49	137	29.7
Corn, green	0.80	0.43	108	28.9
Corn, dried	0.28	0.23	15	
Cucumbers	0.93	0.49	138	30.5
Eggplant	0.90	0.48	132	30.4
Endive	0.90	0.48		30.9
Lettuce	0.95	0.48	135	31.2
Mushrooms	0.90	0.47	130	30.2
Onions	0.90	0.46	127	30.1
Parsnips	0.86	0.45	120	28.9
Peas, green	0.80	0.42	108	30.0
Peas, dried	0.28	0.22	14	
Peppers, sweet	0.90	0.47	130	30.1
Potatoes, Irish	0.78	0.48	105	28.9
Potatoes, sweet	0.75	0.41	99	28.4
Pumpkins	0.89	0.47	130	30.1
Radishes	0.94	0.48	132	
Rhubarb	0.90	0.48	134	28.4
Sauerkraut	0.92	0.47	129	
Spinach	0.92	9.47	129	30.6
Squash	0.94	0.48	127	29.3
Tomatoes	0.95	0.48	135	30.4
Turnips	0.90	0.47	130	30.2
Meats, Fish and Poultry				
Bacon	0.50	0.30	28	
Beef, fresh lean	0.77	0.40	100	
Beef, fresh fat	0.60	0.35	79	
Beef, dried	0.34	0.27	21	
Fish, fresh	0.82	0.41	109	28.0
Fish, dried	0.56	0.34	65	

Table 7. Specific and Latent Heat Data for Foods (Continued)

Product	Specific heat		Latent heat of fusion h_{if}	Average freezing temp F
	Above freezing	Below freezing		
Game, fresh	0.80	0.42	115	
Hams, fresh	0.67	0.42	66	
Lamb	0.67	0.40	73	
Lard	0.54	0.31	90	
Liver, fresh	0.72	0.40	94	
Mutton, fresh	0.70	0.39	95	30.2
Oysters, in shell	0.84	0.44	114	
Oysters, in liquid	0.90	0.46	125	
Pigeons	0.78	0.41	99	
Pork, fresh	0.60	0.38	66	
Pork, salt	0.46			
Poultry, dressed	0.80	0.42	101	
Sausage—fresh (fat to lean)	.58–.72	.40–.43	43–93	
Sausage—smoked (fat to lean)	.58–.55	.35–.43	43–58	
Tenderloins, butts	0.74	0.40	97	
Veal	0.71	0.39	91	
Dairy Products				
Butter	0.64	0.24	19	
Buttermilk	0.92	0.48	121	
Cheese	0.64	0.44	50	
Cream, fresh	0.90	0.38	84	
Eggs	0.76	0.40	100	
Ice cream	0.74	0.40	96	
Milk, fresh	0.90	0.47	127	31.0
Oleomargarine	0.65	0.35	14	
Miscellaneous				
Beer	0.90			22–29
Chocolate—sweet	0.40			
Chocolate—bitter	0.50			
Flour, meal (wheat)	0.38	0.28	29	
Honey	0.35	0.25	26	
Water	1.00	0.508	144	32.0
Maple sugar	0.24	0.22	72	
Maple syrup	0.49	0.31	52	
Nuts, dried	0.28	0.24	14	
Fruits				
Apples	0.86	0.45	120	28.5
Apricots	0.92	0.46	122	
Avocadoes	0.90			27.2
Bananas	0.81	0.43	108	30.0
Blackberries	0.89	0.46	124	29.1

Table 7. Specific and Latent Heat Data for Foods (Concluded)

Product	Specific heat		Latent heat of fusion h_f	Average freezing temp F
	Above freezing	Below freezing		
Cantaloupes	0.92	0.34	128	28.8
Cherries	0.86	0.45	118	24.7–27.8
Cranberries	0.91	0.47	127	27.2
Currants			123	30.2
Dates	0.83	0.44	104	−4.1
Figs, fresh	0.85	0.45	112	29.0
Fruit, dried	0.47	0.32	43	
Grapefruit	0.92	0.47	128	28.4
Grapes	0.92	0.38	84	28.2
Gooseberries	0.92	0.47	130	28.9
Lemons	0.90	0.39	126	28.1
Melons	0.90	0.47	115	
Oranges	0.92	0.39	125	28.0
Peak peaches	0.90	0.46	125	29.4
Pears	0.86	0.45	120	28.5
Persimmons	0.83	0.45		28.0
Pineapple	0.90	0.46	127	29.9
Plums	0.87	0.45	121	28.0
Quinces				28.1
Raisins, dried				32.0
Raspberries	0.89	0.46	124	29.0
Strawberries	0.92	0.47	130	29.9
Watermelons				28.8

clude the door opening, product and miscellaneous loads including latent heat removal of the infiltration air but do not include the removal of latent heat of fusion of the products. The "average usage" factors are applicable to retail and wholesale refrigerators, and the "heavy usage" factors are particularly applicable to coolers subjected to heavy service, such as those in super-markets. Generally, the load may be determined with a much greater degree of accuracy by calculating the product load, air change and miscellaneous load separately than by the use of averaged load factors based on total volume.

In the storage of fresh fruits and vegetables there is an **additional load** which must be included. Fresh fruits and vegetables are alive and carry on within themselves many of the processes characteristic of living things. Due to the actions of enzymes and bacteria, certain changes in the color, texture, and chemical composition of the product take place which may cause serious deterioration of the product or even result in complete breakdown. The most important of these changes is produced by respiration, a process by which oxygen of the air is combined with the carbon compounds of the plant tissues, giving off carbon dioxide gas and water vapor. This chemical reaction is accompanied by the liberation of heat, the quantity of heat depending upon both the product stored and the storage temperature. The heat liberated is known as **heat of evolution.** It is

definitely a part of the refrigeration load and must be considered in addition to the heat load factors already mentioned.

If fruits and vegetables could be cooled instantly to their final temperatures, the addition of the heat of evolution would be at the liberation rate at the final temperature. However, the cooling process requires time, and the greater evolution of heat at the higher temperatures must be taken into consideration. In order to determine the additional heat accurately, it is necessary to know the rate of heat production at each temperature of the product during the cooling period, and the length of time the product is in each temperature range. For practical purposes, the heat of evolution during the pull-down period (cooling product from initial to final temperature) may be estimated by using the value of the heat of evolution at a temperature corresponding to the arithmetic averages of the initial and final temperatures (at freezing or higher). The heat of evolution at the final storage temperature may be calculated separately. Values for the rate of evolution are given in Table 11.

As an example, assume that 100 bu of apples are to be cooled from an initial temperature of 80 F to a final temperature of 35 F in a 3-day period. The refrigeration required to cool this quantity of apples would be calculated as follows:

Average weight per bu, 50 lb
Average temperature during pull-down period, 57 F
Specific heat of apples, 0.86 (from Table 7)
Average value of heat of evolution at 57 F, 4,600 Btu/ton/24 hr (interpolated from Table 11
To cool from 80 F to 35 F

$$100 \times 50 \times 0.86 \times (80-35) = 193{,}500 \text{ Btu}$$

Heat of evolution during pull-down period

$$\frac{100 \text{ bu} \times 50 \text{ lb/bu}}{2{,}000 \text{ lb/ton}} \times 4{,}600 \times 3 \text{ days} = 34{,}500 \text{ Btu}$$

Total product load during pull-down $= 218{,}000$ Btu

Product load per 24 hr

$$= \frac{218{,}000}{3} = 72{,}670 \text{ Btu/24 hr}$$

product load after final temperature has been reached.

Table 8. Load Factors for Large Refrigerators, Btu/hr cu ft F

Load factors include product load, door load and latent heat removal from infiltration air but do not include removal of the latent heat of fusion of the product. Factors based on 10 deg or less of product cooling.

Volume	Average usage	Heavy usage
1,000	.038	.061
1,500	.032	.053
2,000	.028	.047
2,500	.027	.044
3,000	.025	.042
3,500	.024	.040
4,000	.023	.039
5,000	.022	.037
6,000	.021	—
7,000	.019	—
8,000	.018	—
9,000	.017	—
10,000	.017	—
12,500	.015	—
15,000	.014	—
20,000	.013	—
25,000	.0127	—
30,000	.0125	—
40,000	.0124	—
50,000	.0123	
75,000	.0123	—
100,000	.0123	—

Average value of heat evolution at 35 F, 860 Btu/ton × 24 hr (interpolated from Table 11)

$$\frac{5{,}000}{2{,}000} \times 860 = 2{,}150 \text{ Btu/24 hr}$$

Note that in this example that the cooling of only 100 bu of apples is considered. If the apples are being loaded into the refrigerated space at the rate of 100 bu per day over a number of successive days, and still assuming they are to be cooled to the final temperature in three days after loading, then the total product load of 218,000 as calculated above for the three-day period, plus one-half of the respiration load at the final storage temperature, based on the total weight of apples placed in storage during the loading period, becomes the 24-hr product load until the last 100 bu have been loaded. Adding one-half of the respiration load of the total weight stored at final storage temperature is an approximation based on the average weight of the apples in the coolers during the loading period. After the temperature of the apples

Table 9. Heat Equivalent of Electric Motors

Motor hp	Btu/hp-hr		
	Connected load in refr space[1]	Motor losses outside refr space[2]	Connected load outside refr space[3]
$\frac{1}{8}$ to $\frac{1}{2}$	4,250	2,545	1,700
$\frac{1}{2}$ to 3	3,700	2,545	1,150
3 to 20	2,950	2,545	400

[1] For use when both useful output and motor losses are dissipated within refrigerated space; motors driving fans for forced circulation unit coolers.

[2] For use when motor losses are dissipated outside refrigerated space and useful work of motor is expended within refrigerated space; pump on a circulating brine or chilled water system, fan motor outside refrigerated space driving fan circulating air within refrigerated space.

[3] For use when motor heat losses are dissipated within refrigerated space and useful work expended outside of refrigerated space; motor in refrigerated space driving pump or fan located outside of space.

has been reduced to 35 F, the holding load is obtained by multiplying the heat of evolution per unit weight by the total weight of the apples in the storage.

4. Miscellaneous sources of heat include heat added through lighting, motors operating and people working. All electrical energy expended within the refrigerated space eventually becomes heat which must be absorbed by the cooling units and hence becomes a part of the daily load. **Electric lights** and **motors** give off heat in proportion to their electric power input. Each watt of the lighting load is equivalent of 3.42 Btu/hr. The heat equivalent of all motors operating within the refrigerated space must be added to the total load. Particularly, motors operating fans on forced circulation cooling units should not be overlooked, since these motors usually operate 24 hr per day, adding appreciably to the load. Heat equivalents of electric motors are shown in **Table 9.**

People radiate heat at a rate dependent on the type of work, rate of work and the surrounding air temperature, the greater exertion producing more heat discharge and also a lower ambient temperature causing greater heat leakage from the body. The human load, represented by the product of the number of people working, the hours worked per day and the heat equivalent per person in Btu/hr, must be added to the daily load. The hourly load due to occupancy is shown in **Table 10.** Refer to Applications Volume of ASRE Data Book (Sec. I) for a discussion of food freezing.

The foregoing three heat sources are those most usually encountered. If any other heat source is present it should be included in the total of miscellaneous sources.

For a discussion of optimum storage temperatures, relative humidity, and approximate storage life of perishable foods, refer to Chapters 12 to 15 of the Applications Volume of ASRE Data Book.

Thus the four principal sources of heat are individually determined. Calculations are usually facilitated by placing each load on a 24-hr basis. Then the separate loads are totaled to give the total load per 24 hr. If all conditions are not explicitly known, or if the usage will vary from the conditions given, it is generally advisable to add a 10% safety factor to the load obtained and base the equipment selection on the adjusted load.

The hourly refrigeration capacity required is contingent on the load. Generally, in order to obtain proper defrosting cycles and to provide reserve capacity for unusual peaks, it is common practice to base selection of the refrigeration equipment, both condensing unit and evaporators, on the basis of 16-hr operating time per day. Where no defrosting cycle is required, operation may be based on 18-hr operating time per day. In order to determine the Btu/hr capacity required, the total load per 24 hr is divided by the number of hours operating time required; i.e., 16 for 16 hr-operating time.

Example 1. Given: Refrigerator, 18 × 12 × 9 ft. 5 in. cork insulation, standard construction (2 layers of paper and lumber)

Table 10. Heat Equivalent of Occupancy

Cooler temperature F	Heat equivalent/person Btu/hr
50	720
40	840
30	950
20	1,050
10	1,200
0	1,300
−10	1,400

Design temperatures:
95 F ambient temperature
50% relative humidity
38 F refrigerator temperature
2,000 lb of lean beef coming in at 50 F, to be cooled to 38 F in 24 hr
Electrical load, 200 watts lighting, 1 $\frac{1}{8}$-hp fan motor.
Market Service—Heavy Usage

Solution:

972 sq ft outside surface
$95-38°=57°$ temperature reduction

(1) Wall loss
972×0.060 (Table 3) $\times 57 \times 24$
$=79,782$ Btu per 24 hr

(2) Air change load
2,012 (Table 6) $\times 24 = 48,286$ Btu per 24 hr

(3) Product load
Specific heat $=0.77$ Btu/lb (Table 7)
$2,000 \times .77 \times (50 \times 38) = 18,480$ Btu per 24 hr

(4) Electrical load
200 watts (lighting) $\times 3.42 \times 24$
$=16,500$ Btu per 24 hr
$\frac{1}{8}$ hp $\times 4,250$ (Table 9) $\times 24 = 12,750$ Btu per 24 hr
Total electrical load $=29,250$ Btu per 24 hr

(5) Total load, Btu per 24 hr

wall loss	79,782
air change	48,286
product load	18,480
electrical load	29,250
	175,798
add 10% safety factor	17,580
	193,378 Btu per 24 hr

(6) Required condensing units and evaporator capacity for 16-hr operation per day

$$\frac{193,378}{16} = 12,084 \text{ Btu/hr}$$

Using load factor values (Table 8)

(1) Wall loss (same as above) 79,782 Btu per 24 hr

(2) Usage load (heavy) approximate room volume $-1,500$ cu ft
$1,500 \times 0.53$ (Table 8) $\times 57 \times 24$
$=108,900$ Btu per 24 hr

(3) Total load $=79,782+108,900$
$=188,682$ Btu per 24 hr

(4) Required condensing unit and evaporator capacity for 16-hr operation per day

$$\frac{188,682}{16} = 11,800 \text{ Btu/hr}$$

Although in the foregoing example the calculation by the rule of thumb method is very close to the answer calculated by the detailed method, it is always safer to make complete calculations where all data are known. If the quantity of meat placed in the refrigerator had been different, or if it had been received at some other temperature, the result by the rule-of-thumb method remains unchanged, although obviously there would have been a change in the actual load.

Example 2. Given: Refrigerator, $15 \times 15 \times 9$ ft high
4 in. cork, two triple glass windows 6×5 ft each
90 F ambient temperature
40 F refrigerator temperature
Three men working from 8 a.m. to 4 p.m.
Electrical load—
200 watts lighting, 8 a.m. to 4 p.m.
1 $\frac{1}{4}$-hp band saw motor, 8 a.m. to 4 p.m.
1 $\frac{1}{6}$-hp fan motor 24 hr daily
3,000 lb beef to be cooled from 65 F to 40 F from 6 p.m. to 6 a.m.

Solution:
Net insulated surface (less glass area) $=930$ sq ft
Glass area $=6 \times 5 \times 2 = 60$ sq ft
Temperature reduction $=90-40=50$ F

(1) Wall loss
930×0.075 (Table 3) $\times 50 \times 24$
$=83,700$ Btu per 24 hr
60×0.29 (Table 3) $\times 50 \times 24 = 20,880$ Btu per 24 hr
Total, 104,580 Btu per 24 hr

(2) Air change load
(Relative humidity or wet-bulb temperature not given in conditions—assume 50%)
1,680 (Table 6) $\times 24 = 40,400$ Btu per 24 hr

(3) Product load
Specific heat $=0.77$ Btu/lb (Table 7)

$$\frac{3,000 \times .77 \times (65-40) \times 24}{12}$$

$=111,500$ Btu per 24 hr

(4) Miscellaneous Load

Day Load

Men working 3×840 (Table 10) $\times \frac{8}{12}$	1,680 Btu/hr
Saw motor $\frac{1}{4} \times 4,250$ (Table 9) $\times \frac{8}{12}$	706 Btu/hr
Fan motor $\frac{1}{6} \times 4,250$ (Table 9)	710 Btu/hr

Table 11. Approximate Rate of Evolution of Heat by Certain Fresh Fruits and Vegetables when Stored at the Temperatures Indicated*

Commodity	Temperature	Heat evolved per ton of fruits or vegetables per 24 hours*
	F	British thermal units
Apples	32	660 to 1,000
	40	1,110 to 1,760
	60	4,400 to 6,600
	85	6,600 to 15,400
Bananas:		
Green	54	3,300
	68	8,360
Turning	68	9,240
Ripe	67	8,360
Beets	32	2,650
	40	4,060
	60	7,240
Cantaloupes	32	1,320
	40	1,960
	60	8,500
Carrots	32	2,130
	40	3,470
	60	8,080
Celery	32	2,820
	40	4,540
	60	13,520
Cherries (sour)	32	1,320 to 1,760
	60	11,000 to 13,200
Grapefruit	32	460
	40	1,070
	60	2,770
	80	4,180
Grapes:		
Cornichon and Flame Tokay	36	660 to 1,100
	60	2,200 to 2,640
	80	5,500 to 6,600
Sultanina	32	430
	43	1,050
	53	1,690
Emperor	32	350
	43	850
	53	1,810
Ohanez	32	300
	43	740
	53	1,570
Lemons	32	580
	40	810
	60	2,970
	80	6,200
Lettuce	32	11,320
	40	15,990
	60	45,980
Mushrooms (cultivated)	32	6,160
	50	22,000
	70	58,000
Onions (Yellow Globe)	32	660 to 1,100
	50	1,760 to 1,980
	70	3,080 to 4,180
Oranges	32	690 to 900
	40	1,400
	60	5,000
	80	8,000
Peaches	32	850 to 1,370
	40	1,440 to 2,030
	60	7,260 to 9,310
	80	17,930 to 22,460
Pears (Bartlett)	32	660 to 880
	60	8,800 to 13,200
Peppers	32	2,720
	40	4,700
	60	8,470
Potatoes (Irish Cobbler)	32	440 to 880
	40	1,100 to 1,760
	70	2,200 to 3,520
Raspberries	36	4,400 to 6,600
	60	15,400 to 17,600
Strawberries	32	2,730 to 3,800
	40	5,130 to 6,600
	60	15,640 to 19,140
	80	37,220 to 46,440
String beans	32	4,740
	40	6,740
	60	22,630
Sweet corn	32	5,890
	40	8,190
	60	17,130
Sweet potatoes		
Not cured	32	2,440
	40	3,350
	60	6,300
Cured	32	1,190
	40	1,710
	60	4,280
Tomatoes (mature green)	32	580
	40	1,070
	60	6,230
Tomatoes (ripe)	32	1,020
	40	1,260
	60	5,640

* The figures in this table supersede those in Table 1, Circ. 278, U. S. Department of Agriculture, edition of November, 1941.

Lights 200 × 3.42 × $\frac{8}{12}$	455 Btu/hr
	3,551 Btu/hr

Night load
Fan Motor $\frac{1}{6}$ × 4,250 (Table 9) 710 Btu/hr

Comparison Day and Night Loads

	Btu per 24 hours
Night Load (A)	
Wall loss	104,580
Air change	nil
Product	111,500
Misc load 710 × 24	17,040
	233,120
10% Safety factor	23,300
Btu/24 hr	256,420
Day Load (B)	
Wall loss	104,580
Air change	40,400
Product load	none
Misc load 3,551 × 24	85,224
	230,204
10% Safety factor	23,020
Btu/24 hr	253,224

Total (A) is larger than Total (B) and should be used as the load.

Required condensing unit and evaporator capacity for 16 hrs. operation per day is

$$\frac{256,420}{16} = 16,050 \text{ Btu/hr}$$

It is to be noted that the miscellaneous day load of 8-hr duration has been distributed over a 12-hr period to avoid oversizing the condensing unit. This is satisfactory provided this load on an hourly basis does not exceed the capacity of the condensing unit, since in that case control of conditions would be lost. A check will show that condensing unit and evaporator capacity as determined above is greater than the maximum hourly load during any of the periods of heavy load; hence the required capacity as determined is satisfactory.

Example 3. Given: Refrigerator, 8 × 10 × 8 ft
8 in. cork insulation
Room temperature, 90 F
Refrigerator temperature, −10 F
400 lb poultry coming in at 40 F to be frozen in 24 hr
180 watts electrical load

Solution:
Outside surface, 448 sq ft
Temperature reduction = 90° − (−10°) = 100 F

(1) Wall loss
448 × .0375 (Table 3) × 100 × 24 = 40,320 Btu per 24 hr

(2) Air change
(relative humidity or wet-bulb temperature not given in conditions—assume 50%)
Volume (approx) = 385 cu ft
3,000 × 50% (Table 6) × 24 = 36,000 Btu per 24 hr

(3) Product load
Specific heat above freezing, 0.80 Btu/lb (Table 7)
Specific heat below freezing, 0.42 Btu/lb (Table 7)
Latent heat of fusion, 101 Btu/lb (Table 7)
Freezing point not given—assume 28 F
Sensible cooling to freezing temperature
400 × (40° − 28°) × .80 = 3,840 Btu
Latent heat of fusion
400 × 101 = 40,400 Btu
Sensible cooling to final temperature after freezing
400 × (28 − (−10) × 0.42 = 6,384 Btu
Total product load = 50,624 Btu per 24 hr

(4) Electrical load
180 × 3.42 × 24 = 14,750 Btu per 24 hr

(5) Total load

Wall	40,320
Air change	36,000
Product	50,624
Electrical	14,750
	141,694 Btu per 24 hr
Add 10% safety factor	14,170
	155,864 Btu per 24 hr

Required condensing unit and evaporator capacity for 18-hr operation (longer operating time used if no defrosting cycle contemplated),

$$\frac{155,864}{18} = 8,826 \text{ Btu/hr}$$

20. MOTORS AND ELECTRICAL CONTROLS

THE principal applications in the refrigerating industry requiring electric motors are reciprocating compressors, fans and centrifugal compressors. The types of motors required for these applications depend on a number of factors such as the torque requirements of the driven device, the degree of unloading during the starting operation, and the electrical power supply available. Consideration of the proper motor to do the job also determines the type of enclosure required for the motor to protect it against severe atmospheric conditions.

Fractional hp motors. There are two general classifications of motors in the industry—the fractional hp motors, and the integral hp motors. Fractional hp motors comprise ratings smaller than 1 hp at 1,800 rpm, and are of a wide variety of types and sizes depending on the application. Most motors in this classification are single-phase, although occasionally polyphase and dc motors are employed.

The various types of motors and their usual application are as follows:

Type	*Maximum hp*	*Applications*
Split-phase	$\frac{1}{3}$	Small compressors & fans
Capacitor start-induction run	$\frac{3}{4}$	Compressors & fans
Repulsion induction	$\frac{3}{4}$	Compressors & fans
Capacitor start & run	$\frac{3}{4}$	Overhung fans—unit heaters & coolers (not suitable for belted application)
Shaded pole	$\frac{1}{10}$	Overhung fans—unit heaters & coolers (not suited for belted loads)
Direct current	$\frac{3}{4}$	Compressors & fans

Integral hp motors. In the integral horsepower classification, single-phase motors are employed on many applications, usually up to a maximum of 5 hp, but occasionally up to 10 hp in those cases where polyphase power is not available. The various types of motors and their applications in the refrigerating industry are as follows:

Type	*hp*	*Applications*
Single-phase—capacitor start-induction run	1 to 5	Compressors & fans
Single-phase—repulsion induction	1 to 5	Compressors & fans
Squirrel-cage—normal torque 2 or 3-phase	1 & up	Compressors started unloaded and fans
Squirrel-cage—high torque 2 or 3-phase	1 & up	Compressors started loaded
Slip ring—2 or 3-phase	1 & up	Fans, centrifugal compressors and reciprocating compressors
Synchronous—2 or 3-phase	50 & up	Centrifugal compressors and reciprocating compressors unloaded

Single-Phase Motors

Split-phase motors. The split-phase motor depends upon the principle of a stator having two windings, one displaced with respect to the other, causing a magnetic drag on the conductors of a squirrel-cage motor, in order to produce the starting operation. One of the windings in the stator, called the running winding, is made of relatively coarse wire, and the other winding, or starting winding, is made of very fine wire. These two windings are connected in parallel across the single-phase line when starting. The starting winding by reason of its fine wire has high resistance, and, therefore, higher power factor than the main winding, which causes the phase displacement between the two windings necessary to produce a rotating magnetic field. When a predetermined speed is reached, approximately 70% of synchronous speed, a centrifugal device on the squirrel-cage rotor opens a switch in series with the starting winding, and thereafter the motor operates as a single-phase, squirrel-cage induction motor, drawing current only through the running winding.

These motors are available in various ratings from $\frac{1}{8}$ hp up to and including $\frac{1}{3}$ hp, and with either rigid or resilient mounting, depending upon the requirements of the application. They are suitable for application either direct-connected or with belt drives to small fans and blowers.

These motors are provided either with four leads, or with a terminal block, in order to permit reversing the direction of rotation. To reverse the direction of rotation, it is merely necessary to reverse the two leads to the starting winding. A typical speed torque curve of this type of motor is shown in **Fig. 1.**

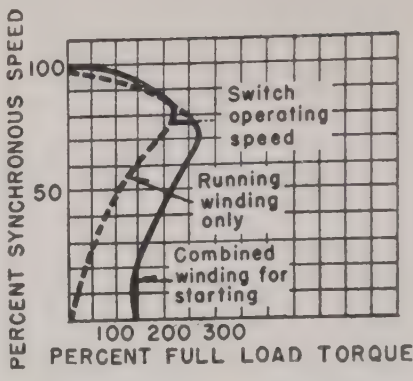


Fig. 1. Speed Torque Curves of General-Purpose Split-Phase Motors

Another type of split-phase motor, called the high torque, split-phase motor, is sometimes used in refrigerating and air conditioning work, although it is not recommended for such applications on account of the fact that the starting currents for this type of motor are high enough to permit possibility of lamp flicker. For this reason this high-torque motor normally used on washing machines, ironers, and home work-shop machines, is not used in refrigeration work requiring fairly frequent starting.

The general-purpose split-phase motor, and all other types of fractional hp motors described in the following paragraphs, have locked rotor currents within the following NEMA Standards for 115-volt motors, which standards are approved by practically all power companies.

$\frac{1}{6}$ hp and below	—20 amperes
$\frac{1}{4}$ hp	—23 amperes
$\frac{1}{3}$ hp	—31 amperes
$\frac{1}{2}$ hp	—45 amperes
$\frac{3}{4}$ hp	—61 amperes

The minimum efficiencies, power factors and apparent efficiencies of 60-cycle, general-purpose motors of fractional hp ratings, are shown in **Table 1.** These figures are, in most cases, conservative compared with the performance of motors which may be obtained on the market. The starting currents for motors on 230-volt lines are half the above figures.

Capacitor start-induction run motors. This type of motor is available not only in fractional hp ratings, but also in integral horsepower ratings up to a maximum of 5 hp. In principle they are fundamentally the same as the split-phase type of motor, except that the starting winding is provided with a capacitor in series with it, and in this manner it is possible to obtain high starting torques without exceeding the

Table 1. Minimum Efficiencies, Power Factor, and Apparent Efficiencies of 60-Cycle General-Purpose Motors, Percentage

(NEMA Standards, MG-18-101, July, 1941)

Horsepower	Efficiency				Power factor				Apparent efficiency			
rpm	3,600	1,800	1,200	900	3,600	1,800	1,200	900	3,600	1,800	1,200	900
poles	2	4	6	8	2	4	6	8	2	4	6	8
$\frac{1}{8}$	45	53	45	38	57	52	43	36	28	30	21	15
$\frac{1}{6}$	49	58	49	42	62	56	46	38	34	36	25	18
$\frac{1}{4}$	53	62	53	45	66	60	49	40	39	42	29	20
$\frac{1}{3}$	54	63	54	46	67	61	50	41	41	44	31	22
$\frac{1}{2}$	55	65	55	47	69	63	52	43	44	47	33	23
$\frac{3}{4}$	57	67	57	49	72	65	53	44	46	49	34	24

Note: Apparent efficiency = (Efficiency × power factor) /100.

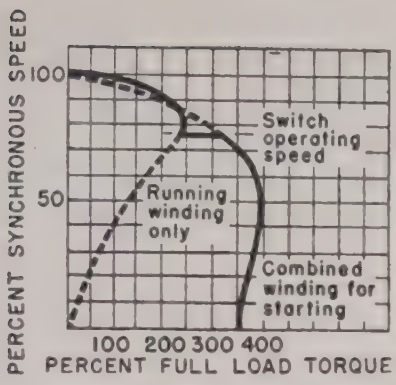


Fig. 2. Speed Torque Curves of Capacitor Start-Induction Run Motors

starting current limitations mentioned in the above tabulation. A typical speed torque curve for this type of motor is shown in **Fig. 2.**

The capacitor start-induction run motors, because of their high starting torque and low starting current, are ideal for compressors, refrigeration and air conditioning equipment. Typical performance specifications for this type of motor are shown in **Table 2.**

The integral hp motors of the same type have speed torque curves fundamentally the same as for the fractional hp rating, except that the various torque values are slightly lower, in order to hold the starting current to a minimum. They will have a starting torque of 300 to 325%, a torque at transfer of 225 to 250%, and a pull-out torque of at least 200%.

There is such a wide variety of possible connections for capacitor type motors that these connections will not be covered. The nameplate on the motor usually gives sufficient information for making the proper connections.

Capacitor start and run motors. In this type of motor there are two windings in the stator, the same as in the capacitor start-induction run motor, except that both windings are left in the circuit at all times. A capacitor is provided in series with the second winding, and the value of the capacitor is selected to give the best operating condition with the motor at full speed. For this reason, this type of motor has an exceptionally low starting torque and therefore is not recommended for belted applications. Its principal field of application is that of unit heaters and unit coolers where the fan is mounted directly on the motor shaft. In these cases the motor is usually totally enclosed, depending for its cooling on the air being drawn over the motor by the fan. Its principal advantage over other types of motors is that it does not require the use of any centrifugal device for cutting out a starting winding. This type of motor lends itself readily to speed control down to 50% of full-load speed.

Shaded pole motors. Very small fans are often driven by small shaded pole motors. These motors, also single-phase, are available in sizes up to about $\frac{1}{10}$ hp. In this type of motor, the stator is provided with a single winding consisting of one coil around each salient pole, and the coils connected in series across the line. In order to produce the rotating magnetic field necessary to produce starting torque, each pole piece is provided with a shading coil which is a single turn of copper surrounding only half of the pole iron. In operation, the al-

Table 2. Typical Specifications for Fractional hp Capacitor Type Refrigeration Compressor Motors

(Values shown are representative for 115-volt, 60-cycle, 4-pole, single-phase motors)

	Horsepower rating				
	$\frac{1}{6}$	$\frac{1}{4}$	$\frac{1}{3}$	$\frac{1}{2}$	$\frac{3}{4}$
Full load current amperes	3.0	3.7	4.6	7.0	10.0
Locked-rotor (starting) current, amperes	16.2	22.0	25.0	40.0	58.0
Full load efficiency, percentage	63.0	68.0	70.0	72.0	73.0
Full load power factor, percentage	57.0	62.0	66.0	64.0	65.0
Full load torque, oz ft	8.1	12.2	16.2	24.4	36.5
Break-down (maximum) torque, oz ft	25.0	34.0	45.0	71.0	106.0
Locked-rotor (starting) torque, oz ft	40.0	57.0	69.0	109.0	155.0

ternating current in the main winding sets up flux in the field core 90 electrical degrees out of phase with the current. This flux by transformer action induces current in the shading coil, and the current in the shading coil sets up flux in the half of the iron which it surrounds. This type of motor has relatively high slip. The rotor is a simple squirrel-cage type and no centrifugal switch mechanism is employed.

Very low starting torques are developed with this type of construction, being of the order of 40 to 50% of full load torque so that the application of the motor is limited to direct-connected fans only, and should never be belted. Its efficiency and power factor are exceptionally low, but these factors are not of much importance in motors of such small size. The physical location of the shading pole determines the direction of rotation, and the rotation of the motor cannot be changed in the field except by dismantling the motor and bringing the shaft out the opposite end of the motor, providing, of course, the construction permits.

Repulsion start-induction run motors. This type of single-phase motor also finds a very wide field of application in the refrigeration and air conditioning industry, particularly for belt drive to small size compressors not only in the fractional horsepower sizes, but also for integral horse power ratings up to 5 hp. Occasionally they will be used up to 10 hp.

A straight single-phase repulsion motor has a distributed winding in the stator, while the rotor is similar to a direct-current armature with windings connected to a commutator. When the brushes are all short-circuited by connecting them together, and are set in the correct position, this motor is capable of developing a very high starting torque. However, instead of having a constant speed characteristic so essential to compressor drive, it has a drooping speed torque curve. To take care of this condition, the repulsion start-induction run motor is provided with a centrifugal device that connects all the commutator bars together at a predetermined speed, thereby converting the armature to a squirrel-cage rotor.

Thus this motor combines the high starting torque of the repulsion motor with the constant speed characteristic of the squirrel-cage motor while running. Some motors of this type are so designed that the brushes are automatically lifted from the commutator immediately after the bars have been connected. This feature eliminates brush noise and wear while running.

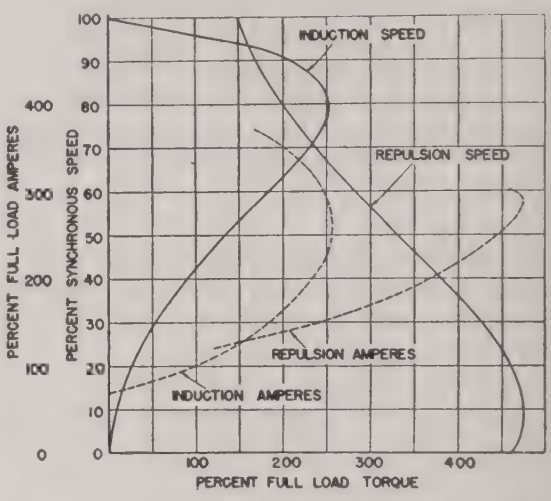


Fig. 3. Speed Torque and Current Torque Curves of Typical Repulsion Start-Induction Run Single-Phase Motor

Fig. 3 shows the typical performance of the repulsion start-induction run motor. The motor torque follows the repulsion torque curve up to the point where the centrifugal device short circuits the commutator. Thereafter it follows the induction curve. The optimum speed for short-circuiting the bars occurs when the repulsion and induction curves cross.

Table 3 shows minimum torque values for fractional hp repulsion start-induction run motors as established by NEMA.

The repulsion-induction motor is similar in construction, but without any device for short-circuiting the commutator or lifting the brushes. Instead, it has a squirrel-cage winding deeply embedded in the rotor underneath the repulsion winding. As a result, the speed torque curve is a modification of the combined torque curves of both windings over the entire speed range.

These two types of motors have the advantage that they develop more starting torque per ampere than any other type of single-phase motor. However, because of commutator and brush maintenance, high

Table 3. Minimum Torque Values of Repulsion Start-Induction Motors, in Percentage of Full Load Running Torque

(NEMA Standards MG8-104, July, 1941)

hp	Rating 60 cycles, 1,725 rpm			Rating 60 cycles, 1,140 rpm		
	Locked rotor	Pull-up	Breakdown	Locked rotor	Pull-up	Breakdown
$\frac{1}{8}$	350	200	200	300	150	185
$\frac{1}{6}$	350	200	200	300	150	185
$\frac{3}{4}$	350	200	200	300	150	185
$\frac{1}{3}$	350	200	200	300	150	185
$\frac{1}{2}$	350	200	200	300	150	185
$\frac{3}{4}$	350	200	200	—	—	—

manufacturing costs and radio interference during starting, they are gradually being replaced by the capacitor start-induction run type motor.

Polyphase Induction Motors

Polyphase induction motors are the simplest and most rugged, and because of these features are more universally used in industry than any other type. They have a wide variety of characteristics, depending on the application, and are available over the complete range of horsepowers from the smallest fractional horsepower up to the largest integral ratings. They can be obtained for either two or three-phase systems for operation on 110, 208, 220, 440, or 550-volt lines, and in the larger sizes for operation on 2,300 volts or even higher. Most systems in the United States are 60-cycle, but motors can also be obtained for 25, 40 or 50 cycles where the power systems are of these frequencies.

For the purpose of describing the principle of operation the two-phase is probably the more simple. It has two sets of stator windings located in the motor 90 electrical degrees apart, and respectively connected across the two phases of the line. The phase displacement between the currents in the two windings produces a rotating magnetic field, causing a drag on the conductors of the rotor, thus enabling it to start and come up to full speed. The rotor is usually made up of laminations with a simple uninsulated bar type winding in it. This type is called the **squirrel-cage motor.** By varying the shape, size and resistance of these bars, the torque characteristics can be varied to meet the particular application for which the motor is intended. The rotors can also be of the slip ring or wound rotor type.

Squirrel-cage type motors are divided by NEMA into several classes giving a wide variety of speed torque characteristics. These classifications are:

Class A—Normal starting torque with normal inrush current
Class B—Normal starting torque with low inrush current
Class C—High starting torque with low inrush current
Class D—High slip motors.

NEMA Standards MG-4-35 have assigned a letter designation to appear on nameplates of all motors ½ hp and larger to indicate the locked rotor KVA per hp input to the motor. This standard is shown in **Table 4.** The locked rotor KVA with respect to the locked rotor starting current bears the following relationships:

$$\text{KVA} = \frac{\sqrt{3} \times \text{volts} \times \text{locked amperes}}{1{,}000} \qquad \text{For 3-phase motors}$$

$$\text{KVA} = \frac{2 \times \text{volts} \times \text{locked amperes}}{1{,}000} \qquad \text{For 2-phase motors}$$

$$\text{KVA} = \frac{1 \times \text{volts} \times \text{locked amperes}}{1{,}000} \qquad \text{For single-phase motors}$$

Class A normal torque-normal inrush motors. This type of motor, usually called the general-purpose motor, is rapidly being superseded by the Class B motor. It may have starting currents as high as eight times or even ten times full load current, and for this reason it is not looked upon with favor, even though its efficiency and power factor may be very slightly better than those of the Class B motor.

It can be applied to fans, blowers, centrifugal pumps and to reciprocating compressors when started unloaded.

Class B normal torque-low inrush motors. The normal torque-low inrush motor is more widely used in the refrigerating and air conditioning industry than any other type. It is successfully applied to fans, blowers, centrifugal pumps and to reciprocating compressors when started unloaded.

The locked rotor or starting torques and the maximum running torques of this class of motors are shown for different pole combinations in **Table 5.**

Fig. 4 shows a typical speed torque curve for this type of motor as well as the Class C high torque motor.

In **Fig. 5** are shown full load efficiencies and power factors for the average motors of different horsepowers and speeds from 1 up to 400 hp. In the average motor, the efficiencies for ½ and ¾ load will be 1 and 2 or 2 and 3%, respectively, below the figures on the curves for full load. A typical efficiency and power factor curve is shown in **Fig. 6.**

Table 4. Letter Code Designation Showing Starting KVA per hp for Alternating Current Motors ½ hp and Larger

Letter designation	KVA per hp
A	0 – 3.14
B	3.15– 3.54
C	3.55– 3.99
D	4.0 – 4.49
E	4.5 – 4.99
F	5.0 – 5.59
G	5.6 – 6.29
H	6.3 – 7.09
J	7.1 – 7.99
K	8.0 – 8.99
L	9.0 – 9.99
M	10.0 –11.19
N	11.2 –12.49
P	12.5 –13.99
R	14.0 –and up

Note: Motors manufactured prior to the date of issuance of this Code do not include the KVA per hp on the nameplate.

Table 5. Locked Rotor and Maximum Running Torque Values for NEMA Class B Squirrel Cage Motors

No. of poles	Min locked rotor torque, %	Max running torque, %
2	150	200
4	150	200
6	135	200
8	125	200
10	120	200
12	115	200
14	110	200

Note: The locked rotor torque and maximum running torque for Class B motors will not be less than the values shown in table above except that some 2-pole designs have lower locked rotor torques than 150%.

Class C high torque-low inrush motors. By means of a rotor having special shapes of bars in the squirrel-cage winding, high starting torques in the neighborhood of 200 to 250% of full load torque are developed without a material sacrifice in running efficiency and power factor. This type of motor, somewhat more expensive than the Class B, is called the Class C high torque-low inrush motor. Its typical speed torque curve is shown in Fig. 4 along with the curve for the Class B motor.

The principal field of application for this motor is for starting reciprocating compressors which are required to be started up with load.

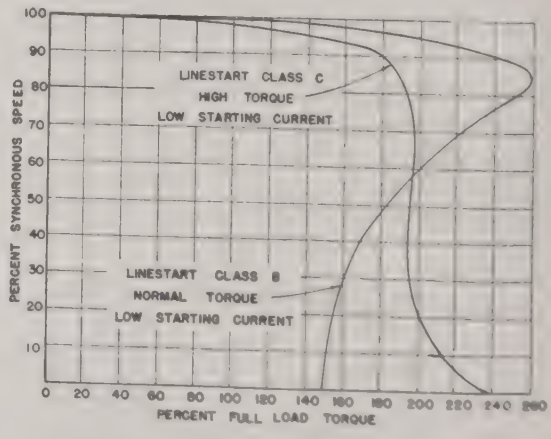


Fig. 4. Speed Torque Curves, Class B and C Motors

Class D high slip motors. This motor is seldom used in the refrigerating and air conditioning industry.

Multi-speed squirrel-cage motors. On many fan applications and compressors it is desirable to have more than one speed and to obtain it as simply and inexpensively as possible. Two-speed, three-speed and four-speed motors in many cases will fulfill the requirements even though the speed control is not so close as desired. Where only two speeds are required with a ratio of two to one in speed, this can be accomplished with one winding in the motor, reconnected externally by means of a drum switch or contactors to give the second speed. The connections for a two-speed variable or constant torque motor such as would be required for a fan drive are shown in **Fig. 7.**

Where a speed range as high as two to one is not desired, the stator must be provided with two separate windings and inter-locked in the control, so that only one can be energized at any one time. In other words, the motor functions as two separate motors in one housing.

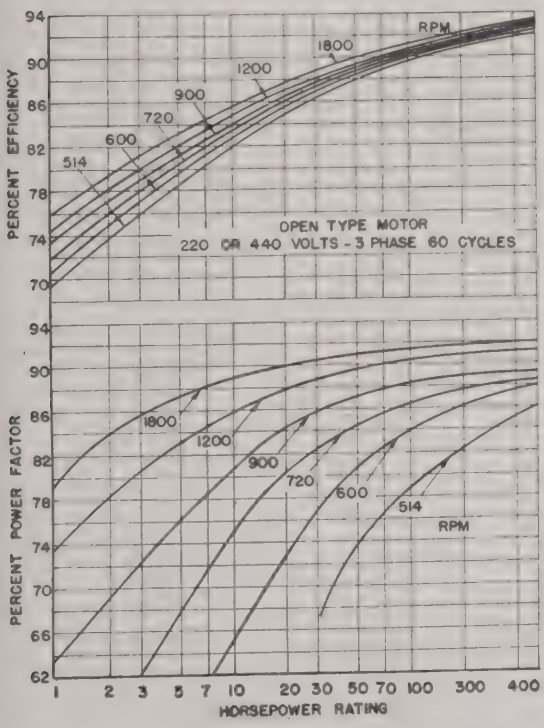


Fig. 5. Representative Full Load Efficiencies and Power Factors of NEMA Class B Squirrel Cage Motors

If three or four speeds are desired, they can be obtained with two stator windings, each one a duplicate of the consequent pole winding shown in Fig. 7.

Wound rotor (slip ring). In those cases where a speed range of two to one is required with quite a number of operating speeds between these limits, or where the motor is of such a large size that the power company system cannot stand the starting current of a squirrel-cage motor, the wound rotor (slip ring) motor is often employed.

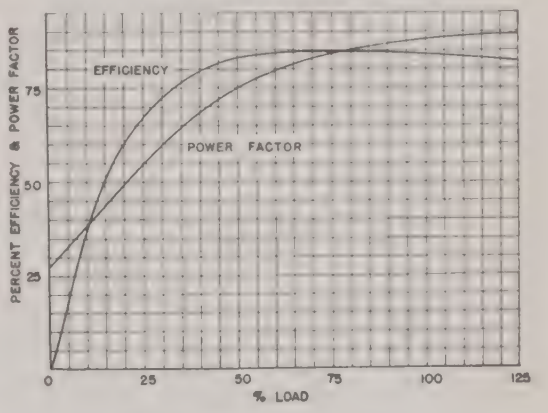


Fig. 6. Motor Efficiencies and Power Factors for Motors at Part Load

This motor is similar in construction to the squirrel-cage motor as far as the stator is concerned, but the rotor, instead of being uninsulated, is provided with the same insulated type of winding and the three leads connected to collector rings. Carbon brushes on the rings are connected to an external resistor and drum controller or contactors to cut the resistance out of the circuit with each successive step of the control. With sufficient resistance in the external rotor circuit the initial starting torque, and likewise the initial starting current, can be reduced to full load values or even lower.

Fig. 8 shows a typical set of speed torque curves of a wound rotor motor with a seven point control driving a fan or centrifugal compressor, where the torque required by the fan varies as the square of the speed. On the first point of the controller, it will be observed that the motor will develop approximately 70% starting torque with approximately 75 to 80% of full load current. Since this torque is in excess of the break-

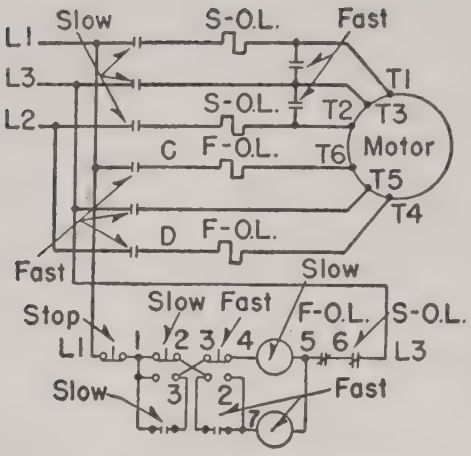


Fig. 7. Schematic Connection Diagram for Two-Speed, Single Winding Variable or Constant Torque, Squirrel-Cage Motor with Magnetic Control

away torque requirements of the fan, the motor will start to accelerate the load and eventually bring the fan up to 55% speed. Then, if the controller is moved over to the second point, speed torque curve, R-2 applies and the motor will accelerate still further up to 65%. By further increase in the controller, the motor and fan will be brought up to the final or top speed without causing excessively high currents to be drawn from the line. If the operator does not remain on point one of the controller, but moves rapidly over to point two, the initial starting torque will be 125% of full load torque, and the initial starting current likewise of the same proportion.

This type of motor, particularly in the larger sizes, is admirably suited to driving centrifugal compressors through step-up gearing. Where reduced refrigerating capacity is required on moderately warm days in air conditioning systems, it is more economical from a power consumption standpoint to operate the compressor at reduced speeds rather than run it at full speed and vary the output of the compressor by damper control.

Hermetic Motors

According to NEMA, "A hermetic motor consists of a stator and rotor without shaft, and shields or bearings for installation in refrigeration condensing units of the hermetically sealed type."

In other words, with the motor and compressor sealed in a common chamber, its

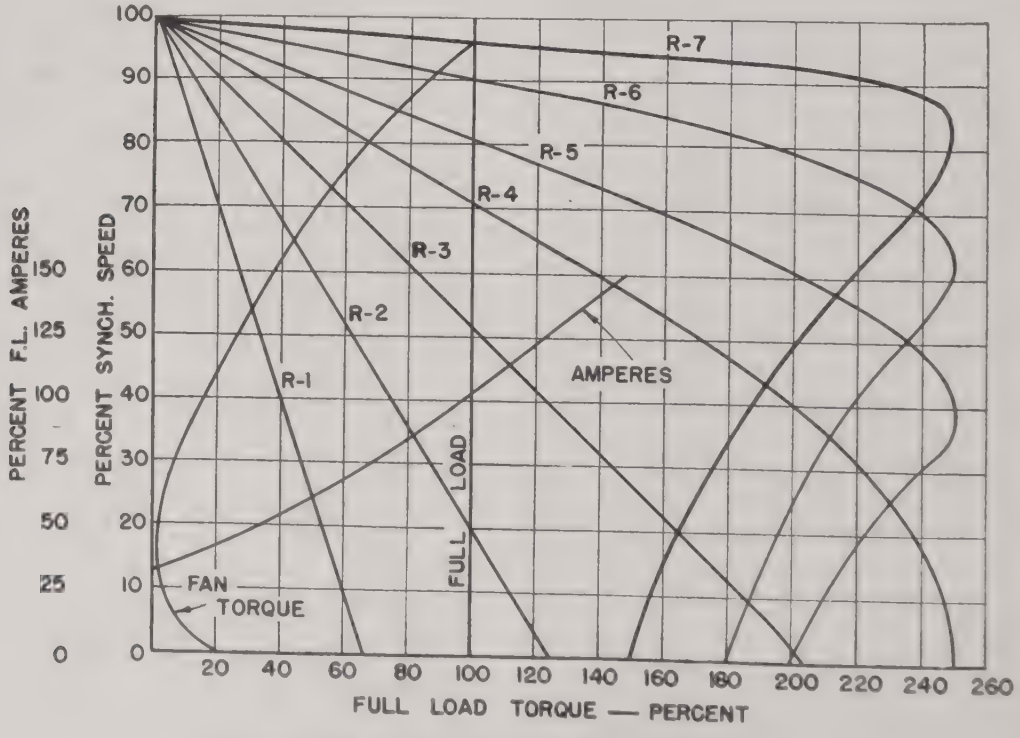


Fig. 8. Speed Torque Curves of Wound Rotor Motor

construction must be such that its insulation can withstand the action of the refrigerant and lubricating oil. Special selected long-fiber cotton treated with special compounds and a few types of resin-coated wires are acceptable to the application in that they neither foul the refrigerant and lubricant, nor deteriorate due to their action.

Hermetic motors are available through the fractional horsepower range in the split-phase, capacitor start and two-value capacitor single-phase types of construction. In these cases, starting switches and auxiliaries are mounted external to the compressor unit.

Integral horsepower ratings are available for single-phase power, but by far the highest percentage of installations above 3 hp are 3-phase. Hermetically sealed units are available up to 100 hp.

Temperature rise. Unlike the normal open-type motor, the hermetic motor can be cooled by a variety of means such as air, water, refrigerant vapor, refrigerant liquid, oil or a combination of these means.

The cooling of an hermetic motor may vary with compressor operating conditions as much as with load, so that the temperature rise may serve as a limitation under one set of compressor operating conditions and the maximum torque requirements under another set.

Inasmuch as the temperature rise of a hermetic motor depends as much upon the design of the compressor and cooling system as upon the motor losses, the motor manufacturer ordinarily does not guarantee the temperature rise of a hermetic motor, and the compressor manufacturer is ultimately responsible for the temperature rise.

Torque values of hermetic motors. In the fractional hp type, hermetic motors are available in low, normal and high starting torque designs. In the normal and high torque designs, the starting torques are in the range of 300 to 400% of full load torque and the break-down torque approximately 250%.

In the integral hp polyphase type, the starting torques will vary from 250 to 300% of full load torque with the break-down torques having the same values.

Locked rotor currents or starting currents of hermetic motors usually do not exceed the values recommended by NEMA for complete induction motors of similar rating.

Synchronous Motors

Synchronous motors, because of their higher efficiency and ability to correct power factor, have a very definite field in the refrigerating and air conditioning industry, particularly in the larger sizes, 100 hp and above. Their principal fields of application are in driving slow-speed reciprocating compressors and in furnishing power through step-up gearing to centrifugal compressors used in air conditioning.

Synchronous motors differ from induction motors in that the rotating field receives its excitation from a source of direct current, either in the form of a direct-connected exciter, or from direct current bus excitation. Usually in the air conditioning and refrigeration field, they are used on 220, 440, or 2,300-volt systems. Whereas induction motors always operate with a lagging power factor, the synchronous motor operates at unity power factor or a leading power factor. Leading power factor machines are usually designed for 80% where a considerable amount of correction is required. Both the 100% and 80% machines, when operating at reduced load, furnish additional power factor correction in accordance with **Fig. 9.**

The unity power factor synchronous motors are usually 1 to 2% higher in efficiency than the corresponding rating of induction motor either of the squirrel-cage or wound rotor types. The 80% power factor synchronous motor, because it has a higher full load current and, consequently, higher losses than the 100% power factor machine, has just about the same efficiency as the induction motor of the same horsepower rating.

Slow-speed synchronous motors. Slow-speed synchronous motors of the engine type construction (that is, without shaft or bearings) are usually employed to drive slow-speed reciprocating compressors. These motors have speeds of 450 rpm and lower. Because of the fact that such compressors are always started up unloaded, the starting torque and pull-in torque requirements are quite low, usually on the

order of 30 to 50% of full load torque. Because of these low torque requirements for starting, the current inrush is usually in the neighborhood of 250 to 275% of full load current for the unity power factor motors, and 225 to 250% of full load current for the 80% machines. With these low values of starting inrush, this type of motor is invariably started across full line voltage.

High-speed synchronous motors. The high-speed synchronous motor, also available for unity power factor or 80% leading power factor operation, finds its principal application in driving high-speed centrifugal compressors through step-up gearing. As mentioned previously, the wound rotor type of motor is more economical from the power consumption standpoint where load variation is required, but there are many installations involving more than one compressor where the base refrigeration load can be handled by the synchronous motor type of drive, and the variations in load taken care of by the slip ring motor on another compressor.

There is also another form of centrifugal compressor drive utilizing the synchronous motor where varying load requirements are encountered, and that is the drive utilizing a variable speed fluid coupling between the synchronous motor and the step-up gearing. The overall efficiency of this drive is just about comparable with the slip-ring motor application. It works out more advantageously in those cases where space is at a premium because of the relatively large space requirements of the secondary resistor for the wound rotor application.

High-speed synchronous motors inherently have higher starting currents than the low-speed type of motor, so that even though the torque requirements for starting the centrifugal compressor are quite low the inrush on full voltage is in the neighborhood of 500% of full load current. Some power systems can satisfactorily handle this inrush, while on others it is necessary to resort to reduced voltage starting. In applications of this nature, it is imperative that the particular power company involved be consulted regarding starting current limitations. The motor and control manufacturer should be provided with complete information as to the horsepower requirements, speed and WR^2 of the compressor, together with start-current limitations and interrupting capacity of the power system involved.

Direct-Current Motors

In some cases where alternating current is not available, direct-current motors are used to drive the various fans, blowers, pumps, and compressors required by this industry. This type of motor, available over the complete range from the smallest fractional horsepower up to the largest integral horsepower rating, will always have ample torque available for acceleration for running loads within its rating. Motors are rated either 115 or 230 volts dc.

Fractional horsepower motors, except the extremely small ratings, are all compound wound, and can be thrown directly across the line. Integral horsepower motors can be furnished either as shunt wound or compound wound machines. If compound wound, they can be thrown across the line up to 3 hp, and for voltages 230 volts and below. Larger compound wound motors, and all integral horsepower, shunt wound

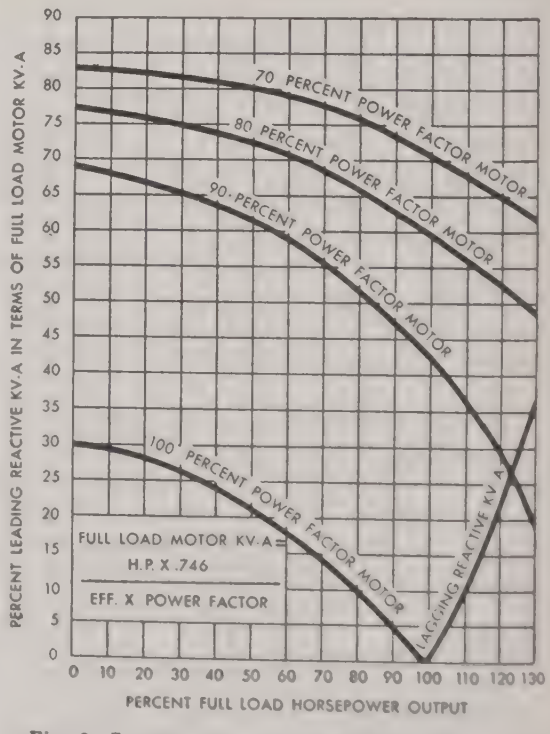


Fig. 9. Leading Reactive KVA Obtainable with Synchronous Motors under Various Load Conditions

motors require a starting resistance in series with the armature.

One advantage of the shunt wound dc motor over any of the ac motors is the fact that by shunt field control, a wide range of operating speed, with a large number of operating points and with efficient operation, can be obtained. For this reason, shunt wound motors are often used for driving fans.

Reciprocating compressors, especially if required to start against load and to start and stop frequently, should use the compound wound type motor in order to be able to accelerate with the minimum amount of current drawn from the line.

For Induction and Synchronous Motors

$$\text{Synchronous speed} = \frac{120 \times f}{P}$$

where f = frequency in cycles/sec

P = number of poles

$$\% \text{ slip} = \frac{\text{synchronous speed} - \text{FL speed}}{\text{FL speed}} .$$

Types of motor enclosures. All of the various types of motors heretofore described can be obtained with a wide variety of mechanical enclosures to protect them from extreme operating conditions. The various types of enclosures normally used in this industry, together with their temperature rise guarantees, are as follows:

Open-type motor	40 C rise continuous
Drip-proof motor	50 C rise continuous
Splash-proof motor	50 C rise continuous
Totally enclosed motor*	55 C rise continuous
Totally enclosed fan-cooled motor*	55 C rise continuous

* Including explosion-proof motors.

Ambient temperatures. All of the above motors are guaranteed to deliver their full horsepower continuously under operating conditions where the ambient temperature does not exceed 40 C. If higher ambient temperatures are to be encountered, special motors should be applied having a temperature rise lower by the amount that the actual ambient exceeds 40 C.

Service factor. All standard open-type motors carry a service factor of 1.15 which means that they will carry 15% overload continuously without reaching dangerous temperatures, providing both the voltage and the frequency are held constant at the nameplate values.

Drip-proof, splash-proof, totally en-

General Information on Motors

Useful Formulae

$$\text{FL amps} = \frac{\text{hp} \times 746}{\text{Eff.} \times \text{P.F.} \times \sqrt{3} \times E} \qquad \text{for 3-phase motors}$$

$$\text{FL amps} = \frac{\text{hp} \times 746}{\text{Eff.} \times \text{P.F.} \times 2 \times E} \qquad \text{for 2-phase motors}$$

$$\text{FL amps} = \frac{\text{hp} \times 746}{\text{Eff.} \times \text{P.F.} \times E} \qquad \text{for single-phase motors}$$

$$\text{KVA input} = \frac{E \times I \times \sqrt{3}}{1{,}000} \qquad \text{for 3-phase motors}$$

$$\text{KVA input} = \frac{E \times I \times 2}{1{,}000} \qquad \text{for 2-phase motors}$$

$$\text{KVA input} = \frac{E \times I}{1{,}000} \qquad \text{for single-phase motors}$$

$$\text{Kw input} = \text{KVA input} \times \text{P.F.}$$

$$\text{Hp output} = \frac{\text{kw input} \times \text{eff}}{.746}$$

$$\text{Hp output} = \frac{\text{torque} \times \text{rpm}}{5{,}250}$$

closed, and totally enclosed fan-cooled motors do not carry this service factor.

Insulation. There are three different classes of insulation used in electric motors. Class A insulation, the kind used in most motors, involves the use of paper, cotton and resinous-coated conductors with similar materials for slot-cell insulation and phase insulation with the entire winding covered or impregnated with an insulating varnish. This class of insulation is good for a maximum hot spot temperature of 105 C.

Class B insulation includes asbestos, mica and glass, together with normal insulating varnish, and is good for a maximum hot spot temperature of 135 C.

Class C insulation includes glass and ceramics for which no maximum temperature limitation has been set. In the present state of the art, this type of insulation is not being supplied for standard motors.

Thermal protection for motors. In many motors, especially of the single-phase type, thermal protective devices are sometimes incorporated in the motor construction. They consist of bimetal discs or strips in intimate contact with the motor iron or windings, and incorporating in their construction contacts which open the circuit in case the motor for any reason reaches a dangerous operating temperature. In the single-phase motors, there is included in this device a heater carrying the motor line current so as to give adequate protection against stalled motor conditions.

In the polyphase type motor, the same type of protection is provided except that the contacts in the device open the control circuit, and since such motors carry currents of appreciable magnitude, no auxiliary heaters are included. This type of thermostatic protection will not protect this motor against stalled rotor conditions, but will protect it against overheating caused by overloads, too frequent starting or lack of proper ventilation.

Effect of voltage variations on motor characteristics. For the following types of ac motors—split-phase, capacitor type, repulsion start-induction run type, or the polyphase induction motor—all torques vary as the square of the voltage.

For example, a repulsion start-induction run motor attempting to start on 90% voltage would have its speed torque curve as shown in Fig. 3 reduced to 81% of the values shown in the curve. In other words, the starting torque would be reduced from 460% to 370%, the torque at transfer from repulsion to induction operation reduced from 235% to 190%, and the maximum torque reduced from 250% to 200%. The operating speed would be decreased a very slight amount, the efficiency would be decreased several per cent, but the power factor would be increased. The average motor would operate at full load with a slightly higher temperature rise. Conversely, if the voltage were 10% high, the torque values would increase 21% above the figures shown in Fig. 3.

In the case of the synchronous motors, the starting torques and pull-in torques vary as the square of the voltage, but the pull-out torque or maximum torque varies directly with the voltage.

All motors are guaranteed to operate satisfactorily and deliver their full rated horsepower at rated frequency under voltage conditions 10% above or below nameplate rating without reaching dangerous operating temperatures.

Effect of frequency variations on motor characteristics. All motors will operate satisfactorily under conditions of plus or minus 5% frequency variation, providing the voltage is held constant. They will also operate satisfactorily where the sum of the voltage and frequency variation does not exceed 10% (provided the variation in frequency does not exceed 5%) above or below normal rating as stamped on the motor nameplate.

The question often arises as to whether or not a 60-cycle motor can be used on 50 cycles. Usually standard open-type polyphase motors will operate satisfactorily, delivering their rated load with a 50 C temperature rise, instead of 40 C rise.

Single-phase, 60-cycle motors are not usually adapted to operation on a 50-cycle circuit on account of the fact that the speed at which the centrifugal device operates will be too high.

Adequacy of power supply. Because of the harmful effect which low-voltage conditions have on the speed torque curves and operating temperatures, it is imperative that all transformers and feeders serv-

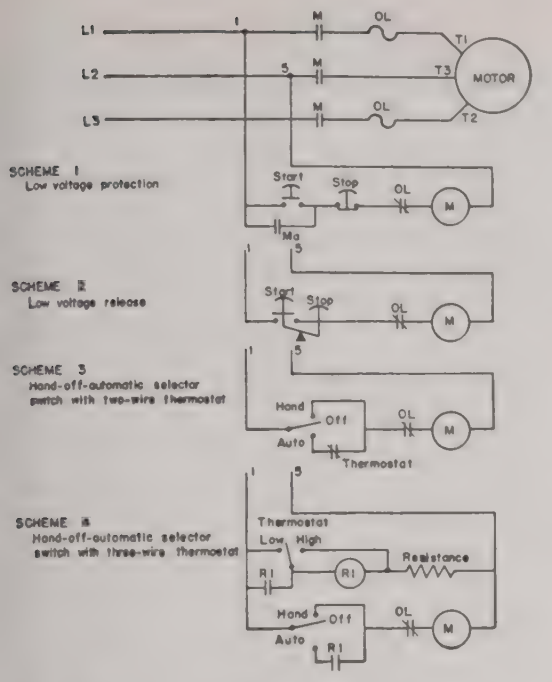


Fig. 10. Schematic Diagram of Across-the-Line Magnetic Starter

ing air conditioning and refrigerating loads be of adequate size to take care of normal running loads, and also starting currents without causing too much voltage drop. Many refrigerating and air conditioning motors are applied on the basis of a duty cycle loading, so that under certain conditions of operation, they are required to start and come up to speed against a very material overload. It is important, therefore, that voltages be held up to normal at all times. Many installations have experienced serious difficulties because of inadequate transformer and feeder capacities.

Electrical Controls for Motors

Functions of control. The principal functions of any control device are as follows:

1. Start and stop the motor either manually, magnetically by means of push buttons or magnetically by means of controlling devices such as thermostats, pressure relays, etc.
2. Protection to the operator
3. Protection to the motor
4. Control the speed
5. Limit the current drawn from the line, during starting or during running.

Manual control is provided for many single-phase motors and polyphase motors up to and including 3 hp. This control consists of a manually operated set of line contacts, provided with a set of thermal overload relays which will trip the contacts to the open position in case a sustained overload occurs.

Other manually controlled devices include a wide variety of rheostats and speed regulators for obtaining limited speed control on capacitor motors used on unit heater and unit cooler drives.

Across-the-line magnetic starters. This type of starter, more widely used than any other on polyphase induction motors, is adaptable to either manual starting and stopping by means of push-buttons, or it can be automatically controlled by means of thermostats or pressure relays. **Fig. 10** shows a schematic diagram of this starter, with push button control or with selector switch and thermostat control.

Where push buttons alone are employed, they may be wired up for either low-voltage release or low-voltage protection, both of which are shown in Fig. 10. Low-voltage protection, which means protection to the operator, utilizes a three-wire start-stop pushbutton station so arranged that in case of voltage failure the motor will not start up again on resumption of voltage until the start pushbutton is pressed. Low-voltage release involves the use of a two-wire walking beam type of pushbutton whereby, in case of power failure, the motor will stop but will start up again automatically upon resumption of voltage.

Auto transformer starters. The auto transformer type of starter is used on larger squirrel-cage motors above 20 or 25 hp where it is necessary to limit the starting current drawn from the line. A schematic diagram of this type of starter is shown in **Fig. 11.**

With this type of starter the current drawn from the line varies as the square of the voltage applied across the motor terminals. It has the disadvantage that open transition takes place upon changing from starting to running, and for this reason is sometimes considered objectionable by the power company on whose lines it is operating.

Resistance type (increment type). This

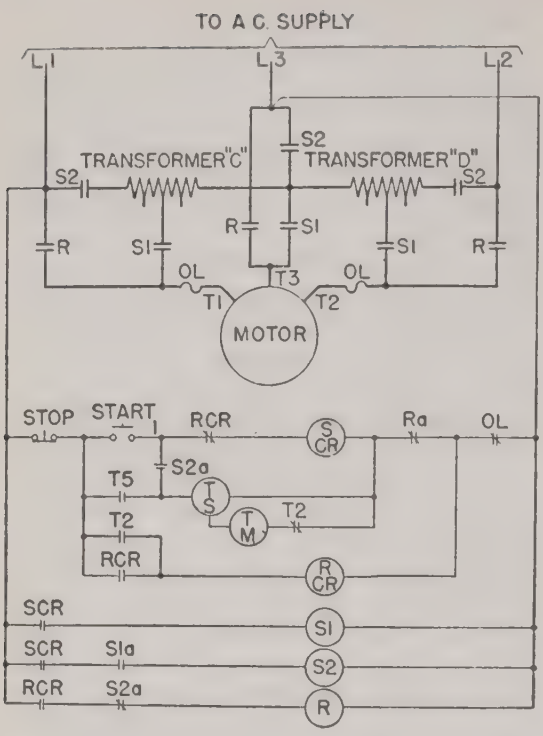


Fig. 11. Typical Schematic Diagram of Magnetic Auto-Transformer Type Starter for Squirrel Motors

type of starter for the squirrel-cage induction motor is often used on the 208-volt network systems in the larger cities. It employs a line contactor, one or more steps of resistance in the leads going to the motor, and one or more accelerating contactors to short out this resistance in bringing the motor up to speed. The advantage of this type of starter is that it does not open the circuit to the motor at any time during the starting operation, but it has the disadvantage that the current drawn from the line is directly proportional to the voltage across the motor terminals. Where two or more steps of resistance are employed, as a result of power company limitations, these limitations may be such that the motor may not start on the first step of the control, which condition should not cause any undue alarm.

The power company regulations may be such that the full locked rotor current will not cause any disturbance on the line, providing this value of current is reached by increments occurring not oftener than once every one and a half or two seconds. A schematic diagram for this type of starter is shown in **Fig. 12.**

Wound rotor motor control. This type of control usually employs a combination of an across-the-line starter in the primary circuit to the motor in combination with a manually-operated or motor-operated drum controller and resistance controlling the secondary winding of the motor to give the desired speed control.

Synchronous motor starters. Synchronous motor starters may be of the full voltage or reduced voltage type as required. If of the reduced voltage type they may incorporate either resistance, reactance or auto-transformer type starting. Also, they may be either of the semi-magnetic or full magnetic type.

In the semi-magnetic type, manually operated breakers are used to apply reduced voltage or full voltage to the motor stator windings, after which at the desired time or speed direct current is automatically applied to the field. In the full magnetic type, the entire operation is automatically handled in the correct sequence merely by pressing a start pushbutton.

Various sequence and extra protective relays, not normally furnished with the starters used for squirrel-cage motors, are required in the synchronous motor starter.

Maintenance of Electrical Equipment

When a new motor is first received on the job, a thorough **inspection** should be made to insure that it has not been damaged in shipment, and that it is in good working condition. If possible, the shaft should be turned by hand to see that it rotates freely.

Bearings should be inspected before power is applied to see that they are provided with adequate lubricant. Fractional hp motors are normally provided with waste packed sleeve bearings, and are usually supplied with enough oil to last three or four months in operation. An inspection of the bearings will reveal whether or not oil should be added. This type of motor should use a light grade of good machine oil (S.A.E. #10 W).

If the motor is provided with ring oiled sleeve bearings, oil must be added to the proper level before it is placed in service.

If the motors are provided with ball bearings, they will have sufficient lubricant for at least three or four months' service. Some motors are provided with pre-lubricated sealed ball bearings which will only require cleaning and re-packing once every five years.

Cleanliness is of utmost importance in the satisfactory operation of any motor. Periodically, depending upon the amount of dirt in the atmosphere, motor windings should be thoroughly cleaned by blowing them out with air, using care that the blast of air from a compressed air hose does not come too close to the motor windings and cause damage to the insulation. Every three to five years it is advisable to clean the motor windings thoroughly and give them another coat of insulating varnish.

When new motors are received, they should be stored in a warm, dry place until ready to be used. If, for any reason, they have been exposed to undue moisture conditions, they should be thoroughly dried out before full voltage is applied to them.

The most effective method for **drying** the larger higher voltage motors is to pass current through the windings, using a voltage low enough to be safe when the windings are moist. For 2,300-volt motors, 220 volts will usually be satisfactory for circulating drying out current. Thermometers should be placed on the windings to see that they are being heated uniformly, and that the temperature does not exceed 90 C. For medium size and lower voltage motors, drying out may be accomplished by enclosing the machine in a temporary housing of sheet metal, and placing space heaters about the machine.

Where **brushes** are employed on direct-current, slip-ring or synchronous motors, the brushes should be inspected periodically to see that they are free in the holders, and that they are not worn excessively. The brush riding surfaces should be inspected, and if it is found that they are picking up metal from the rings or from

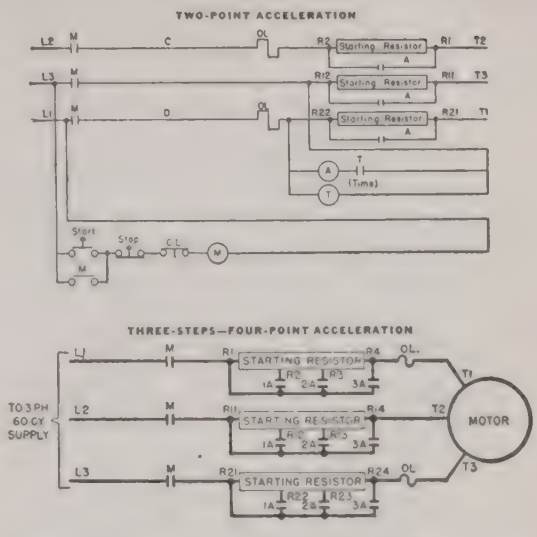


Fig. 12. Schematic Wiring Diagrams of Resistance Type Magnetic Reduced Voltage Starters for Squirrel-Cage Induction Motors

the commutator, the metal particles should be removed, and the brush face resurfaced. Brush tensions on the average machine should be on the order of 2 to 2.5 psi of projected brush area.

Air gaps in the average induction motor vary from 0.012 in. in the smaller motors up to 0.038 in. in the larger ratings. It is important that the air gaps be checked periodically to insure that bearing wear has not permitted the rotor to strike the stator. When the air gap on one side of the motor reaches a point where it is half or two-thirds that on the other side, it is an indication that the bearings are probably worn, and that they should be replaced.

All **control apparatus** should be inspected thoroughly each time the motor is given an inspection. If contacts are badly pitted they should be dressed up, and if badly worn they should be replaced. All control connections should be checked up periodically to see that they are tight. Most control contactors do not require any lubrication and should, therefore, not be oiled.

21. COOLING TOWERS

Introduction

THE general designation, **cooling tower,** has been applied to a structure designed to bring about the cooling of water by evaporation and heat transfer as the water is circulated through the structure in direct contact with a current of atmospheric air.

Cooling towers are widely employed in refrigeration, power, and industrial processing plants where the cycle of operation and conditions of the installation call for the dissipation of heat from cooling water to the surroundings, together with the saving and continual re-use of the main body of cooling water with only a small makeup required. Spray ponds or large cooling ponds are less frequently employed for the same purpose, depending upon particular features of the installation and suitable surroundings.

Innumerable structural variations in cooling towers have been considered by designers in efforts to produce effective cooling through close interaction of water and air, at the same time meeting all practical requirements of durability and economy in construction and operation. Design trends which have become dominant have emphasized sound engineering economics throughout, while making best use of recent research information.

This chapter will deal with: (1) physical principles of evaporative cooling; (2) design procedures from the standpoint of the physical principles; (3) generally available design data for mechanical-draft towers; (4) empirical design procedures for atmospheric towers; (5) spray pond performance; and (6) economic aspects of design and selection.

Physical Principles of Evaporative Cooling

A. Rate Equations for Heat and Mass Transfer

The fundamental physical mechanisms of evaporative cooling reside in the molecular rates of transfer of heat (sensible heat) and mass (latent heat) across the interfacial boundary surface between the water and the air. The true picture of what actually occurs in the processes at the boundary is hidden in microscopic phenomena which are as yet incompletely understood and which cannot be expressed in direct, useful terms. For engineering purposes, the transfer-rate equations are conveniently expressed in terms of measurable overall transfer "potentials" and experimental coefficients designated as **unit conductances.** The magnitudes of these unit conductances are design data expressing the rate of heat or mass transfer per unit of interfacial area and per unit of the "potential" acting to cause transfer across that portion of the boundary region being considered.[1]

The cooling tower, in particular, is generally treated in terms of an unknown interfacial "surface area per unit of tower volume," to be experimentally determined along with the unit conductance. In this way, the basic transfer rate equations are written for a small unit of tower **volume,** and the product of the unit conductance and the corresponding surface area per unit volume is treated as a variable for design calculations.

The various transfer-rate equations are expressed as listed below.

1. Rate of (sensible) heat transfer between the main body of air at temperature t and the interfacial water surface at temperature t_i,

$$dq_s = h_g(adV)(t_i - t) \qquad (1)$$

2. Rate of heat transfer between the interfacial water surface at temperature t_i and the main body of liquid at temperature*, ν,

$$dq_w = h_L(adV)(\nu - t_i) \qquad (2)$$

3. Rate of mass transfer between the main body of air at specific humidity H and the interfacial surface at specific humidity H_i (the saturated air-vapor state at temperature t_i),

$$dm_i = k_g(adV)(H_i - H) \qquad (3)$$

4. The rate of heat transfer to the interface in equivalence to the rate of latent-

* For adiabatic humidification, $\nu = t_i$ = thermodynamic wet-bulb temperature.

heat conversion is obtained upon multiplying Equation (3) by the latent heat at the interface temperature, r_i,

$$dq_L = r_i k_g (adV)(H_i - H) \qquad (4)$$

According to the prevailing engineering conception of the process, no resistance to **mass transfer** exists between the main body of the liquid and the interface.

B. Steady-State Heat and Mass Balances

Consider a small section of tower volume as sketched in **Fig. 1.** The section there shown illustrates counterflow of water and air, which is the most common case. (Parallel flow and crossflow may be dealt with similarly.) The usual basis of postulating steady-state conditions, with no heat flow to or from the surroundings of the tower, will be adopted throughout the analysis.

1. The mass balance equation. Since the rate of mass transfer from the water stream must equal the rate of moisture gain by the air stream,

$$dL = GdH \qquad (5)$$

2. The heat balance equation. Since the entire heat removal from the water stream must be taken up by the air stream, and if the very slight change in L due to evaporation is neglected,

$$Lc_{p_w} dv = Gdh \qquad (6)$$

The air-stream enthalpy may be expressed as $h = c_{p_a}(t - t_o) + H(r_o + c_{p_v}(t - t_o))$. Thus, the differential change in the air-stream enthalpy, dh, becomes, with $s = c_{p_a} + Hc_{p_v}$,

$$dh = sdt + (r_o + c_{p_g}(t - t_o))dH, \qquad (7)$$

where the first right-hand term represents **sensible heat,** and the second **latent heat.**

Making the above substitution for dh, Equation (6) becomes

$$Lc_{p_w} dv = Gsdt + G(r_o + c_{p_g}(t - t_o))dH \qquad (8)$$

Design Procedures for Mechanical-Draft Cooling Towers*

Combinations of the equations given in the preceding sections form the basis of all rational design procedures. The physical data needed to carry out a design are those establishing the unit conductances and areas per unit volume in the transfer-rate equations. It is very important that the steps in developing practical design procedures be clearly set forth, for a fundamental understanding is essential to making the most of present and future research data.

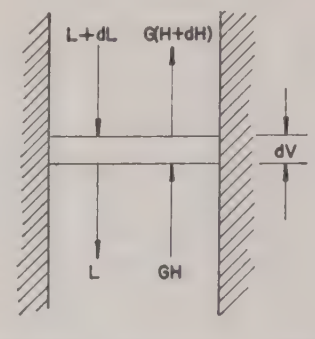


Fig. 1. Elementary Evaporative Cooling System

A. Fundamental Equations

1. Inter-relation of the **sensible heat** and **mass transfer rates** on the basis of the Lewis relationship.

From Equations (1) and (8) a sensible heat relation may be written as

$$Gsdt = h_g(adV)(t_i - t) \qquad (9)$$

From Equations (3) and (5) a mass exchange relation may be written as

$$GdH = k_g(adV)(H_i - H) \qquad (10)$$

Upon dividing Equation (10) by Equation (9), there is obtained the significant expression,

$$\frac{dH}{dt} = \frac{(h_g)}{(k_g s)} \frac{(H_i - H)}{(t_i - t)} \qquad (11)$$

Once the magnitude of the ratio $(h_g/k_g s)$ is established, Equation (11) allows either the temperature or the humidity difference to be expressed in terms of the other (on the basis of t_i being known).

The so-called Lewis relationship states that, for all ordinary accuracy,

$$\frac{h_g}{k_g s} \approx 1 \qquad (12)$$

for an **air-water evaporative cooling system.**[2] On this basis, Equation (11) attains maximum convenience.

2. The overall conductances and working approximations.

Rigorous design applications in terms of the basic Equations (1) through (8) are seriously restricted by the lack of an adequate extent of data on each individual

* A different approach to the design problem, developed in terms of the **sigma function,** is described by Simons.[12b]

unit conductance. This lack of fundamental information has been allowed to prevail without becoming a complete stumbling block for the very practical reason that **working approximations** are possible which circumvent the need for such detailed data, and yet which do not introduce design errors of a serious magnitude in comparison to the complex, unsteady, and often erratic behavior of the variables influencing cooling tower operation in a field installation.[3] The situation in design procedures is one of those in which an engineering balance between rigorous principles and the practical art will decide between success and failure in the ultimate economic valuation of the results.

The customary procedure is to pass over the interface conditions given in Equations (1), (2), and (3) and to write an **overall** sensible heat-transfer equation and an **overall** mass-transfer equation as follows:

$$dq_s = U_g(adV)(\nu - t) = Gsdt \quad (13)$$

$$dm = K_g(adV)(H_\nu - H) = GdH \quad (14)$$

Following Equations (13) and (14), the Lewis relationship, Equation (12), does not require in itself that the ratio

$$\frac{U_g}{K_g s}$$

should be unity.* Koch[4] finds that 0.9 is a representative value for cooling towers. On the other hand, London, Mason, and Boelter,[5] Simpson and Sherwood,[19] and others suggest unity as a good approximation.

For most practical applications, the controversy in this regard becomes unimportant once it is recognized that U_g and K_g are **overall** conductances, and not fundamental entities. Pending further research data, the recommendation of this chapter is that the assumption be made, for purposes of **practical convenience** and until supplemented by more reliable information, that

$$U_g \cong K_g s, \quad \text{or} \quad \frac{U_g}{K_g s} \cong 1 \quad (15)$$

* Ideal adiabatic humidification is a special case which offers an exception because the overall and air-film conductances then are the same.

3. The practical design equation.

The next step towards obtaining a simple and useful design equation is the substitution of Equation (13) and (14) into Equation (8) and the simultaneous use of Equation (15). This yields

$$Lc_{p_w}d\nu = K_g adV[s(\nu - t) + (r_o + c_{p_v}(t - t_o))(H_\nu - H)] \quad (16)$$

Upon introduction of the expression for the enthalpy of a water-vapor air mixture, $h = st - c_{p_a}t_o + H(r_o - c_{p_v}t_o)$, Equation (16) becomes

$$LC_{p_w}d\nu = K_g adV[(h_\nu - h) + c_{p_v}t(H_\nu - H)] \quad (17)$$

At this juncture, great simplification results from adopting Merkel's procedure[6,7] and neglecting the second term in square brackets in comparison to the first. This leads to an approximate design equation which is very simple and usually of adequate precision,

$$\frac{c_{p_w}d\nu}{(h_\nu - h)} \approx \frac{K_g adV}{L} \quad (18)$$

The left side of this equation is composed of thermodynamic variables only, and the factors on the right side depend upon the tower design and conditions of operation. The integrated form is the **recommended basis** of **performance analysis** and **practical design**,

$$\int_{\nu_2}^{\nu_1} \frac{c_{p_w}d\nu}{(h_\nu - h)} = \frac{K_g aV}{L} \quad (19)$$

It is important to note that the product $(K_g a)$ in Equation (19) has been taken at a mean constant value for integration, such that the equation is satisfied. The cumulative effect of all approximations will therefore appear in experimental magnitudes of $(K_g a)$ when analyzing performance according to this method.

B. Evaluation of the Practical Design Equation

An excellent discussion of the application of Equation (19) to design problems has been given by Lichtenstein.[7]

It may be noted that approximations are possible to simplify the integration of the equation (such as an assumed linear relation between ν and h_ν, which leads to the logarithmic-mean enthalpy potential), but the results are then of limited value and

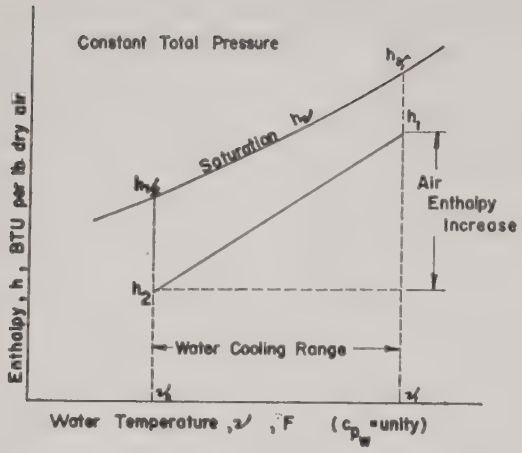


Fig. 2. Counterflow Cooling Tower Operation Represented on an Air-Vapor Mixture Enthalpy vs. Water Temperature Diagram

apply only to a narrow range of design conditions.

It is recommended that a procedure of solution be followed which will introduce no further approximations beyond the derivation of Equation (19). The steps in such a procedure will be noted in detail for reference.

1. Graphical evaluation of

$$\int_{\nu_2}^{\nu_1} \frac{c_{p_w} d\nu}{(h_\nu - h)}$$

Magnitudes of the enthalpy for water vapor-air mixtures may be obtained from thermodynamic tables.* Customary practice is to carry through design calculations for the standard barometric pressure of 29.92 in. of mercury.

Consider a diagram, as shown in **Fig. 2,** in which the enthalpy of the water vapor-air mixture is plotted versus the water temperature, ν. It is to be understood that this latter coordinate actually represents the product $c_{p_w}\nu$ under the practical simplification that c_{p_w} is unity for all ordinary purposes. The curve of saturation enthalpy is given directly by the thermodynamic data.

From the integration of Equation (6), neglecting the very slight change in L due to evaporation, it can be established that the slope of any straight line on Fig. 2 is

$$\frac{h_1 - h_2}{c_{p_w}(\nu_1 - \nu_2)} = \frac{L}{G} \tag{20}$$

From the illustrative case shown in Fig. 2, the following conditions and relations may be summarized concerning the operating variables involved:

(a) $(\nu_1 - \nu_2)$ = water cooling range.

(b) $(h_1 - h_2)$ = enthalpy increase of the counterflow air stream.

(c) h_2 is established by the initial air state; in particular, there are design wet-bulb and dry-bulb temperatures corresponding to h_2.

(d) $(h_{\nu 2} - h_2)$ is the (counterflow) water-exit enthalpy "approach" which indicates how closely the process comes to the limiting condition of $(h_{\nu 2} - h_2) = 0$.

(e) (L/G) represents the slope of the operating line.

(f) $(h_\nu - h)$ at any point, representing the energy-transfer "potential," is indicated by the vertical distance between the saturation line and the operating line.

The **approach to the wet bulb,** being the difference between the water-exit temperature and the air-inlet wet-bulb temperature, is widely employed in field practice to indicate tower performance. The enthalpy difference $(h_{\nu 2} - h_2)$ does not, properly speaking, uniquely establish the "wet-bulb approach" under all circumstances. The relation between the enthalpy and wet-bulb approaches may be derived from the equation

$$(h_{\nu 2} - h_2) = c_{p_a}(\nu_2 - t_2) + (H_{\nu 2} - H_2)(r_o - c_{p_v} t_o) + c_{p_v}(H_{\nu 2} - H_2 t_2) \tag{21}$$

The specific humidity H_2 is a function of the wet and dry-bulb temperatures, while $H_{\nu 2}$ is established once ν_2 is known (for some specified pressure). The enthalpy approach will determine the wet-bulb approach uniquely only when ν_2 and t_2 are otherwise specified. Occasionally this restriction may be set aside to form an approximation, for limited practical convenience, that the wet-bulb and enthalpy approaches the inter-dependent, but this must always be regarded with caution.

2. Carrying out the performance integration. The integration of the thermodynamic term of Equation (19) must be

* Refer to Chapter 10.

carried out for a complete sequence of operating conditions covering the entire useful range. This integration is easily accomplished graphically from a plotting of

$$\frac{1}{(h_\nu - h)} \text{ vs. } \nu$$

In considering the choice of parameters for the integrations, the initial air state may be generally regarded as known, and so some point may be established on Fig. 2 with coordinates (ν_2, h_2). A series of different values of ν_2 may then be chosen for any one h_2. Starting from any point (ν_2, h_2), operating lines may be drawn with various slopes, from horizontal (case of $L=0$ or $G=\infty$) to a line intersecting the saturation-enthalpy curve at some water-inlet temperature, ν_1. This latter condition represents an ideal limiting case of air leaving saturated at the water entrance temperature. The upper limit to L/G thus depends on ν_1, which is to be chosen successively over points in the range of interest.

Each integration will establish a magnitude of the characteristic factor ($K_g aV/L$). A summary of the results can be made by plotting the magnitudes of ($K_g aV/L$) versus (L/G) in a set of curves covering the ranges of variation for the other performance factors ($h_{\nu_2} - h_2$), ($\nu_1 - \nu_2$), initial air state or h_2, and either ν_1 or ν_2. A single barometric pressure is understood to apply for all calculations and the pressure within the tower is assumed to depart by a negligible amount from the barometric pressure; otherwise the calculations must be extended to cover other pressures.

The general shape of the performance curves when plotted with ($h\nu_2 - h_2$) as a parameter and the other variables constant is shown in **Fig. 3.**

C. Performance Selection of a Cooling Tower

The first step in designing a cooling tower is that of establishing the tower performance in terms which will lead to the determination of the overall dimensions and the air and water rates. This reduces in essence to the requirement of obtaining a cooling tower which will have a magnitude of the characteristic ($K_g aV/L$) such that the desired water cooling range is attained under the imposed magnitudes of the independent operating variables.

The proper extent of basic experimental data must be available for each type of tower construction to be considered in a performance study. Simultaneously with

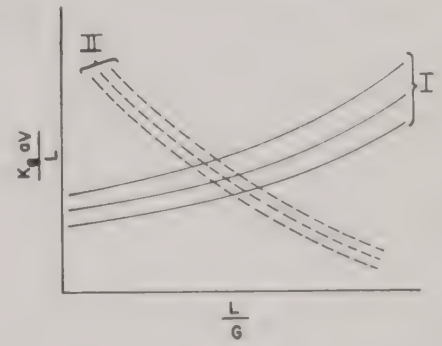


Fig. 3. Graphical Performance Selection of a Mechanical-Draft Cooling Tower

Curves in group I are established by a thermodynamic integration of Eq. 19. The form sketched illustrates lines of constant enthalpy approach.

Curves in group II are established by experimental data for the tower construction being considered.

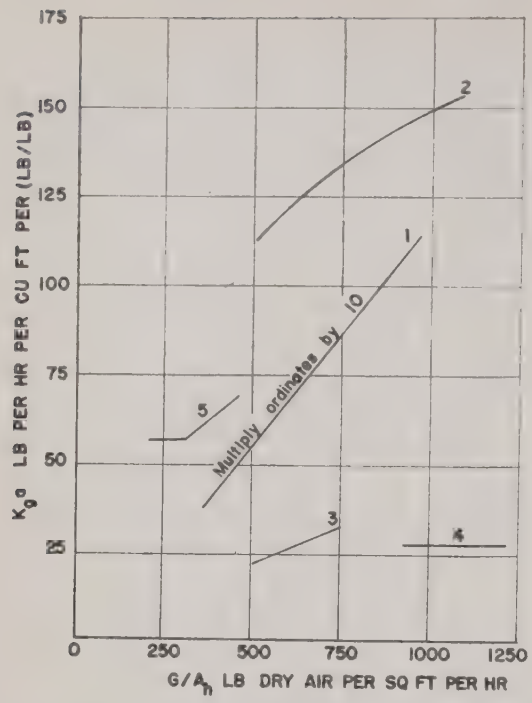
Intersections of corresponding curves from group I and group II determine the operating points attainable.

the thermodynamic-function relations defined by Equation (19) and plotted in Fig. 3, there must be available experimental data on the conductance ($K_g a$) establishing **experimental design functions** which need to be expressed as a series of curves of

$$(K_g aV/L) \text{ vs } (L/G)$$

The general nature of these latter curves is also indicated in Fig. 3. The simultaneous **solution** (intersection point on Fig. 3) of the **experimental relations** and the **thermodynamic functions** of Equation (19) will establish the operating point of a cooling tower, according to the magnitude of the independent variables belonging to the intersecting curves. A complete performance survey may be carried out very rapidly with the aid of Fig. 3, merely by reading these intersection points.

Following determination of the operating magnitude of ($K_g aV/L$) required for some particular application problem, it is necessary to select or establish the indi-



ASHVE Guide, 1946

Fig. 4. Unit Conductance Data for Various Packing Constructions

Constructions described on page 365.

vidual quantities L, G, and V. The tower volume V is broken down into the height and cross section according to additional requirements. A single set of magnitudes for $(K_g a V/L)$ and L/G does not uniquely establish a tower, but it does establish a necessary inter-relation between L, G, and V.

Design Data for Mechanical-Draft Packed and Spray Cooling Towers

Available design data are quite meagre. Much more published research is needed before it will be possible to design all types of cooling towers properly without the assistance of experimental development work.

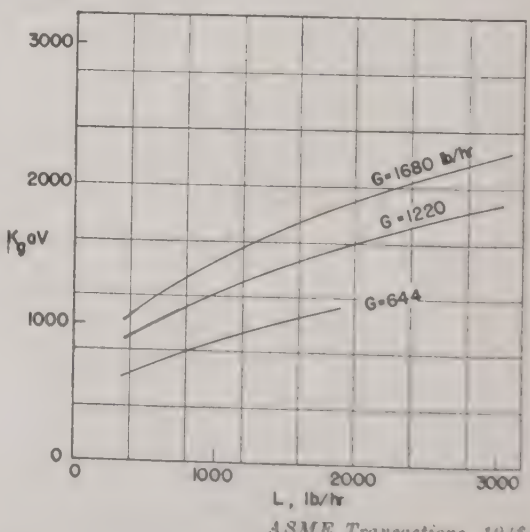
Whenever experimental development tests are carried out, it is recommended that the suggestions of this chapter be followed in correlating magnitudes of the characteristic $(K_g a V/L)$, and that all such experience be reported in the literature.

A. Conductance Data for Packed Towers

Fig. 4 presents approximate data for various packings which will serve for estimating purposes. The curves should not be considered as generally applicable to other packing structures than those for which the original data were obtained. Neither should they be used for high initial water temperatures (say above 110 F) without recognized caution, for mean values of $K_g a$ may vary with the initial water temperature.

The data of Fig. 4 may be noted as presenting $K_g a$ to be independent of the liquid rate. This condition would indicate that: (1) the contribution of the jet and drop surfaces in the spaces between the packing appears as an effect independent of the water rate; and (2) there is no appreciable variation in the extent to which the packing is wetted with water rate. These conditions would not necessarily hold for operation outside of the range of the data reported.

Fig. 5 presents data[7] on a slat-packed tower 6 ft × 6 ft in cross section with a splash-trough distribution deck. The slats were $\frac{3}{8}$ in. × 2 in., spaced $1\frac{1}{8}$ in. on centers. The packing consisted of ten decks on 15-in. center spacing. An effect of both L and G upon $K_g a$ is demonstrated; however,



ASME Transactions, 1945

Fig. 5. Conductances for a Slat-Packed Tower

Supplementary Information for Fig. 4

Curve No.	Literature ref	Tower dimensions	Water distribution arrangement	Packing details	Operating conditions for data reported
1	6	Cylindrical packed section, 9.84 in. diam. by 20.1 in. high.	Single spray	Packing composed of "dumped-in" sheet metal rings of dimensions sketched. 0.36 lb of rings have 1 sq ft of surface. Total ring surface in packed section is 107.6 sq ft.	Counterflow, forced draft. Total water inflow 331 lb/hr at inlet temp of 86 F.
2	5	Gross tower cross-section 7.6 sq ft. Net tower cross section 5.5 sq ft, 22 decks.	3-in. I.D. pipe distributing main crosswise to slats. Diametrically opposite holes in pipe distribute water to grooved-top flats which further spread the water.	Ovate slats in staggered rows. Slat length 2.75 ft. Spacing of slats 2½ in. on centers. 13 slats per deck. 22 decks total. Free fall of water drops per deck 0.3 ft.	Counterflow, forced draft. Water rates from 935 to 1,650 lb/(hr)(sq ft of gross cross-section). Water inlet temp 82 to 103 F.
3	20	Packed section 6.56 × 6.56 ft. in cross-section, 16.1 ft. high.	Header and overflow troughs with notched sides to form 172 separate streams to the packing.	Each deck comprised 14 slats at 45° inclination as sketched. Spacing of decks 13.8 in. on centers. 14 decks total.	Counterflow, natural draft chimney tower. Water rates 480 to 1,110 lb/(hr)(sq ft gross cross-section)
4	20	Same tower as No. 3.	Spray nozzles placed in tower basin.	No. 3 with packing removed.	Same as No. 3.
5	21	Quarter-scale model tower at Univ. of California.	Low-pressure sprays.	Slat-deck atmospheric tower, Slats ¾ in. × ⅛ in. Spacing 8 in. between decks.	Operates counterflow, natural draft, with negligible wind; cross-flow with appreciable wind.

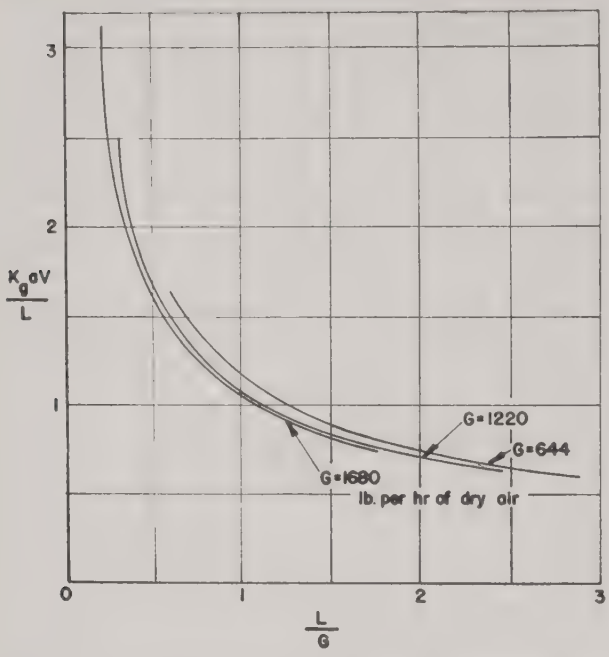


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Fig. 6. Tower Characteristics from the Data of Fig. 4

when plotting the same data as $K_g aV/L$ vs. L/G, the result is very nearly a single line. **Fig. 6** shows this.

Test data have been reported[19] for small spray-distribution packed towers of the type shown in **Fig. 6a.** Details of the internal constructions investigated are summarized in **Table 1.** Conductance data determined are presented in **Fig. 6b.**

Fig. 6c shows the data of Fig. 6b when the method of presentation is taken as a plotting of $K_g aV/L$ vs. L/G according to the discussion in this chapter. The points come very close to defining a single line, similarly to Fig. 6. (In Ref. 19 it is commented that the packing of their test towers may have been incompletely wetted in some instances at low values of L/G and, if so, this would slightly distort magnitudes of $K_g a$ deduced from such runs.)

It should be noted that Figs. 6b and 6c refer specifically to an inlet water temperature of 110 F. Experience with the packings tested showed that the overall mean conductance, $K_g a$, decreased slightly with increasing water inlet temperature. **Fig. 6d** shows this temperature effect for tower R-1 (Table 1, Figs. 6b, 6c). Further systematic data of this nature are desirable for various packings and all combinations of water and air rates.

Pressure-drop data for the air stream passing through the packing are available for four of the towers listed in Table 1, see **Fig. 6e**[19].

These brief presentations indicate the extent of data presently available for packed cooling towers. A few additional data are available for similar absorption systems,[1] and these may be found of value in guiding design thoughts.

B. Conductance Data for Spray Towers

The problem of determining general data on $K_g a$ for sprays has seen only limited progress. Complications introducing experimental difficulties are: (1) the factors K_g and a are interdependently related in a complex manner for sprays, for the size and distribution of the drops are strongly influenced by spray nozzle design and arrangement, spray pressure, and air and water flow rates, besides varying markedly from point to point throughout the spray-chamber volume due to the considerable differences encountered in spray-

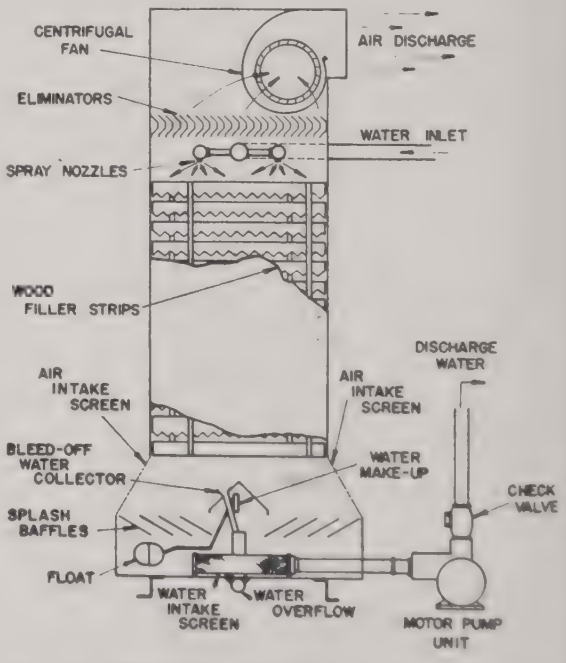

Fig. 6a. Schematic Diagram of Small Spray-Distribution Packed Cooling Tower

Table 1. Physical Data on Experimental Constructions Studied by Simpson and Sherwood[19]

Tower Designation	W-1	W-2	R-1	R-2	M-1	M-2
Tower width, in. (internal)	$41\frac{1}{4}$	$41\frac{1}{4}$	$41\frac{5}{8}$	$41\frac{5}{8}$	$41\frac{5}{8}$	$41\frac{5}{8}$
Tower depth, in. (internal)	$15\frac{5}{8}$	$15\frac{7}{8}$	$23\frac{7}{8}$	$23\frac{7}{8}$	$23\frac{7}{8}$	$23\frac{7}{8}$
Tower height, in.	$77\frac{3}{16}$	$77\frac{3}{16}$	84	74	84	84
Packed height, in.	36	36	$41\frac{3}{8}$	$41\frac{3}{8}$	$41\frac{3}{8}$	$41\frac{3}{8}$
Volume occupied by spray space and packing, cu ft	185	185	318	318	318	318
Packing material	[a]	[a]	[b]	[c]	[c]	[c]
No sprays	12	12	18	18	18	18
Packing spacing						
Hor. centers, in.	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$
Vert. centers, in.	—	—	$2\frac{5}{8}$	[d]	—	—
Packing thickness, in.	—	—	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{8}$
No vertical sheets or assemblies	83	83	61	61	61	77
Total no slats	—	—	1008	854	—	—

[a] ½-in. mesh hardware cloth, in sheets $15\frac{3}{4}$ in. by 36 in.
[b] Redwood slats $23\frac{1}{4}$ in. long by 2 in. high, with bottom edge serrated.
[c] Masonite sheets $23\frac{1}{4}$ in. by $41\frac{1}{8}$ in. by $\frac{1}{8}$ in.
[d] Varying from $3\frac{1}{8}$ in. at the top to $2\frac{1}{8}$ in. at the bottom.

drop distribution. (2) The water surface in the usual spray tower is a combination of drop surface and the surface of wetted walls or structural members, and the different kinds of surfaces follow different laws for the conductance data, which requires a separation of the effects for design analyses.

Data available[8] for order-of-magnitude estimates on small counterflow spray towers are shown in **Figs. 7, 8,** and **9.** These data show the overall **spray-drop** and **wall-surface** conductances both separately and in combination. An empirical equation may be written for the **combined spray** and **wall conductance** in the form:

$$K_g a = 5.89 \frac{L^{0.76} G^{0.17}}{T_2^{1.09}}, \quad \frac{\text{lb}}{(\text{sq ft})(\text{hr})(\text{lb/lb})}. \quad (22)$$

The tower employed to obtain the above relation was a counterflow, forced-draft metal spray tower, 7 ft high and with a 2-ft × 3-ft cross-section spray chamber. Six swirl-chamber spray nozzles were employed in the test section, each nozzle yielding a fine-drop, wide-angle, hollow-cone spray. Nozzles yielding a flat, large-drop spray produced only 50 to 70% of the conductances from Equation (22).

In further regard to the data cited, it is to be noted that the tests yielded $U_g / K_g s$

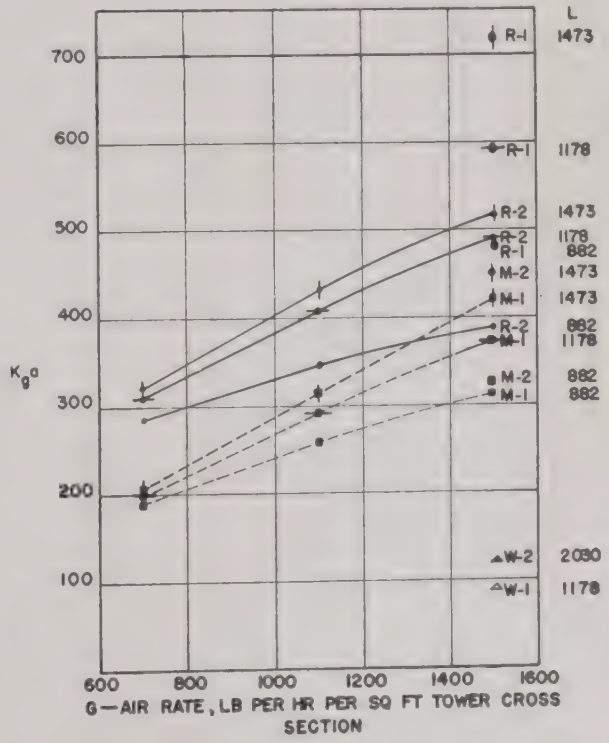

Fig. 6b. Unit Conductances for Small Spray-Distribution Packed Towers at an Inlet Water Temperature of 110 F (see Table 1 for packing data)

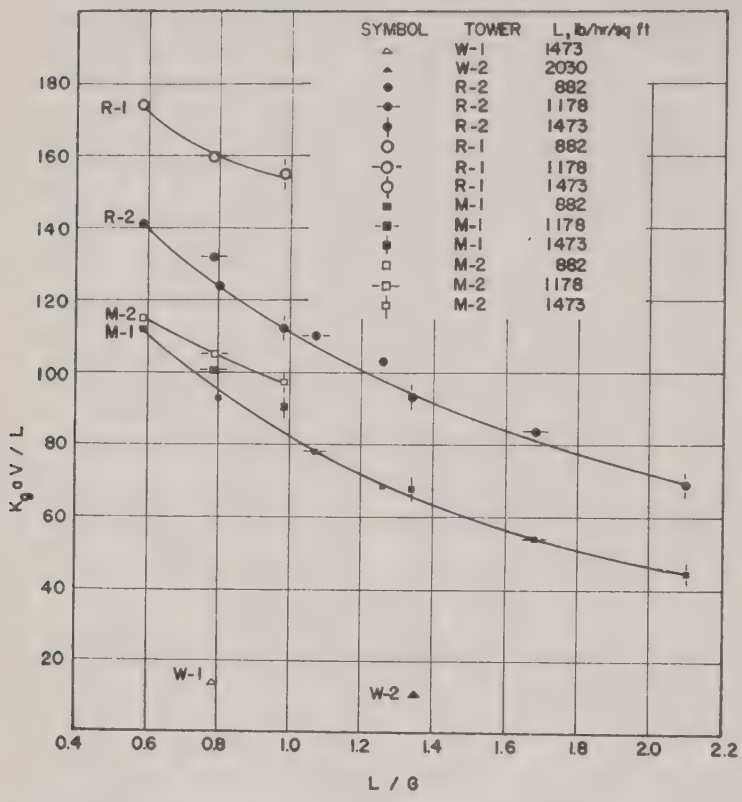

Fig. 6c. Conductance Data of Fig. 6b Presented as ($K_g aV/L$)

vaporization of small drops before completing their passage through the tower, which caused a shift in the initial balance between sensible and latent heat transfer.

Further data on the overall mean unit conductance for different nozzle conditions in the same test tower are available.[9] These latter results illustrate the effects of changing: (1) the number of nozzles; (2) the location (height) of the nozzles; and (3) the water pressure at the nozzles. One series of test runs was also made to study the effect of adding two No. 4 mesh, and two No. 8 mesh screens, respectively, to the spray chamber while maintaining fixed air and water flow rates; an increased overall conductance was found to result.

Designers may be interested in studying related data on spray-type absorption equipment using other fluids. Reference 1 presents a good summary and discussion.

<1, and this is to be kept in mind when judging the results. The primary reason for this effect was believed to be the complete

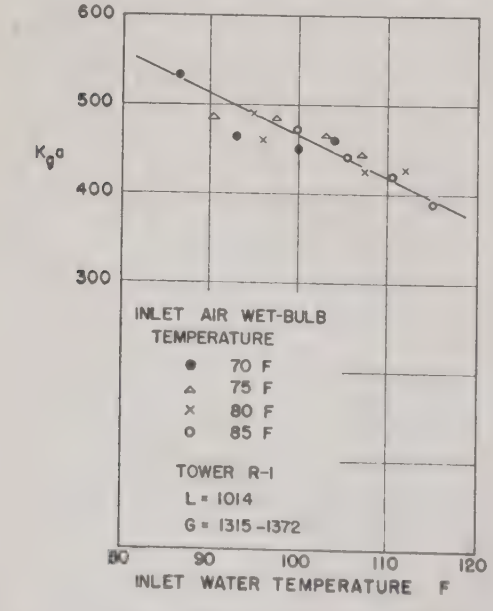

Fig. 6d. Effect of Water Inlet Temperature on ($K_g a$) for Tower R—1[19]

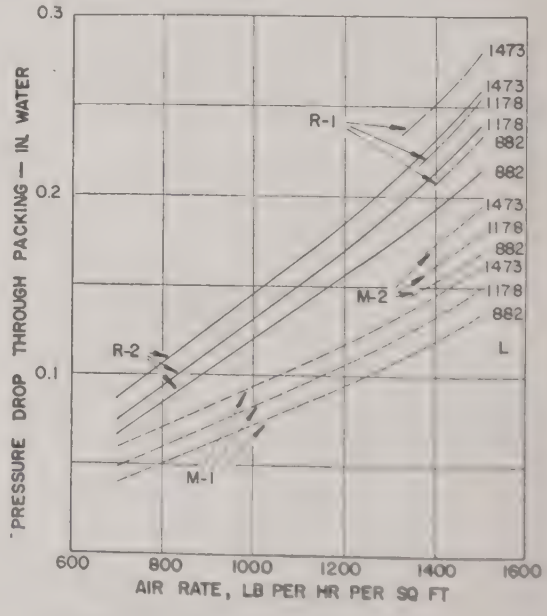

Fig. 6e. Pressure-Drop Data for the Packed Towers of Table 1

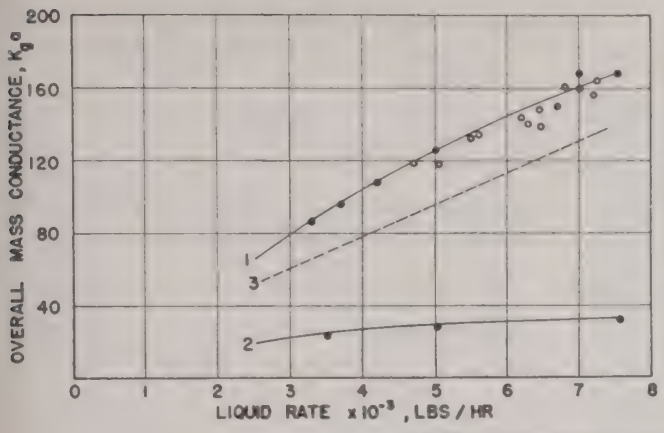

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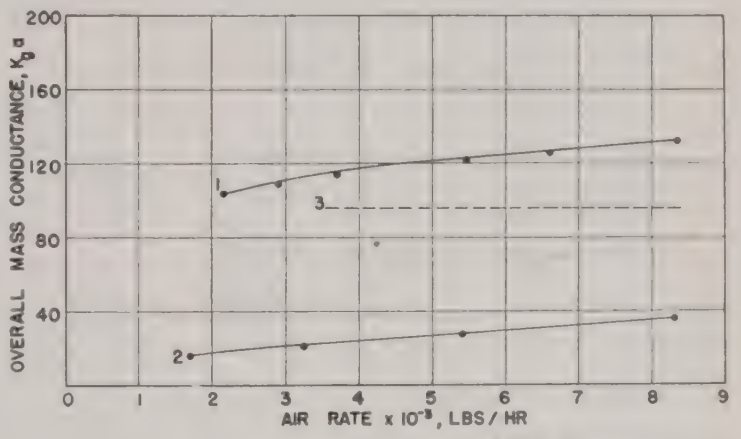
Fig. 7. Overall Mass Conductance $K_g a$, lb/ft³hr(lb/lb) Versus Liquid Rate lb/hr. Air Rate Equals Approximately 5,000 lb/hr

1. Conductance for spray and walls combined
2. Conductance for walls alone
3. Conductance for sprays alone (obtained by subtraction)

Fig. 8. Overall Mass Conductance, $K_g a$, lb/ft³hr(lb/lb) Versus Air Rate, lb/hr. Liquid Rate Equals Approximately 5,000 lb/hr

1. Conductance for sprays and walls combined
2. Conductance for walls alone
3. Conductance for sprays alone (obtained by subtraction)



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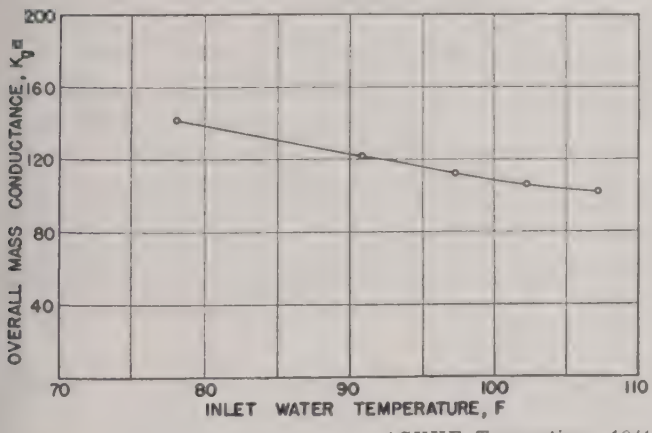

ASHVE Transactions, 1941

Fig. 9. Overall Mass Conductance, $K_g a$, lb/ft³hr(lb/lb) Versus Inlet Water Temperature, F

Liquid rate equals approximately 5,000 lb/hr
Air rate equals approximately 5,000 lb/hr

Empirical Design Methods for Atmospheric Cooling Towers

Overall integrated forms of the fundamental evaporative cooling equations as developed in this chapter must not be too rigorously applied to atmospheric cooling towers. The open and variable nature of the air-flow circuit makes any analysis based upon a single air stream applicable only as a rough idealization.*

Extensive data on the distribution of air flow in atmospheric towers are badly needed. Rational designs could then be undertaken in terms of conductances for

* Curve 5 of Fig. 4 indicates data from a limited attempt in this direction.

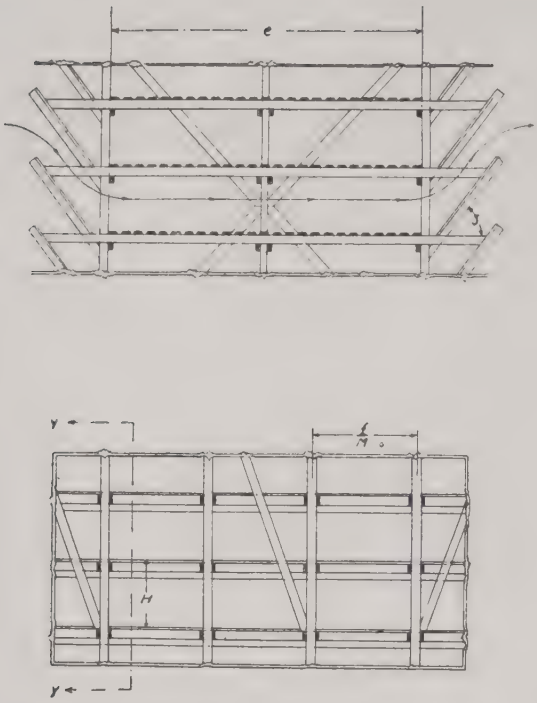

Fig. 10. Atmospheric Splash-Deck Tower

heat and mass transfer from the splash decks, jets, and drops.

A. Atmospheric Splash-Deck Towers

Methods of an essentially empirical nature are presently employed for atmospheric tower design. Nance[10] presents a discussion of the general problem for large towers of one particular splash-deck construction. Some exploratory work on splash-deck performance has been undertaken by Marchant[11].

Simons[12,12a] offers information applicable to the design of small and medium sizes of atmospheric towers. **Fig. 10** illustrates the type of tower to be considered. A complete design study involves many details, more than can be included within the scope of this discussion. For rapid estimates on average constructions, however, a very simple relation can be used to relate the number of splash decks required, the water temperatures, the wind velocity, the water "loading," and the tower plan dimensions.

For constructions involving approximately 2.7 sq ft of wetted-surface area per sq ft of active plan area of the tower, and for average louver constructions with a lower inclination angle of 55 degrees with the horizontal, the number of splash decks required, N, may be determined from the equation

$$N = \frac{r(\Delta C) V_1}{m} \sqrt{\frac{30}{B}} \qquad (23)$$

The quantities in Equation (23) are obtained as noted below:

Symbol	*Designation*	*Source*
ΔC	Cooling factor	Fig. 11
r	Deck factor, (sq ft)/(gpm) (deck)	Fig. 11a
V_1	Wind velocity factor	Table 2
m	Loading factor, (sq ft)/(gpm) (deck)	Fig. 11b
B	Barometric pressure, in. of mercury	

In one use of Equation (23), the initial and final water temperatures may be presumed known with the number of decks to be determined. Magnitudes of the cooling factor, ΔC, are then obtained from **Fig. 11** as the **difference** of the ordinates for the proper atmospheric wet-bulb curve corresponding to the values of T_2 and T_1. The inverse problem, with N given and either T_2 or T_1 to be solved for (one of them being known), is solved by using Equation (23) to determine ΔC, and Fig. 11 to establish the unknown temperature with ΔC as the ordinate **difference.**

B. Small Atmospheric Spray Towers

A type of small atmospheric spray tower occasionally employed in air conditioning or refrigeration jobs is illustrated in **Fig. 12.** With broadside winds of 3 to 5 mph, and with efficient swirl-cone spray nozzles, the volume of the spray space required may be estimated from **Table 3.** At a barometric pressure of 30 in Hg, the performance of

Table 2. Values of Wind Velocity Factor for Equation (23)

Wind velocity, mph	Factor V_1
5	1.00
4	1.10
3	1.25
2	1.45
1	1.70
0	2.10

Magnitudes of the factor V are for broadside wind. If the wind direction is to be lengthwise through the tower, use the factor for a wind velocity of zero mph.

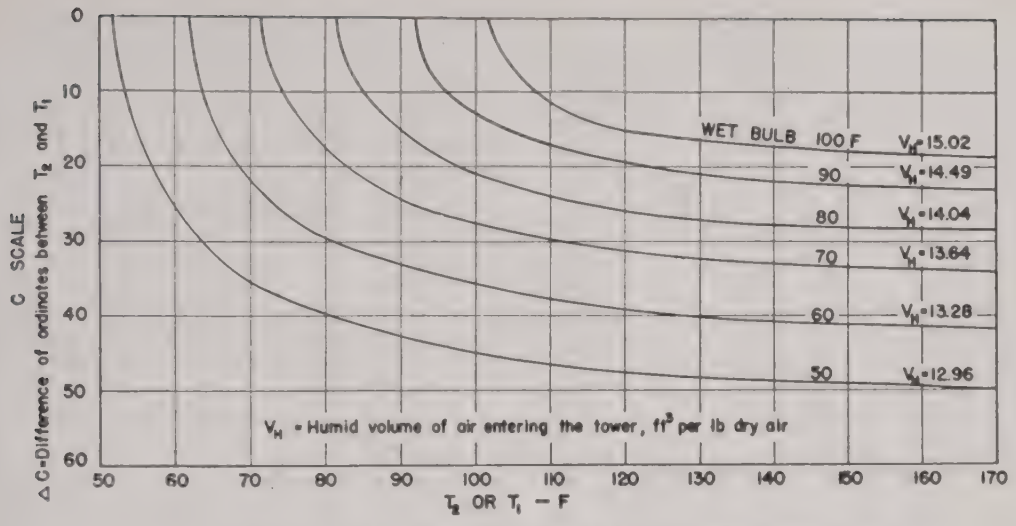

Fig. 11. Cooling Chart for Air Flow at Right Angles to Water Flow and a Barometric Pressure of 30 inches of Mercury

such a tower may be estimated with the aid of Fig. 11. For this design, the factor ΔC is determined as

$$\Delta C = \frac{39}{T_1^{0.4}} \qquad (24)$$

where T_1 is the final temperature of the cooled water, deg F.[13,14] The relation between the wet-bulb temperature and the initial cooling-water temperature which can be handled then follows from corresponding points on Fig. 11, with the factor ΔC always taken as the difference in the ordinates corresponding to the initial and final water temperatures on the curve for the prevailing wet-bulb temperature.

C. Atmospheric Screen-Deck Towers

This type of tower is built with decks of meshed wire screen. A representative design employs $\frac{1}{2}$-in. mesh, 18-gage galvanized screen stretched more than 2×4-in. supports spaced not over 4 ft apart. The sides and ends are louvered similarly to the splash-deck tower. Water distribution may be by means of a perforated flume or low-pressure nozzle system. In either case, the streams fall onto splash boards before contacting the screens.

The vertical deck spacing is 18 in. on centers for the representative tower on which data are known. With an active tower width of 12 ft, 15 cu ft of internal tower volume is provided for each gpm of water circulation. An average loading is 1.25 gpm per sq ft of horizontal active area; this should not exceed 1.50.

When a screen-deck tower is constructed to the specifications indicated,[15] its performance for a barometric pressure of 30

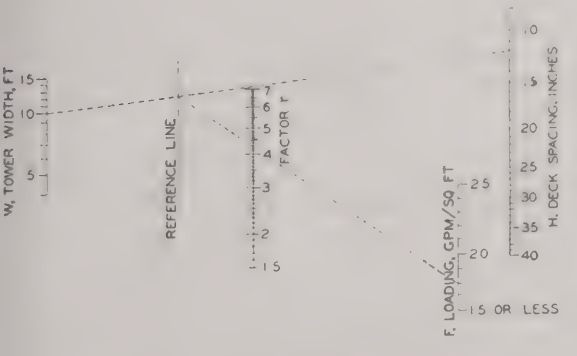

Fig. 11a. Alignment Chart for Evaluating r in Equation 23

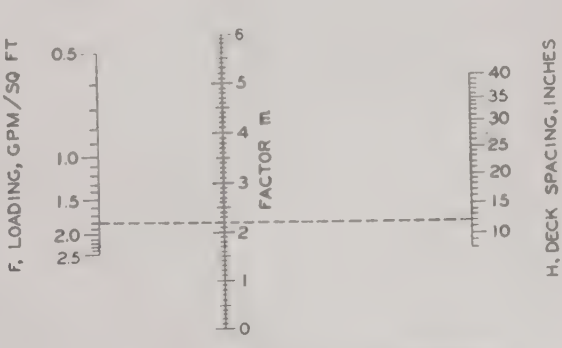

Fig. 11b. Alignment Chart for Evaluating m in Equation 23

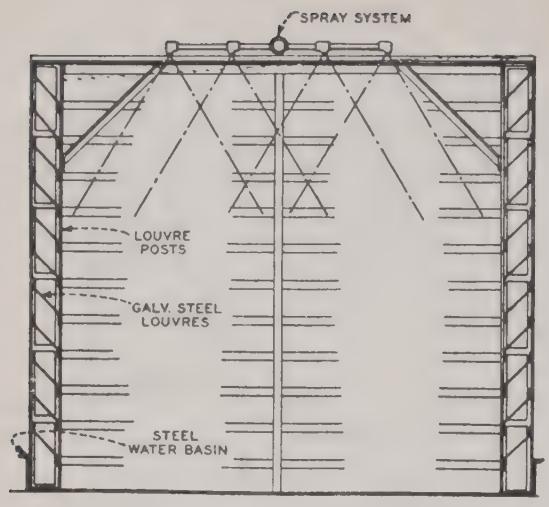

Fig. 12. Atmospheric Spray Tower Equipped with Louvres to Reduce Water Loss Due to Windage

in. Hg may be estimated from Fig. 11 and the relation

$$\Delta C = \frac{43}{T_1^{0.4}} \tag{25}$$

where T_1 is the final cooling water temperature, deg F, and ΔC is the ordinate difference as before.

Screen-deck atmospheric towers are to be used only where a high rate of windage loss is not objectionable from either the standpoint of economy or that of drift nuisance. Use of more than 15 decks will produce heavy drift loss at the bottom for even medium winds.

Spray Pond Performance

A spray pond may be employed where sufficient ground area and suitable surroundings exist. A louvered fence should be installed for drift prevention when the distance between the spray nozzles and the edge of the pond is 25 ft or less.

Fig. 13 shows a basic layout employing clustered nozzles. The cooling performance of this arrangement may be estimated with the aid of Fig. 11 and the proper value of ΔC. For a nozzle pressure of 7 psi, a broadside wind of 5 mph, and a barometer of 30 in. Hg, the operation of the **single line** of nozzles may be approximated by the following values of ΔC:[16,17,18]

(a) Individual nozzle capacity of 25 gpm

$$\Delta C = \frac{49}{T_1^{0.4}} \cdot \tag{26}$$

(b) Individual nozzle capacity of 40 gpm

$$\Delta C = \frac{87}{T_1^{0.6}} \tag{27}$$

where T_1 is the final water temperature, deg F, and ΔC is the ordinate **difference** from Fig. 11 as before.

If a second row of nozzles is added in a leeward position, the pond operation may be estimated by considering the same air stream to be acted on in two stages. The procedure in this latter case may be outlined as follows:

1. Determine the broadside cross-section flow area per cluster. This is equal to the product of the maximum height of the spray cone above the

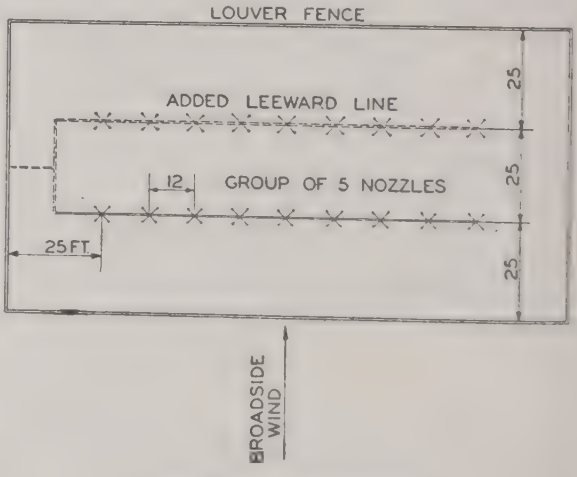

Fig. 13. Spray Pond Plan

Table 3. Factors for Sizing Small Atmospheric Spray Towers

Active tower width, ft	6	8	10	12	14	16
Gpm per sq ft active cross-section area	1.95	1.70	1.52	1.38	1.25	1.14
Cu ft of spray space per gpm	5.80	8.00	9.60	11.0	12.2	13.1

water surface and the center spacing between clusters.

2. Select the prevailing wind velocity, the dry-bulb and wet-bulb air temperatures, and the air density.
3. Calculate the weight of the air flowing per unit time through the cross-sectional area allotted to each cluster, W_c.
4. Determine the estimated performance of the first row of nozzle clusters from Equation (26) or (27) and Fig. 11.
5. Determine the estimated mean wet-bulb temperature of the air entering the second row of clusters from the approximate relation,

$$\frac{L}{N_C}(T_2 - T_1) = W_c(h_2 - h_1) \qquad (28)$$

where $(h_2 - h_1)$ is the enthalpy increase of the air stream in passing over the first row of nozzles. In many practical cases, it is satisfactory to employ the sigma-function difference as an approximation to the enthalpy difference,

6. Determine the estimated performance of the second row of nozzles as in step (4).
7. Since the nozzle lines are identical and fed from a common supply, the resultant cooling range will be the average of the ranges for the two separate rows.

Economic Aspects of Cooling-Tower Design and Selection

No simple rules can be given to cover the many-sided questions of cooling-tower economics. The tower with the lowest annual cost, all factors considered, is desired for each job. Possible nuisance effects of drift, noise, and size may be very difficult to appraise. Close cooperation between the purchaser and the manufacturer is the most satisfactory basis of economic selection.

The designer's requirements are assigned in terms of the water rate, the cooling range, the design atmospheric conditions, and the enthalpy (or wet-bulb) approach. The water rate, cooling range, and cooling approach in turn are subject to economic requirements in the province of the purchaser. Cooling-tower costs and process costs are inseparably linked together.

Economic selection is usually made by the procedure of selecting a few representative designs, calculating the costs, and choosing the minimum total cost. Within this procedure the cooling-tower designer has a considerable range of possibilities to judge between. The general nature of the **design variables** for study may be indicated by the outline below.

A. Considerations Affecting Plan Cross-section Dimensions
   1. Lower limit of liquid loading, lb per hr per sq ft, where incomplete packing wetting begins*
   2. Upper limit of liquid loading, lb per hr per sq ft, where flooding occurs**
   3. Spray patterns obtained in spray towers from different nozzles
   4. Uniformity of distribution for both water and air; no "channeling."

B. Considerations Entering the Fan Selection for Forced-Draft Towers
   1. Air flow rates required to obtain specified performance
   2. Flow losses in the air flow circuit and fan power requirements corresponding
   3. Forced or induced draft
      a. Ease of base-level fan mounting
      b. Danger of air recirculation with low towers and base-level fan intake.

C. Considerations Affecting Height of Tower
   1. Required tower volume along the air and water flow paths to obtain the specified cooling performance
      a. Influence of different packings on required volume
   2. Water pumping power requirements
      a. Balance with fan power for greater or less tower height for different packings
      b. Pressure required for spray-nozzle operation

* For slat-packed towers this is around 500 lb/(hr) (sq ft).

** For slat-packed towers this is around 3,000 lb/(hr) (sq ft).

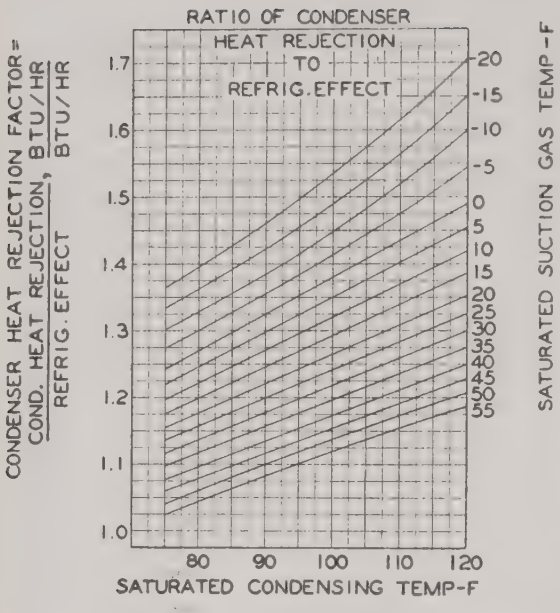

Fig. 14. Chart for Estimating Heat Rejection from Refrigeration Condensers (Mechanical Refrigeration)

3. Stack effect with emergency fan shutdown
4. Structural requirements.

D. Considerations of Assembly, Maintenance, and Operating Costs

1. Materials and fabrication
2. Packing costs vs performance advantages of different packing forms
3. Foundation and catch basin
4. Annual operating cycle in relation to justifiable investment
5. Cleaning and corrosion prevention.

For cooling-tower applications in **refrigeration** and **air conditioning jobs** using **mechanical compressors,** an approximate estimating basis may be cited for determining:

1. The heat to be rejected from the condenser cooling water for each unit of refrigeration effect, and
2. The rate of cooling water circulation for each unit of refrigeration effect.

Fig. 14 is a chart for estimating the heat rejection or cooling-tower load. This may be applied for preliminary estimates to any modern mechanical-refrigeration unit. A common first-estimate figure, where no other data are available, is a magnitude of 1.25 for the condenser heat rejection factor. (This is equivalent to 250 Btu/min per ton of refrigeration effect.)

While the heat-rejection load is dependent upon the system served by the cooling-tower, this system merely introduces general limits on the water circulation rate, established by the condenser temperature to be maintained, the limiting cold-water temperature attainable, and water circulating capacities suitable for the system being served. The final economic balance between the water rate and the water cooling range is usually determined by cooling-tower costs and hence is primarily within the province of the cooling-tower engineer.

A common, rough, first estimate for mechanical-refrigeration air conditioning units is to assume a cooling range of 10F. For a condenser heat rejection factor of 1.25, this determines a water circulating rate of 3 gpm per ton of refrigeration. Taking this condition as a starting point, and keeping the cooling-tower load constant, probably the simplest way to present the trend of cooling-tower performance with a varying water rate is to set up a comparative tabulation for a typical case. Such an example is given below.

Table 4. Sample Trend of Cooling Tower Performance with Constant Load and Various Water Rates

(Wet-bulb temperature = 70°F)

Water rate, $\left(\frac{\text{gpm}}{\text{ton}}\right)$	Water cooling range, F	Wet-bulb approach, F	Cold water temperature, F	Water inlet temperature, F
2	15	9.4	79.4	94.4
2.5	12	9.8	79.8	91.8
3	10	10.0	80.0	90.0
4	7.5	10.1	80.1	87.6
5	6	10.2	80.2	86.2
6	5	10.3	80.3	85.3

The performance tabulated brings out the important general observation that, for the constant conditions stated, the wet-bulb approach is raised a small amount as water is circulated at a greater rate. Hence, if it were desired to attain a closer wet-bulb approach, in this particular instance it would be desirable to select an entirely different tower construction

rather than to circulate more water. On the other hand, if the system being served should be one which required a relatively low water cooling range, then a high water rate might be the economic answer.

Furthermore, an air conditioning system which would circulate 3 gpm per ton to a mechanical-draft cooling tower would probably find it more economical to circulate 4 to 5 gpm per ton to a spray pond or atmospheric cooling tower. Then, consider the case of an ice plant operating many more hours per year than an air conditioning system; here, a larger initial investment might be justified by a lower annual operating cost, and a circulation rate of 4 to 5 gpm per ton might be the economic selection.

The preceding considerations serve to demonstrate what is worth repeating again by way of conclusion, that cooling tower selection is always a problem in sound engineering and practical economics.

Nomenclature

A_h = plan cross-sectional area of active portion of cooling tower, sq ft
a = area of air-water interface per unit of tower volume, sq ft per cu ft
B = barometric pressure, in. Hg
ΔC = cooling factor, Fig. 11, taken as the difference in ordinates between T_2 and T_1 for the wet-bulb curve involved
c_{p_a} = unit heat capacity of dry air at constant pressure, Btu/(lb)(deg F)
c_{p_v} = unit heat capacity of atmospheric water vapor at constant pressure, Btu/(lb)(deg F)
c_{p_w} = unit heat capacity of liquid water at constant pressure, Btu/(lb)(deg F)
d = differential operator signifying a very small increment of a quantity
e = active tower width, ft, Fig. 10
F = loading, (gpm)/(sq ft), Figs. 11a, 11b
G = dry air flow rate, lb/hr
H = vertical distance center to center of splash decks, in., for Figs. 10, 11a, 11b only
H = Specific humidity (humidity ratio) of the main gas body, lb vapor per lb dry air
H_i = Specific humidity (humidity ratio) at the air-water interface (saturation at t_i), lb/lb
H_ν = Specific humidity (humidity ratio) of saturated air at the temperature of the main liquid body, lb/lb
h = enthalpy of the air-vapor mixture, Btu/(lb dry air in the mixture)
h_ν = enthalpy of the air-vapor mixture saturated at temperature ν, Btu/(lb dry air in the mixture)
h_{ν_1} = value of $h\nu$ at temperature ν_1
h_{ν_2} = value of h_ν at temperature ν_2
h_g = unit conductance for sensible heat transfer between the air-water interface and the main gas stream, Btu/(hr)(sq ft)(deg F)
h_L = unit conductance for heat transfer between the main water body and the air-water interface, Btu/(hr)(sq ft)(deg F)
K_g = overall unit conductance for mass transfer between the saturated air-vapor mixture at the main liquid-body temperature and the air-vapor mixture in the main gas body, lb/(hr)(sq ft)(lb/lb)
k_g = unit conductance for mass transfer between the air-water interface and the main gas body, lb/(hr)(sq ft)(lb/lb)
L = water through-flow rate, lb/hr
m = mass transfer rate between the water and air streams (Eq. 14), lb/hr
m_i = mass transfer rate between the air-water interface and the main gas body, lb/hr
N = number of atmospheric tower splash decks, Eq. (23)
N_c = number of nozzle clusters in a spray pond, Eq. (26)
q_s = rate of sensible heat transfer between the air-water interface and the main gas body, Btu/hr
q_w = rate of heat transfer between the main water body and the air-water interface, Btu/hr
q_L = rate of latent heat transfer between the air water interface and the main gas body, Btu/hr
r_0 = latent heat of water evaporation at the reference temperature t_0, Btu/lb
r_i = latent heat of water evaporation at the interface temperature t_i, Btu/lb
s = unit heat capacity (so-called "humid heat") of the air-vapor mixture at constant pressure, Btu/(deg F)(lb dry air in the mixture)
t = dry-bulb temperature of the main body of the air-vapor mixture, deg F
t_i = dry-bulb temperature at the water-air interface, deg F
t_0 = datum temperature for water-vapor enthalpy, deg F
t_2 = dry-bulb temperature of air entering cooling tower, deg F
T_1 = final cooling water temperature for Eqs. (24, 25, 26, 27, 28) and Fig. 11, deg F

T_2 = initial cooling water temperature for Eqs. (22, 28) and Fig. 11, deg F

U_g = overall unit conductance for sensible heat transfer between the main water body and the main air-vapor body, Btu/(hr) (sq ft) (deg F)

V = internal volume of the cooling tower (including both free space and any packing), cu ft

V_H = volume of moist air saturated at the wet-bulb (Fig. 11), cu ft/(lb dry air)

W = tower width, ft, Figs. 11a, 11b

W_c = weight flow rate of atmospheric air per nozzle cluster in a nozzle row broadside to the wind, Eq. (28), lb dry air per hr

ν = temperature of the main water body, deg F

ν_1 = water temperature at cooling tower inlet, deg F

ν_2 = water temperature leaving cooling tower and entering basin, deg F

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22. AUTOMATIC CONTROL DEVICES AND INSTRUMENTS

Part I. Control Devices

IN ORDER to utilize fully any form of refrigeration system, whether it be domestic, commercial, industrial, or air conditioning, it is desirable to have **automatic control** of temperature and of the equipment in the system. This can be accomplished by means of devices that are sensitive to these conditions and as a result thereof operate to start, stop, or regulate the source of cooling.

Automatic controls for refrigeration can be classified into three general types:

A. Temperature controls
   1. Room thermostats
   2. Remote-bulb controllers
B. Pressure controllers
C. Differential controllers
D. Refrigerant-flow controllers.

This chapter will deal with these controllers in the above order, covering their construction, function, and some application factors inherent in the various designs.

A. Temperature Controls

1. Room thermostats. A room thermostat may be defined as a temperature-sensitive device which is mounted entirely within the space it controls. It consists of five basic parts: (1) element; (2) adjustment means; (3) switching means; (4) a base upon which the first three items are mounted; and (5) a screen or cover to protect the above parts from physical harm and frequently to make the instrument more appealing to the eye. Quite often a thermometer is mounted on the screen and the adjustment means includes a scale.

Room thermostats may be classified by their **temperature-sensitive elements.** The earliest and perhaps the simplest element is made of bimetal, so named because of its fabrication from two dissimilar metals securely bonded into a duplex strip of flat metal. A change in temperature of the bimetal causes the inactive metal, usually Invar, an alloy having a very low thermal expansion coefficient, to change its length a relatively small amount while the active metal, brass or steel, measurably increases or decreases its length, depending upon the nature and amount of the temperature change. Since the two metals are bonded together, a force is exerted to relieve the strain imposed by the heat upon the bimetal. The element therefore deflects until the force developed by the heat applied to the element just balances the resistance to strain inherent in the metal. This power and deflection, although small, is usually used to operate an electric switch or air valve.

The bimetal element of a thermostat is frequently "U" shaped and mounted on a hinge attached to the thermostat base. One free end of the element rides on a cam or lever arrangement which is the **temperature-adjustment means.** The other free end either carries one of the electric contacts or operates a small **switching mechanism.**

Fig. 1 is a schematic diagram of a simple

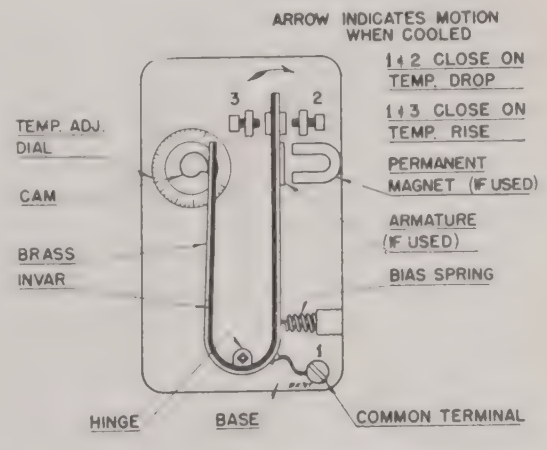

Fig. 1. Simple Room Thermostat (SPDT Switching)

room thermostat having a SPDT (single-pole-double-throw) switching mechanism.

When a SPDT thermostat is used in a circuit requiring momentary closing of one circuit to start an operation and the opening of this circuit and momentary closing of a second circuit to stop the operation, the permanent magnet shown in the figure and its related armature are not used. If a SPDT thermostat is used to control separate loads, the magnet and armature are used to prevent contact chattering and sputtering with the resultant undesirable rapid starting and stopping of the equipment. This magnet is mounted in the path of the armature and it is the air gap between the two, when No. 2 contact is closed, that determines the differential of the unit; the smaller the air gap the larger the differential, and conversely.

A damper or valve motor is sometimes controlled by a thermostat of the former type and is moved in one direction by closure of one contact, in the opposite direction by closure of the other contact, and allowed to remain at rest whenever the movable contact is contacting neither stationary contact. Such a system is frequently called **"floating control."**

In some SPST (single-pole-single-throw) thermostats but one stationary contact is used and the magnet and armature are incorporated for reasons stated above. There are a few thermostats used in specific circuits that have no magnet and employ slow-moving contacts to produce very low differentials. These contacts are necessarily very limited in their load-carrying capacity.

In one special type of room thermostat, differential is introduced electrically by employing the start-stop-holding circuit principle. In this thermostat two contacts are closed or opened sequentially, the circuit requiring them both to be closed before the system can start and both to be open before the system can stop. Such a thermostat gives small operating differential but is also limited in its load-carrying ability.

"Manual differential" is the difference in **scale movement** that is necessary to cause starting and stopping of the equipment being controlled. "Operating differential" is the difference in **temperature** necessary to cause similar results. In a simple thermostat the latter is always greater than the former because of the inherent lag of the element plus the heat capacity of the base, screen, and other parts. Artificial heat is frequently applied internally in heating thermostats to increase the rate of response, which in effect reduces the operating differential.

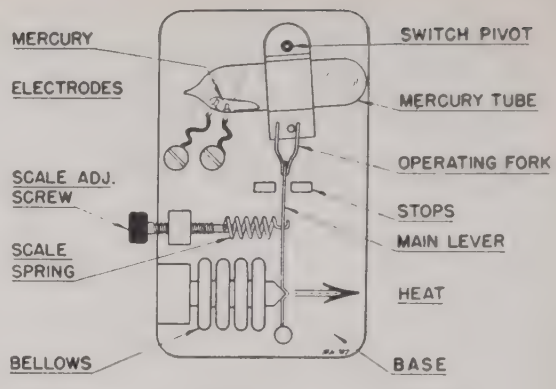

Fig. 2. Vapor-Filled Refrigeration Room Thermostat

A widely used element for refrigeration room thermostats is the **vapor-filled type.** In this type a metallic bellows is filled with a small quantity of a volatile liquid and the vapor pressure exerted on the bellows is used to operate electric switches or valves. **Fig. 2** is a schematic diagram of a thermostat of this type which employs a sealed contact structure.

The bellows of this instrument is evacuated and then filled with a small quantity of a liquid whose vapor pressure curve is within the desired temperature operating range. Freon, propane, ether, and other similar compounds are examples of fills used in these controls. A scale-adjusting spring is provided to determine the pressure against which the bellows must operate and consequently to determine the temperature at which the thermostat will control.

The main lever of the thermostat is pivoted at one point, the bellows and spring act at two other points, and at a fourth point is a power takeoff for operating the controlling member, either by electric contacts or a pneumatic valve. Differential is usually increased by means of an adjustable degree of lost motion in the drive mechanism.

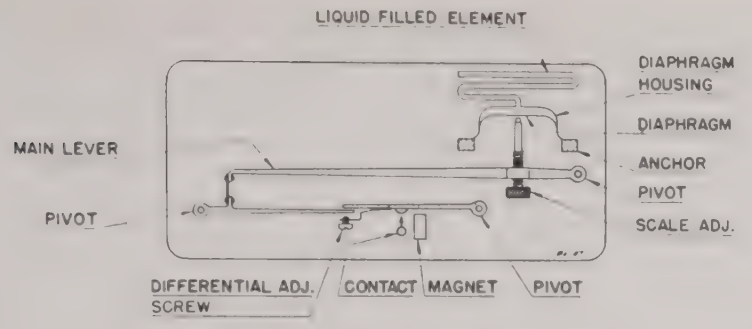

Fig. 3. Liquid-Filled Diaphragm Thermostat

If **open contacts** are used, they may be of the type described under the bimetallic type element, excepting that the detent may be accomplished by means of an over-center action rather than a permanent magnet. A popular switching mechanism is the so-called closed or **sealed contact** wherein two electrodes are sealed in a glass or metal envelope and the circuit is made by means of a small globule of mercury that flows back and forth within the envelope as it is tilted by the thermal element. Either type of contact structure can be had in SPST, SPDT, or special switching means.

Another type of sealed switch has a small globule of mercury in a sump in the glass enclosure. Hinged above it is the other electrode to which a small armature is fastened. A permanent magnet is then operated toward and away from the outside of the glass enclosure and consequently makes and breaks the circuit between the hinged electrode and mercury.

In operation the scale is set at the desired control point. An increase in temperature causes an increase in internal pressure which when it is sufficient overcomes the force of the spring and operates the main lever to tilt the switch, closing its contacts. A drop in temperature causes the converse to happen.

Another very popular switching means for this type thermostat consists of a small **"micro" switch** having either an over-center snap action or a magnetic detent. The advantage of this switch and also the sealed mercury switch is their ease of replacement following failure in the field. The micro switch is usually available in SPST or SPDT switching action.

Generally speaking, this type of thermostat may handle fairly large electrical loads directly. Units have been manufactured which can handle 45 amps at 115 volts. Also, this type of thermostat, because of its greater mass and the larger motion required to operate the contacts, will have a greater operating differential than the bimetallic types. If sealed contacts are employed, they will naturally require no cleaning during their life and are unaffected by moisture or other corrosive atmospheres. If the mercury switches are used, the controls must be accurately levelled as a small degree of tilt will throw the instrument off calibration or in some cases will cause it to short-cycle.

A third common type of element used for room thermostats is a relatively new one, consisting of a **hydraulic fill system** wherein the expansion and contraction of a liquid in a closed system caused by temperature changes is used to operate a metallic diaphragm which in turn operates a switch or a valve. **Fig. 3** is a schematic diagram of this type of thermostat.

The element usually consists of a short length of small diameter tubing, one end of which is fastened to a diaphragm housing so that the inside of the housing and the inside of the tubing are connected. The diaphragm is then securely fastened inside the housing, leaving very little space between the housing and diaphragm. After the diaphragm is put in place, the system is evacuated and filled with a liquid having the desired characteristics and the free end of the tubing is sealed. A lever system designed to multiply the diaphragm motion greatly is provided with an adjustment to vary the position of the diaphragm at which it moves the contacts.

Operating differential is increased by an

adjustable spacing in the lever system. The available motion in this type of element is quite small but the force developed is large. Because of this last characteristic it is absolutely essential that the diaphragm housing and tubing be of sturdy construction. It is also imperative that the diaphragm be sufficiently rigid to insure its motion occurring at the point of contact between it and the push-rod, and not at other points in its area. This last requirement is quite often accomplished by means of ribbing the diaphragm radially from the center. These elements can often be damaged beyond repair by denting the tubing or housing, which causes a reduction in the internal volume of the system.

Because of the small motion and tremendous force that is available, these units usually operate through a tremendous leverage before actually driving the contacts. In spite of the large forces developed by a liquid fill the actual work possible from it is less than with other control elements, unless the bulb is made extremely large. This is due to the extremely small motion produced for a given temperature change.

2. A remote-bulb temperature controller may be defined as a temperature-sensitive device in which the element or bulb is placed at the point in space whose temperature is to be controlled and the remainder of the instrument is placed elsewhere in the space or in a separate room. This type of controller consists of: (1) remote bulb and its capillary tubing; (2) case; (3) bellows or diaphragm; (4) scale and differential adjustment means; and (5) switching or valve-actuating means.

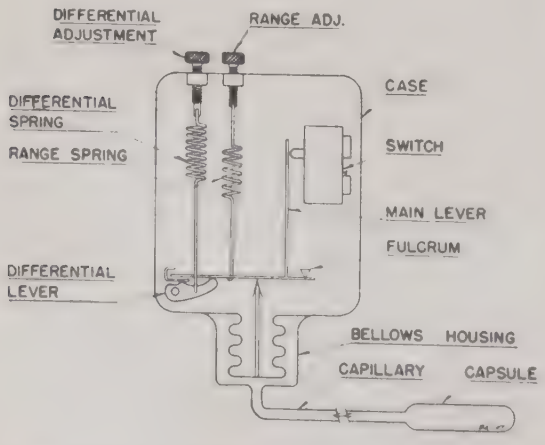

Fig. 4. Remote Bulb Temperature Controller

Here again the best division of types of remote-bulb temperature controls appears to be by types of sensitive elements. The most common type uses the **vapor-tension fill** whose basic operating principle has been earlier described. **Fig. 4** is a schematic diagram of a controller of this type.

Since these controllers are seldom used in public places or rooms where appearance is important, they may be less attractive than the room thermostat, and have a more functional appearance. Methods of operation, types of switching means available, and structural features of this unit are similar to those given under room thermostats of the same type with two basic exceptions.

First, the bellows, instead of being the sensitive element and power source, as in a room thermostat, becomes the power source only. The temperature-sensitive element is in the remote bulb as within it is contained the active liquefied portion of the vapor fill. The remote bulb is connected to the bellows by means of flexible capillary tubing or by means of long rigid tubing.

Secondly, the differential is usually increased by means of a second spring which adds additional loading to the main lever through a differential lever which engages the main lever for a portion of the time when the main lever is in position to operate the switch. If adjustable differential is not required this mechanism is omitted from the instrument.

Even with a small bulb, which permits good control sensitivity, the work available from a vapor-filled element is greater than with the other types of elements, being limited only by the size and type of bellows.

Fig. 5 illustrates three common types of vapor-tension fills employed in bellows assemblies. On the left is a so-called fade-out or **limited fill,** which is employed for low-temperature applications in which the bulb is always at a lower temperature than the bellows. Because it is at a lower temperature, the liquid in the system will always exist in the bulb and nowhere else, and since it is the temperature of this liquid surface which determines the pres-

sure in the rest of the system, proper control obtains from variations in bulb temperature. Such bulbs are of small size, relatively smaller than illustrated.

The right-hand illustration in Fig. 5 shows the so-called **high-temperature fill** in which the bulb is always at a higher temperature than the bellows. Note that in this diagram there is enough liquid in the charged system to fill the bellows and capillary completely and to fill the bulb partially. Thus even though the liquid is cooler in the bellows than in the bulb, there will always be a liquid surface in the bulb which controls the pressure in the rest of the system, and the remaining liquid merely becomes a hydraulic pressure transmitting means. Thus the bulb will always be the controlling point as long as it is warmer than the bellows.

The center shows a common **cross-ambient control arrangement** in which a larger bulb is employed, sufficiently large so that if the bulb is colder than the bellows the entire charge is contained in the bulb, and the bulb becomes the controlling point of the system. If, on the other hand, the bellows is colder than the bulb, it fills with liquid, as does the capillary, and yet due to the size of the bulb it retains sufficient liquid to do the controlling. Thus, regardless of whether the bellows temperature is above or below the bulb temperature, the desired control point is obtained at the bulb. This distinction is very important in all types of work where the ambient temperature at the bellows or capillary varies from above to below the setting at which the bulb must operate the control.

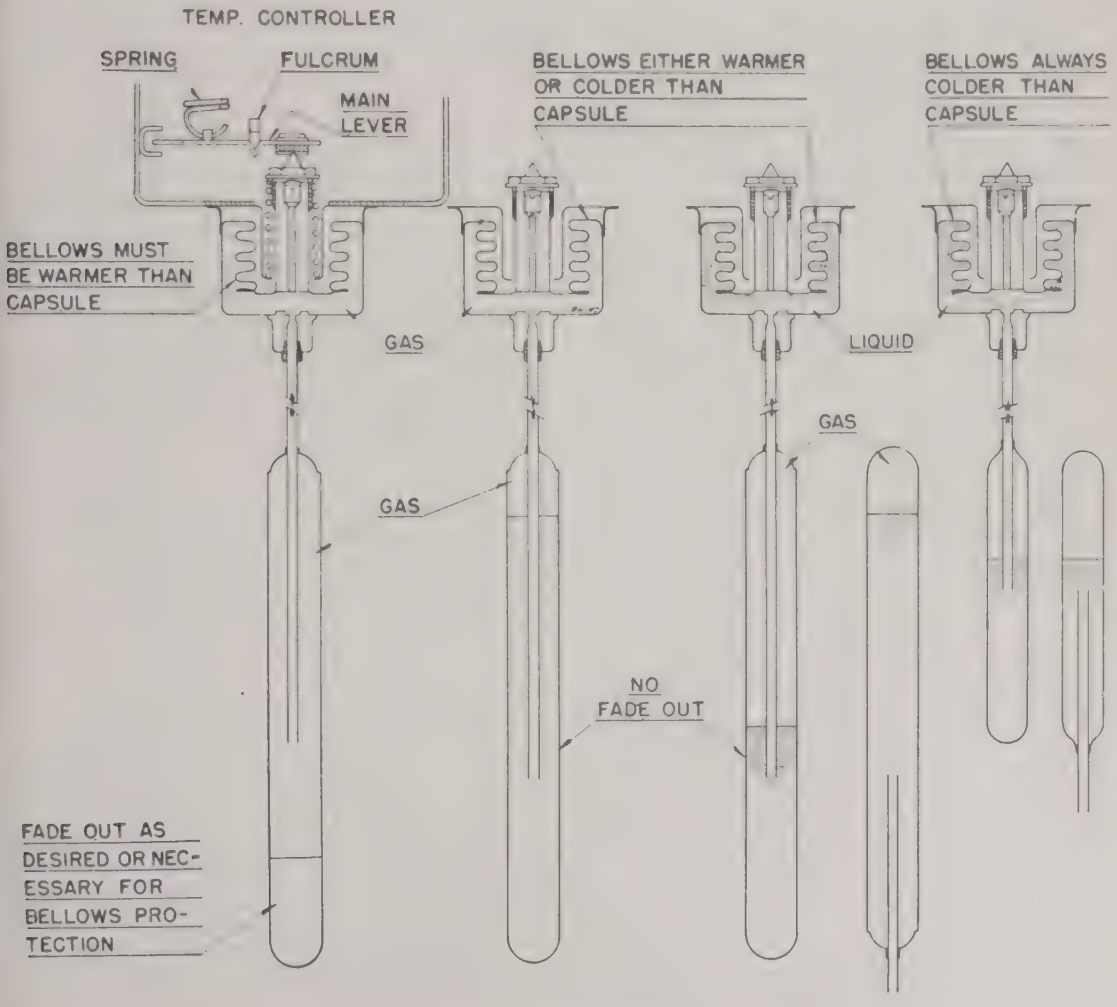

Fig. 5. Filling Arrangements for Bellows Assemblies

Another type of remote bulb controller is the liquid-filled type which in outward appearance can resemble the room thermostat with liquid-fill element earlier described. The active element, as the name applies, consists of a length of capillary tubing that extends out of the case of the instrument instead of being coiled up within it and usually terminates in an enlarged bulb. This element can therefore be placed at the position in space requiring the temperature control and the case can be mounted elsewhere, even in a separate room if desired.

B. Pressure Controllers

Pressure controls for refrigeration may also be classified by means of their power elements. There are three types now in general use: (1) bellows; (2) diaphragm; (3) bourdon tube.

The **bellows type** pressure control is identical in construction to the remote-bulb temperature control except that the pressure connection to the power element is made by connections such as pipe threads or flare connections, or by a short length of capillary enlarged on the free end to a standard tube size.

Its internal operation is also identical with that of the temperature controller, since the bellows is now subjected to refrigerant pressure and not the pressure developed by the fill. In order for the bellows type control to operate in the subatmospheric range, it is necessary to incorporate a compression spring within the bellows housing. This spring is in addition to the scale adjustment spring, but its force is opposite to that of the latter.

The force available for doing work in a control of this type is dependent first upon the pressure in the system and secondly upon the area of the bellows. If the bellows is large enough, a tremendous force can be had for operating heavy switches, water valves, or even refrigerant valves. In these heavy-duty controls the minimum differential is necessarily quite large because of the spring rate of the bellows and opposing spring system.

A second type of element in which the available motion is much less than the bellows type is the **diaphragm type.** Since the motion is very limited, large leverages are needed to produce motion adequate to operate most switches. In this type of control the leverage and spring system is quite similar to those earlier described. It is essential, however, that the diaphragm be restrained either by a backing plate or reinforced with radial ribs in order to guarantee the motion occurring at the point of contact between diaphragm and push rod.

The third type of element is the **bourdon tube** type used for many years in pressure gages. **Fig. 6** is a schematic diagram of this type.

As shown by the cross-sectional view of the element, it consists of a flattened tube rigidly held at one end and free to move toward or away from the free end by means of the scale adjustment screw. It is the position of the switch relative to the end of the bourbon tube that determines the scale setting. Differential can be increased by employing a linkage wherein the lost motion between bourdon tube movement and switch is increased.

C. Differential Controllers

Controllers used to maintain a given difference in pressure or temperature between two pipe lines or spaces, are known as differential controllers. These controllers, since they must sense conditions in two different locations, must have two elements, either pressure or temperature-sensitive.

Fig. 7 is a schematic diagram of a **differential-pressure** controller, using bellows for elements.

As shown, the two elements are rigidly connected by means of a rod so that motion of one causes motion of the other. On the connecting rod is a power takeoff that operates contacts or valves (SPDT contacts are shown). A compression spring has

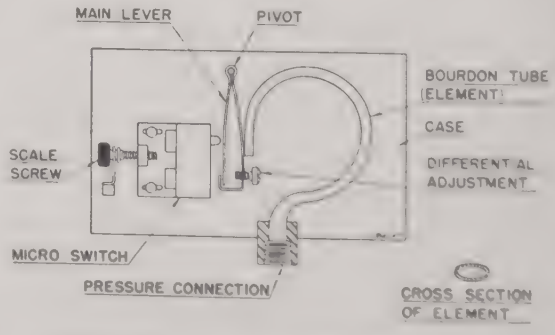

Fig. 6. Bourdon Tube Pressure Control

been provided to permit setting of the differential pressure at which the device operates. The sum of the force developed by the low-pressure bellows and the scale spring equals the force developed by the high-pressure bellows at the control point.

In discussing this type of controller, two terms must be defined and thoroughly understood. **Instrument differential** is the difference in the pressure or temperature that is maintained between the high and low-temperature bulbs of the system. **Operating differential** is the difference in pressure or temperature that is necessary to operate the switch at the instrument differential that is to be maintained. Operating differential can be increased by means of a second spring that acts in the same direction as the first and which takes effect within the operating differential of the instrument. A second method could be adjustment of the distance between collars Z-Z on the connecting rod; the farther these are apart the larger the operating differential.

The foregoing discussion on differential controllers has dealt entirely with the pressure type. The **temperature-sensitive** type is made by merely replacing the pressure bellows with vapor tension elements, whose theory of operation has been earlier described. Where a constant instrument differential must be maintained throughout a large band of temperatures, it is usually necessary to use a different fill in one element than in the other if of the vapor type, or to use elements and drive of the liquid-filled type.

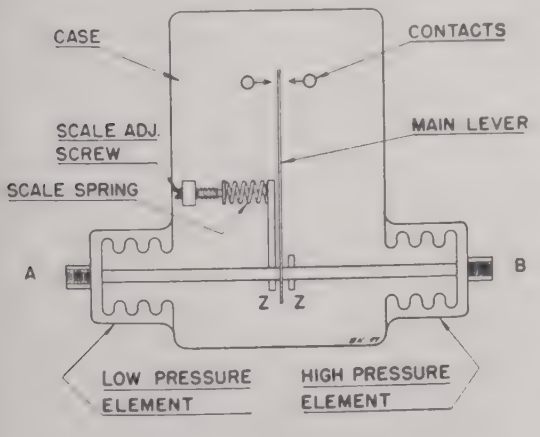

Fig. 7. Differential-Pressure Controller

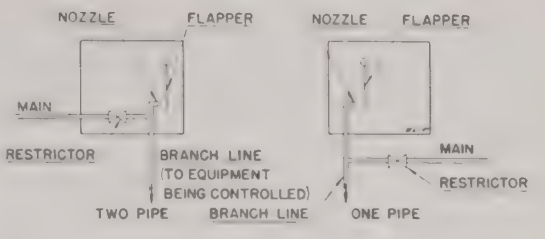

Fig. 8. Pneumatic Controllers

A second type of differential-temperature controller employs two capsules and capillaries connected to one bellows and a liquid fill. This is known as a **constant-volume** type fill, since the operating point corresponds to a constant volume of the two capsules, capillaries, and bellows. It should be noted that if the two bulbs have equal volume, a rise in the temperature of one bulb requires an equivalent fall in the temperature of the other to maintain the operating point.

The foregoing discussion on controls has dealt entirely with those having electric contacts to do the actual operating of controlled equipment. Another very popular type uses a small air valve, and positions dampers or valves by means of regulating the air pressure on the bellows or diaphragm of the equipment being controlled. These are known as **pneumatic controllers** and can be classified as either one-pipe or two-pipe devices. **Fig. 8** shows the difference between these.

In these controls compressed air at about 15 psi pressure is used as the activating medium and is supplied by the main to the controller. In the two-pipe system, the restrictor is built into the controller in the main to the nozzle, and controls the rate at which air is supplied to the nozzle and consequently the pressure in the branch line. If the flapper completely seals the nozzle, full main pressure is built up in the branch line, and if the nozzle is not sealed at all by the flapper, no pressure is built up in the branch line, as the air leaks out the nozzle faster than it can be supplied through the restrictor in the main. In the one-pipe system the restrictor is external and inserted in the main ahead of the tee that goes to the controller and controlled equipment.

To understand automatic controls thoroughly and use them intelligently, some knowledge of their applications, limita-

tions, and operating characteristics is essential.

Effects of altitude changes. Any control that relies on pressure to operate a switch or valve is subject to variations in operating point as the atmospheric pressure changes. Normal variations in atmospheric pressure do not noticeably change the operating point, but a change in altitude will affect the control point by an amount governed by the change in absolute pressure. The greater the altitude, the lower will be the operating point of a conventional pressure or temperature control. If a pressure control is being considered, the change is directly proportional to the change in altitude. If a temperature control is being considered, the change in calibration equals the change in vapor pressure translated to degrees temperature, at the scale setting being used. These characteristics are especially important when controls are being selected for use in planes, buses, and trains that in normal use are subjected to wide variations in altitude.

Bimetallic and liquid elements and special bellows-type elements using a compensating bellows are not affected by changes in altitude, so if altitude is the only problem to overcome, their use for these applications should be considered.

Mobile uses of controls. In discussing mobile applications, it must be remembered that there is always vibration present which will affect the operation of some types of controls. The types that are wholly unsatisfactory for mobile applications are those employing mercury tubes, slow-moving contacts, or any that are subject to leveling. There remains, then, only those controls that have positive contact action and employ mechanical or magnetic detents on their contacts. Generally speaking, the so-called micro-switches are also satisfactory for this application.

Levelling of controls. Any control that employs a mercury switch is extremely sensitive to leveling and should be mounted so that the sides of the base are vertical or the top is absolutely level. Great care is taken at the factory to calibrate the instrument in the correct position, and in order to use its accuracy fully, it must be properly installed. Some remote-bulb and pressure controllers are conveniently equipped with a plumb-bob to facilitate proper installation. In some instances off-level installation will not only upset the calibration but will also render the instrument completely inoperative.

Effect of internal heat. All room thermostats employing contacts are to some degree affected by the electrical load being handled. This is because of the electrical resistance of the current carrying parts generating heat when current is flowing. In refrigeration thermostats, this small amount of heat tends to increase the operating differential, a seldom desirable characteristic. A common name for this phenomenon is **"contact heat."** It is probably most noticeable in bimetallic-type controls, because the element itself frequently carries current and the contacts are usually fastened to the element. Vapor-tension and liquid-filled thermostats are somewhat affected but usually to a smaller degree, because the heat generated by the load-carrying parts reaches the element mainly by convection and not by thermal conductivity. Thermostats employing micro-switches can be designed to minimize greatly the effect of contact heat by isolating the switch from the element with material having low thermal conductivity. As is obvious in a given thermostat, the contact heat increases as the square of the contact load in amperes and consequently the greater the load the greater the operating differential.

Thermostat location. Since any temperature control has inherent in it some thermal lag, its performance is affected by the rate of temperature change and rate of fluid motion to which the element is subjected. The operating differential varies directly as the rate of temperature change and inversely as the rate of fluid motion across the element. Proper selection of element location can often minimize the effect of these two variables by subjecting the control to average and not extreme conditions. For example, it would probably be undesirable to mount a room thermostat where it would get the immediate full blast of cooled or heated air. Mounting of the capsule of a remote-bulb controller in a duct having a high air velocity will usually produce closer control than if it is mounted in a duct having a lower air velocity. In

some remote cases it might be desirable to use a small blower to circulate air over the element.

Some unique designs have been marketed to minimize these effects. One remote-bulb control has been developed which includes fins on the thermal bulb; a thermostat has been marketed which includes a shunt type of heater to reduce operating differential. The value of the former is questionable as the mass added by the fins almost offsets the gain made by increased element area. The latter method is entirely practical since it requires merely a high-resistance heater across the switch contacts which generates heat when the unit is off. This heater merely accelerates the element action back to an "on" position of the contacts. The heat generated by the heater affects the element in the same manner as the increase in space temperature resulting from the equipment being inoperative.

Explosive atmospheres. An important consideration when selecting controls for an operating room is the explosiveness of the atmosphere that is present during operations. So-called "sealed-in-glass" contacts are seldom considered explosion-proof, and other methods must be used to eliminate any possible electric spark within the atmosphere. Quite frequently an explosion-proof case can be used to surround the control case and contacts, permitting only the capsule and capillary to extend within the conditioned space. It is frequently possible to use a long capillary tubing and mount the instrument case in an explosion-free atmosphere elsewhere in the building.

Cross-ambient effects. All remote thermal elements are to some degree affected by the temperature of the operating means, whether bellows or diaphragm, and by the temperature of the capillary that transmits pressure to it. The degree and type of effect is dependent upon the type of fill, which is illustrated in **Fig. 9.** In the examples shown it is assumed that when the capillary and the head of the instrument are at 80 F, the instrument is on scale.

In a liquid-filled element, a decrease in the ambient temperature surrounding the aforementioned parts creates a gradual and continuous increase in the actual control point. Increasing the ambient temperature will cause the converse to happen. The degree of this change depends on the relative volume of the element subject to the controlled and to the ambient temperature.

In a vapor-tension fill of the cross-ambient type, the actual control point coincides with the desired control point except for a small span near the temperature at which the ambient temperature and scale setting are equal. Within this span the differential is apt to be increased and erratic operation may be experienced. This phenomenon is caused by the transfer of the liquid back and forth between bellows and capsule.

In the fadeout or limited type of vapor-tension fill, the actual control point coincides with the desired control point as long as the ambient temperature surrounding the head and capillary of the instrument is above the capsule or bulb temperature. When the former falls below the latter, complete loss of control will result as the small droplet of active fill leaves the capsule and enters the bellows housing; actually, then, the bellows becomes the temperature-sensitive part of the thermal system.

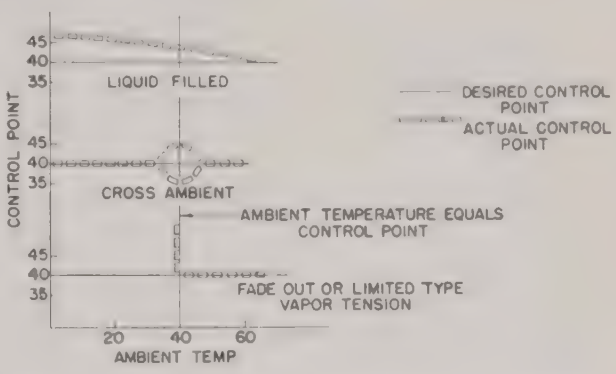

Fig. 9. Effect of Ambient Temperature on Controllers

Limit controls. A further and most important factor to consider when selecting automatic controls is the type of operation needed, controlling or high-limit (safety). In the former of these, quite frequent operation of the control mechanism occurs and any failure of the control is likely to be noticed before damage is done. In the latter the control may go for days, months, or possibly years without operation, and then

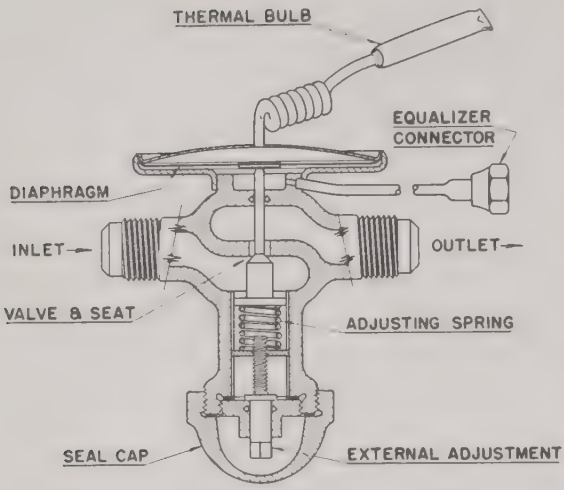

Fig. 10. Thermostatic Expansion Valve

because of some unsafe condition, which the control was installed to protect against, it will be required to operate immediately. At this time operation is a "must," or serious damage to equipment or even property might result.

D. Refrigerant-Flow Controllers

Expansion valves. One of the most important control functions within the refrigerating system is that of the admission of the refrigerant to the evaporator. An expansion valve is normally employed for this purpose except in the very smallest systems. Expansion valves fall into two general classes; (1) those which are operated by changes in the level of the liquid refrigerant in a chamber; and (2) those which are actuated by changes in pressure. This second class of valves has many of the same control characteristics that exist in other automatic control devices and will therefore be discussed in some detail.

Thermostatic expansion valves. A typical thermostatic expansion valve is shown in **Fig. 10.** It consists of a valve body containing the valve, which is positioned by the diaphragm and the adjusting spring. The position of the diaphragm depends on the difference in pressure above and below it and the force of the adjusting spring. The space above the diaphragm usually contains the same refrigerant which is used in the refrigeration system, and the pressure therein is determined by the temperature of the thermal bulb which is inserted in or strapped to the suction line of the evaporator being controlled. The pressure below the diaphragm is made the same as the pressure within the evaporator by means of the equalizer connection.

The amount of refrigerant which a thermostatic expansion valve allows to enter the evaporator will always be such as to keep the difference in pressure above and below the diaphragm equal to the force exerted by the adjusting spring. Since the pressure within the thermal bulb can be greater than the pressure in the equalizer connection only in an amount proportional to the superheat of the refrigerant in the suction line, these valves are sometimes referred to as constant superheat valves.

The action of the valve is to fill the evaporator with refrigerant containing some portion of unevaporated liquid, except for the relatively small distance near the suction side which is required to superheat the gas. Thus the evaporator is operated at high efficiency over a large range of load conditions.

For the best control with a thermostatic expansion valve, and the best use of the evaporator, the superheating region should be in contact with the warmest portion of the air or liquid being cooled. Where this is not possible or where the difference in temperature between refrigerant and the air or liquid being cooled is extremely small, a separate heat exchanger can be used to produce the superheat to operate the valve.

Where rapid changes in evaporator temperature or pressure are likely to occur some hunting may be experienced with thermostatic expansion valves. Hunting usually can be reduced either by slightly insulating the thermal bulb from close contact with the suction line or by the use of stiffer adjusting springs.

Automatic expansion valves. The automatic expansion valve shown in **Fig. 11** has its valve positioned only by changes in pressure in the evaporator. The pressure in the inlet side of the evaporator is transmitted through the pressure equalizer hole to the space below the diaphragm, and an increase in evaporator pressure tends to close the valve. The pressure under the diaphragm is opposed by the adjusting spring. The adjusting nut allows the selec-

tion of the evaporator operating pressure.

The diaphragm is usually subjected to the same temperature that exists in the evaporator, and moisture from the air must be prevented from coming in contact with the diaphragm by the flexible seal cap. In order for the automatic expansion valve to operate at pressures within the vacuum range, it must contain a vacuum spring in order to close the valve against atmospheric pressure.

The automatic expansion valve is used wherever it is desired to have a refrigeration system or evaporator operate at constant load. It should not be used where loads much lighter than normal might occur, since this would result in too great an amount of liquid entering the evaporator.

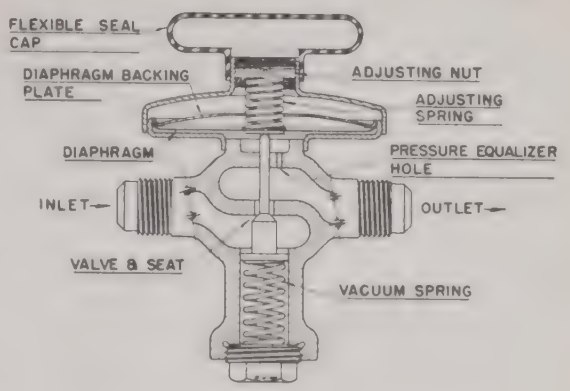

Fig. 11. Automatic Expansion Valve

The non-evaporated portion would be returned to the compressor as liquid and might damage the compressor.

Part II. Instruments*

THIS section deals with instruments used for test and other measurement purposes. The instruments described are primarily for indicating but some of them may also be used for recording. The section is intended to serve as a guide in the selection of measurement equipment. More complete details for the use of the instruments are available in the references.

Temperature Measurement

Table 1 shows the principal types of instruments used for temperature measurement, their applications, range, order of accuracy and limitations. Principally used in the ordinary temperature range are glass-stemmed thermometers, thermocouples and resistance thermometers, in that order.

While they are direct-reading and simple to use, **glass-stemmed thermometers** are subject to corrections for initial graduation, aging, pressure and depth of immersion.

Thermocouples are especially well suited for remote reading and of a number of stations. They are ideally suited for measurement of temperature of solids as well as of fluids. In a fluid circuit such as a refrigerating system, it is possible by the attachment of thermocouples on the surface of the pipe to determine the approximate temperature of the fluid within. Insulation of the pipe improves the accuracy of this method. In order to develop their full possible accuracy, thermocouples should be used with a potentiometer equipped with a standard cell and a cold junction held at constant temperature (or the equivalent electrical compensation). They should also use carefully standardized and calibrated wire. They may be used with lesser accuracy with a microammeter and high resistance in place of the potentiometer.

Thermocouples may be used in parallel to indicate the average temperature but it is important for this purpose that the wires be carefully equalized in resistance. They may also be used in series for the same purpose without the necessity of equalizing resistance. They are also sometimes used in series as a thermopile in order to increase the output voltage and to simplify the problem of measurement. Many times it is convenient to arrange the thermocouples for differential temperature measurements in such instruments as a pyrheliometer, heat-flow meter, etc.

Resistance thermometers are of especial value for remote readings of gases requiring a high degree of accuracy. They are useful where a grid of resistance wires can be used to indicate average temperature of a stream.

* This part of Chapter 22 was prepared by C. M. Ashley.

Table 1. Measurement of Temperature

No	Measurement Means	Application	Range, deg F	Order of accuracy, deg F	Limitations
1	Glass-stem thermometers				
	Mercury-glass thermometer	Temp of gases and liquids by contact	−38/575	Less than 0.1 to 10	In gases, accuracy affected by radiation
	Alcohol-glass thermometer	"	−100/100	"	"
	Pentane-glass thermometers	"	−200/70	"	"
	Jena or quartz mercury nitrogen thermomenters	"	−38/1,000	"	"
2	Gas thermometer	Primary standard	−459/1,000	Less than .01	Requires considerable skill to use
3	Resistance thermometers				
	Platinum-resistance thermometer	Precision; remote readings; temp of fluids or solids by contact	−320/1,800	Less than .02 to 5	High cost; accuracy affected by radiation in gases
	Nickel-resistance thermometer	Remote readings; temp by contact	−150/300	.3	Accuracy affected by radiation in gases
4	Thermocouples				
	Pt–Pt-Rh thermocouple	Standard for thermocouples	500/3,000	.1 to 5	High cost; also, requires expensive measuring device
	Chromel-alumel thermometer	General testing of high temp; remote rapid readings by direct contact	Up to 2,200	0.1 to 15	Less accurate than above
	Iron-constantan thermometer	"	Up to 1,500	0.1 to 15	Subject to oxidation
	Copper constantan thermometer	Same as above, especially suited for low temp	Up to 600	0.1 to 15	
5	Beckman thermometers (metastatic)	For differential temp in same applications as in glass stem thermometer	5 C diff	0.01 C	Must be set for temp to be measured
6	Bimetallic thermometers	For approx temp	0/1,000	1, usually much more	Time lag; unsuitable for remote use; unreliable
7	Pressure-bulb thermometers				
	Gas-filled bulb	Remote-testing	−200/1,000	2	Caution must be exercised so that installation is correct
	Vapor-filled bulb	"	20/500	2	"
	Liquid-filled bulb	"	− 50/1,200	2	"

Table 1. Measurement of Temperature (Continued)

No	Measurement Means	Application	Range, deg F	Order of accuracy, deg F	Limitations
8	Optical pyrometers	For intensity of narrow spectral band of high temp radiation (remote)	1,500 upward	15	
9	Radiation pyrometers	For intensity of total high temp radiation (remote)	Any range		
10	Seger cones (fusion pyrometers)	Approx temp (within temp source)	1,000/3,600	50	
11	Melting and boiling points of materials	Standards	All except extremely high temp	Extremely precise	For laboratory use only

Radiation effects. Gas temperature readings may be affected by radiation from surrounding surfaces at different temperature, particularly at high temperature levels. Glass-stemmed thermometers are particularly affected, due to the large bulb with correspondingly low convection transfer and to the high emissivity of glass. The use of thermocouples having a minimum wire size is preferred where radiation is a problem. The radiation effect may be minimized by applying directly to the couple a coating of low-emissivity metal such as aluminum, silver or platinum. Surrounding shields of low-emissivity metal are helpful where the gas velocity is appreciable. Complete correction of the radiation effect may be obtained by: (1) using an aspirating tube around the couple, reading temperature at a series of gas velocities and projecting to infinite velocity; (2) using a series of thermocouples of different sizes and projecting readings to zero diameter; (3) using two couples of known and different emissivity; and (4) using heated shields.

Sampling and averaging. Temperature is usually a point measurement, but it is frequently assumed representative of the value over an area as on a surface or in a flowing stream. However, such an assumption is rarely safe without an exploration. Where good accuracy is desired in the measurement of average temperature as of a stream, it is desirable to divide the stream cross-section down into a number of equal parts and measure the temperature at the center of each. Even the average of these readings may not be right if the velocity is not also uniform. While this can be compensated by multiplying the temperature by the velocity at each point and averaging, the more desirable solution is to find means of minimizing the variation. In drawing air from a room, the temperature variation can be minimized by drawing all of the air from the same level. In measuring temperature in a duct or pipe following a heat transfer element, it is desirable to use a mixer. While mixture may be obtained by turbulence alone, a method many times more effective is by transposition—the stream is divided down into parts and the alternate parts passed through each other. Mixers

of this sort may be arranged to re-mix fluid having almost any type of stratification.

Thermometer wells. Sometimes it is necessary to use thermometer wells in the measurement of fluid temperatures. The well should be made to fit the thermometer as snugly as possible and should be filled with a high conductivity liquid (oil, water or mercury if suitable) in order to cut conduction losses to a minimum. The well should be as long and thin-walled as practicable. The pipe adjacent to and including the well should be well insulated to prevent temperature difference which would affect the accuracy of the reading.

Effect of velocity. When thermometers are placed in a moving stream of fluid, the temperature indicated reflects a part of the energy of motion of the stream, usually in the range of 60 to 80%. At ordinary velocities this factor is negligible, but at higher velocities it becomes extremely important, and under these conditions it is necessary to design special temperature-measuring instruments.

Measurement of Relative Humidity

Table 2 shows the principal types of instruments used for the measurement of relative humidity.

Relative humidity is normally measured by the use of a **wet** and **dry-bulb thermometer,** the humidity being indicated on a psychrometric chart. Good accuracy requires the use of velocities of 1,000 to 1,500 fpm over the bulbs. In a room this is normally obtained by swinging a sling psychrometer by hand or better by means of an aspirating-type psychrometer. In a duct the air velocity is usually of the right order. If not, the reading can be taken in a special sampling duct which is adjusted to the proper velocity, or a velocity correction can be applied to the reading.

The wet bulb surrounded by a wick is wet with water close to the wet-bulb temperature and readings taken at frequent intervals until the temperature is stabilized. At high wet-bulb depressions the wet-bulb temperature may not be reached before the wick starts drying out. In this case the air can be measured as to temperature, drawn through a heat exchanger for cooling, the wet and dry-bulb temperatures measured and the original conditions calculated.

Below 32 F a thin film of ice may be frozen directly on the bulb. Because of the small wet-bulb depression in this region, great care must be taken in its measurement. The psychrometric chart over ice is to be used.

At sub-zero temperatures, **other means** must be used. Among these are: a cell whose electrical conductivity varies with humidity; a cell containing hygroscopic salt which is heated electrically to indicate dew point; a compressed air stream expanded adiabatically to the fog point; and dew-point indicators and hygrometers. These methods used for sub-zero temperatures are in general also usable in the normal temperature range, and the first two are particularly well adapted for recording.

Hygrometers are widely used for approximate indication of relative humidity for temperatures up to over 100 F. They are actuated by hygroscopic elements (usually hair). Their accuracy tends to be impaired by exposure to very high and low relative humidities and temperatures over 125 F. Periodic recalibration is desirable.

Pressure Measurement

Absolute pressure. Instruments for measuring absolute pressure are shown in **Table 3.**

Very low absolute pressures (25 to 5,000 microns of mercury) may be measured by means of the McLeod gage, the Pirani gage and electronic discharge types. In the **McLeod gage,** a known volume of gas is trapped and raised to a predetermined pressure at which its remaining volume indicates the initial pressure. While the accuracy of measurement is good, the instrument is not direct-reading since it must be carried through a series of operations to obtain the reading. Furthermore, it is not well suited to the measurement of total pressure where appreciable quantities of condensible vapors are present, since these tend to condense out during the compression of the gas. The **Pirani gage** measures the thermal conductivity of the gas; thus the accuracy of the measurement depends upon the knowledge of the composition of the gas whose pressure is being de-

Table 2. Measurement of Humidity

No	Measurement Means	Application	Range, deg F	Order of accuracy, deg F	Limitations
1	Wet-bulb thermometer with dry-bulb thermometer	Room of building, outside air, air moving in ducts; standard method	0 to 500	.1 to 1	Should be used in air stream moving 1,000 to 1,500 fpm or correction made. Unsuitable for recording at sub-freezing temp
2	Dew-point apparatus				
	Condensation type	Occasional laboratory work; not widely used	−180 to 200	2	Cumbersome, expensive, difficult to use with precision; unsuitable for remote use
	Fog type	Simple wide range method	−100 to +100	2	Series of readings necessary for determination
	Electrical conductivity type	Accurate measurement of dew point; suitable for remote use	0 to 200	1	Unusable below −20 F and below 15% rh or for any moderate humidity at temp under 0 F
3	Hygrometer (for relative humidity)				
	Hair gyrometer	For direct rh in air where motion is slight	0 to 100% rh −40 to 150 F	3% rh	Considerable lag; low sensitivity; adversely affected by temp above 125 F and rh below 20%; frequent calibration required when used at extremes of range
	Hygroscopic material other than hair	Same as above—also for high sensitivity	0 to 100% rh −40 to 150 F	2 to 5% rh	Frequent calibration required when used at extremes of range
	Electrical conductivity type	For remote use	"		Requires frequent calibration
4	Infra-red radiation	While usable at all temperatures it is uniquely applicable to extremely low temp			Requires the use of very costly equipment
5	Standard salt	For standards			
6	Chemical analysis				
7	Absorption and weighing				Cumbersome process

termined. Other gages working in this pressure range use a discharge of electrified particles impinging on a screen.

For absolute-pressure measurements in the range of 0.1 to 10 in. of mercury, an **absolute-pressure manometer** is a preferred method of measurement. Unfortunately, most of these instruments are not readily portable. In a somewhat higher pressure range (0.5 to 30 in. Hg), a **differential mercury manometer** is frequently used to measure gage pressure and corrected to absolute pressure by the use of a **barometer.** Barometers of the mercury manometer type are special absolute-pressure manometers designed for the measurement of atmospheric pressure. For good accuracy, manometers and barometers must be corrected for temperature.

In refrigerating systems, the pressure of the high or low-side is frequently determined by the measurement of the temperature of condensing or evaporating refrigerant. In obtaining this measurement, care must be taken to make certain that there is neither sub-cooling, superheat or non-condensible gas present.

Differential pressure. Instruments for measuring differential pressure are shown in **Table 4.**

Very low pressure differential can be measured by means of a **micro-manometer.** One form of this is a hook gage with micrometer adjustment and magnifying-means to increase the accuracy, distinctly a laboratory instrument.

For somewhat higher pressures in the range of 0 to 10 in. of water, an inclined **draft gage,** as illustrated in **Fig. 12,** is commonly used. While this instrument must be filled properly and leveled, it is generally suitable for use either in the laboratory or in the field, and is commonly used for air velocity measurements together with the **pitot tube.**

For higher pressure readings where good accuracy is required, a "U" or **well-type manometer** is available. It is generally used either with water or mercury as a working fluid. When a mercury manometer is used for the measurement of liquid pressure difference, care must be taken to see that the connecting lines are filled with liquid and correction for the liquid density and for temperature must be made.

For the measurement of medium to high pressures, **bourdon tube gages** are normally used. These are available in a wide variety of ranges and accuracy. They are quite subject to damage due to vibration, pressure pulsation or excess pressure, and for accurate work must be calibrated frequently against a standard, such as on a dead-weight tester.

Velocity Measurements

Instruments for measurement of velocity are shown in **Table 5.**

No completely satisfactory instrument is available for measurement of low air velocity (0–100 fpm). Several types of **hot-wire anemometers** have been designed, both directional and non-directional. Unfortunately, the accuracy of many of these is questionable in the low end of the range and they should, therefore, be used with great caution. A **swinging-vane type** anemometer is also available for use in this range.

Another useful device for very low velocities is to time the rate of movement of a smoke puff, feather or piece of lint. The **Kata thermometer** uses a very large bulb which is heated and then allowed to cool between two set temperatures, the rate of cooling in conjunction with the ambient temperature indicating the velocity. Unfortunately, it is subject to several errors and can be considered as approximate only. While the above methods are well suited to measurement of velocities in an open

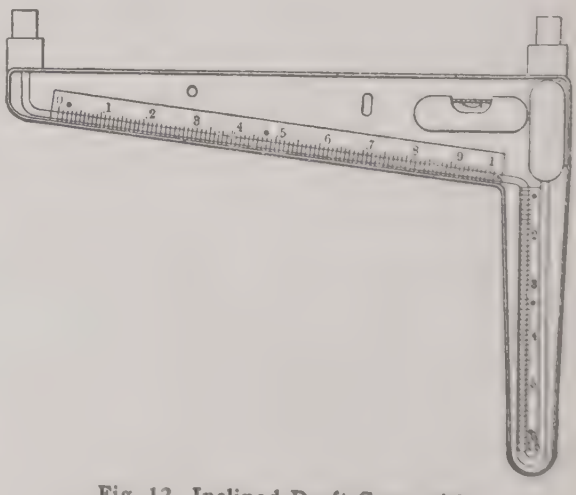

Fig. 12. Inclined Draft Gage with High-Range Leg

Table 3. Measurement of Absolute Pressure

No	Measurement Means	Application	Range	Order of accuracy	Limitations
1	McLeod gage	Very low abs press	0–0.1 in. Hg	2–5%	Not direct reading; vapors tend to condense out
2	Pirani gage	"	"	2–5%	Must be calibrated for gas composition on which used
3	Absolute-pressure manometer	Moderately low abs press	0–30 in. Hg or above	0.01 in.	Not readily portable
4	Barometer, mercury—manometer type	Atmospheric press		0.001–0.01 in.	"
5	Differential manometer and barometer	Moderately low abs press	0–60 in. Hg or above	0.01 in.	Awkward observation; requires two readings
6	Evaporating liquid temp	On refrig machine and for use during evacuation	Depends on liquid	1%	Liquid must be kept supplied; can only be used where liquid evap will not damage system

Table 4. Measurement of Differential Pressures (Gage)

No	Measurement Means	Application	Range	Order of accuracy	Limitations
1	Micromanometer	Very low press diff	0.000–6 in. H_2O	0.0005 in. H_2O	Not readily portable; not easy to use with pulsating pressure
2	Draft gages	Moderately low press diff	0.0–10 in. H_2O	0.005 in. 0.05 in. H_2O	Must be leveled carefully; needs calibration
3	Manometer	Medium press diff	0–100 in. H_2O or Hg	0.05 in.	Where used with liquid must be compensated for liquid density
4	Swinging-vane type gage	Moderately low press diff	0–$\frac{1}{2}$ in. 0–20 in. H_2O	5%	Generally usable to atmospheric pressure only
5	Bourdon-tube type	Medium to high press diff, usually to atmosphere	Any	As high as 0.01 psi or 0.01% of full scale	Must be calibrated; subject to damage due to over press-shock or pulsation

Table 5. Measurement of Velocity

No	Measurement Means	Application	Range of air, fpm	Order of accuracy	Limitations
1	Hot-wire anemometer	Low air velocities; directional and non-directional available	5–1,000	5–20%	Accuracy of some types not good
2	Kata thermometer	Low air velocities in rooms; non-directional	5–300	5–15%	Awkward to use; affected by radiation
3	Smoke puff or air-borne solid	Low air velocities in rooms; highly directional	5–50	10–20%	Awkward to use but valuable in tracing air movement
4	Swinging-vane type anemometer	Air velocities in rooms, at outlets, etc.; directional	30–24,000	5%	Not well suited for duct readings; needs periodic check calibration
5	Venturi-type multiplying pitot tube	Low air velocities in rooms and ducts; directional	100–2,000 with micromanometer 180–2,000 with draft gages	1–5%	Accuracy falls off at low end of range; requires calibration
6	Revolving-vane type anemometer	Moderate air velocities in ducts and rooms; somewhat directional	100–2,000	5–20%	Extremely subject to error with variations in velocities with space or time; easily damaged; needs periodic calibration
7	Pitot tube	Std instrument for measurement of duct velocities and pressures	180–10,000 with micromanometer 600–10,000 with draft gages 10,000 up with manometer	1–5%	Accuracy falls off at low end of range
8	Impact tube and sidewall or other static tap	Extremely high velocities, small tubes and where air direction may be variable	120–10,000 with micromanometer 600 10,000 with draft gages 10,000 up with manometer	1–5%	Accuracy depends upon constancy of static pressure across stream section

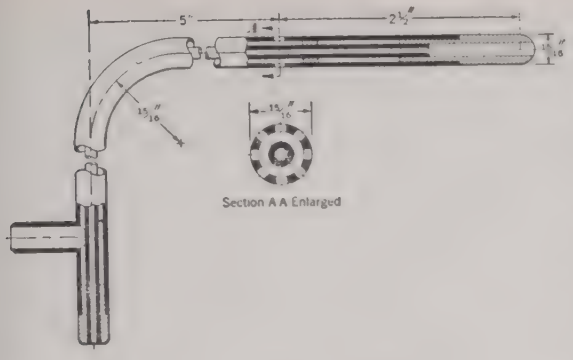

Fig. 13. Standard Design of Pitot Tube

space, some of them are not readily used in ductwork.

For observing velocities in free air in the range of 100 to 1000 fpm, the swinging-vane type of anemometer is probably as satisfactory as any other. The **revolving-vane type** anemometer is also extensively used in this range. However, these instruments are subject to errors as great as 30 to 40%, due to variations in direction and velocity of the air over the face of the instrument. Also, while they are made as a portable instrument, they are very subject to damage in handling. Both types of instruments are sometimes used for measuring velocities in ducts and plenums, but are awkward for this use since the instrument and the observer must be within the duct. Venturi-type multiplying **pitot tubes** are also available for measurement of velocity in this range. These have pressure multiplication factors as high as 10:1.

For relatively high velocities, the pitot tube is considered the standard instrument for velocity measurements. It may be used with a micromanometer, draft gage or "U" manometer, depending upon the velocity and the order of accuracy required. The order of accuracy at the low end of the range is poor but increases rapidly with velocity since the pressure varies as the square of the velocity. A standard form of pitot tube is shown in **Fig. 13.** A pitot tube is directional but is not extremely critical for direction. Where the static pressure of the stream varies with position, care must be used to see that the static holes are at the position for which the pressure and velocity are desired.

For high velocities, for use in very small tubes or where the direction of the air may be variable, velocity readings are frequently made using an **impact tube** together with a sidewall tap for static pressure. The impact tube is customarily a small tube used transversely of the stream. It is closed at the end and with a fine hole drilled close to its end on the upstream side. Static pressures may also be determined by revolving the impact tube through 180°, the true velocity pressures being obtained by multiplying the differential by a factor determined from calibrating tests. Static pressure may also be determined by means of a **Fechheimer tube** having two holes set at an equal and exactly determined angle to the air stream. This instrument, which must be made very precisely, is also subject to variation with velocity.

Volume or Flow Rate Measurement

Means of measuring flow rate are shown in **Table 6.**

Flow of gas and liquids in fair volume through ducts or pipes are most often made by the measurement of the pressure difference across an **orifice, nozzle** or **Venturi tube.** The orifice is more easily changed than are the nozzle or Venturi tube and is less affected by change of Reynolds number. The nozzle is preferred to the orifice by many engineers, due to its relative freedom from influence by approach conditions and the close approach of the nozzle coefficient to 1.0. The Venturi tube is in essence a nozzle with a recovery tail to reduce the pressure drop.

Standards have been established for orifices, nozzles and Venturi tubes covering the shape, surrounding conditions, location of pressure taps and values of discharge coefficient.

Liquid flow may be measured with good accuracy by **timing** a given **weight flow.** While this method is rather generally used in the test laboratory for calibration of other methods, it is particularly useful where the flow rate is low or intermittent and where a high degree of accuracy is required. A convenient direct-indicating flow meter for liquids is the **rotometer,** which comprises a transparent tapered tube in which the rate of flow is indicated by the position assumed by a bob or plug rotated by the liquid.

Table 6. Measurement of Volume or Flow Rate

No	Measurement Means	Application	Range	Order of accuracy	Limitations
1	Orifice and manometer	Flow through pipes, ducts and plenums—all fluids	Above Reynolds number of 5,000	1%	Coefficient and accuracy influenced by approach conditions
1	Nozzle and manometer	"	"	1%	"
3	Venturi tube and manometer	Same as 1 and 2 above but used where permissible pressure drop is limited	"	1%	"
4	Rotameters	Normally used for liquids	Any	1%	Must be calibrated for the liquid with which used
5	Timing a given weight flow	Liquids only—used for calibrating other flow means	Any	0.1%	—
6	Displacement meter	Relatively small volume flow at high press drop	As high as 1,000 cfm depending on type	0.2–2.0% depending on type	Some types require calibration
7	Gasometer or volume displacement	Short duration tests; used for calibrating other flow means	Total flow limited by available volume of containers	0.5–1.0%	—
8	Element of resistance to flow and manometer	Used for check where there is calibrated resistance element in the system	Lower limit set by readable press drop	1–5%	Secondary reading depends on accuracy of calibration
9	Thomas meter (temp rise of stream due to electrical heating)	Where elaborate setup is justified by need for good accuracy	Any	1%	Uniform velocity, usually used with gases
10	Heat input and temp change with steam or water coil	Check value in heater or cooler tests	Any	1–3%	—
11	Instrument for measuring point velocity	Primarily used in installed systems where no special provision for flow measurements have been made	Lower limit set by accuracy of velocity measurement	2–4%	Accuracy depends upon uniformity of flow and completeness of traverse

For measuring total liquid or gas flow over a period of time, a wide variety of **displacement meters** is available. The Thomas meter has been used in the laboratory for measuring of the flow rate of a large gas quantity with a small pressure drop. The gas is heated by electric heaters and the temperature rise is measured by two resistance thermometer grids. Where the flow of air through heating and cooling coils is being measured, one of the best check methods is through the determination of the heat output or input and the temperature change.

For field measurement of the air flow rate, a frequently used method is to make a **velocity traverse** of the **duct,** using for the purpose a pitot tube and draft gage or other velocity-measuring instrument. Since such an instrument measures the velocity only at a specific point in the stream and since the velocity usually varies from point to point, the accuracy of the reading will depend upon the number of elements into which the area of the duct is divided for determination of the velocity. **Fig. 14** shows typical points for velocity readings in a round and rectangular duct. The round duct readings are taken at the centers of area of equal concentric areas, whereas in the rectangular duct readings are taken in the center of equal square or slightly rectangular areas. Since the lower air velocity around the periphery of the duct is not measured, the air flow reading taken by this method is usually slightly high. In choosing a suitable location for such a traverse, it is desirable to have as long a straight section of duct as possible ahead of the traverse. Where the duct velocity distribution is poor, a greater number of readings should be taken.

Another approximate indication of flow measurement sometimes used for field readings is to take the **pressure drop** across elements having known flow-pressure drop characteristics, such as heating and cooling coils, fans, etc. Such data should normally be used only for check purposes, however, because the resistances of such elements may differ from rating data due to the surrounding conditions.

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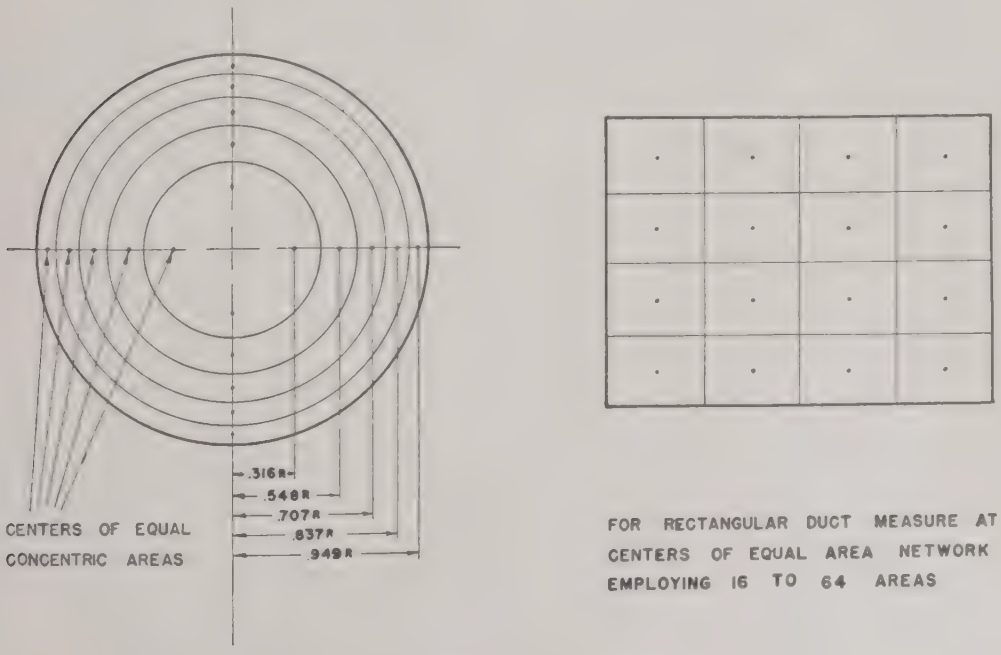

Fig. 14. Pitot Tube Traverse Points

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PART IV

INDUSTRIAL SYSTEMS

Associate Editor, H. B. Pownall, Chief Industrial Application Engineer, York Corporation, York, Pa.

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23. COMPRESSORS

Part I: Reciprocating Compressors

THE compressor is one of the five essential parts of the compression refrigeration system, along with the condenser, the expansion valve (or its equivalent), the evaporator and the interconnecting piping. It is the principal moving part, hence knowledge of its construction and behavior is of fundamental importance to the refrigeration engineer.

This chapter discusses the mechanical construction and design of reciprocating compressors. In the selection or design of a compressor, one must analyze the refrigerating cycle as a whole. Once the refrigerant has been chosen, operating temperatures and capacity determined, other features of the compressor follow as a result, including the required displacement, the operating pressures, the materials that can be considered for its construction and the power requirements. Chapter 2 treats of the theoretical cycle and compressor operating characteristics and efficiencies. Combined with this chapter, it gives a comprehensive discussion of the subject.

Classification

1. All reciprocating compressors are essentially alike, consisting of a reciprocating piston and suitable valves, yet their designs vary with numberless possible combinations of details. They may be divided into two main classifications covering the great majority of modern compressors:

a. Single-acting enclosed
b. Double-acting, with crosshead.

The distinctive features of single-acting enclosed compressors are pistons of the trunk type (i.e., driven directly through a pin and connecting rod from the crankshaft without piston rod or crosshead), and a pressure-tight crankcase.

2. Various **combinations** of the following **design features** are used:

a. Number of cylinders—two to as many as 16, sometimes one.
b. Cylinder arrangement—horizontal, vertical, *V*, *W*, or radial.
c. Drive—
    1. Electric motor with rotor mounted on crankshaft inside of crankcase, no moving parts outside.
    2. One end of crankshaft extending out of crankcase sealed with packing or ground joint type of shaft seal and driven by: (a) Electric motor with rotor mounted directly on shaft; (b) electric motor, steam engine or internal combustion engine coupled to shaft; (c) flywheel on shaft belted (V-belt, chain, gear drive) to motor or engine.
d. Lubrication—splash (sizes usually not over 15 hp), or force feed.
e. Suction and discharge valve arrangement—
    1. Uniflow—suction valves in top of hollow piston, with suction gas entering through cylinder walls, discharge valves in head (usual design for ammonia compressors).
    2. Same as (1) but with suction admitted through crankcase.
    3. Suction and discharge valves in head.
f. Type of suction and discharge valves—poppet, ring plate, or flexing.
g. Cylinder cooling—
    1. Water jacket: (a) cylinder wall only; (b) cylinder wall and head.
    2. Air cooled by fins.
    3. No cooling.
h. Head—
    1. Safety head, spring loaded, capable of being lifted off seat by foreign body or liquid.
    2. Bolted tight head.
i. Bearings—
    1. Sleeve.
    2. Anti-friction (usually main bearings only).

j. Capacity control, if provided—manual or automatic.
   1. Variable speed drive.
   2. Adjustable clearance: (a) movable piston in clearance cylinder; (b) one or more fixed-volume clearance pockets.
   3. Bypass from cylinder port (at appropriate fraction of stroke) to suction, preventing compression until covered by advancing piston.
   4. Means to hold suction valve of one or more cylinders open, thus putting them out of service.
   5. Bypass from individual cylinder heads to suction.
   6. Gas bleed from discharge to suction.
   7. Variable stroke.

k. Unloading or starting bypass—
   1. Bypass from discharge to suction with manual or automatic valve.
   2. Suction valves held open.

l. Speed—250 to 3,600 rpm, piston speed usually not over 600 fpm, with 750 as upper limit.

Double-acting compressors are always provided with a piston rod and crosshead. The rod works through a stuffing box in one of the cylinder heads so that the packing is on a reciprocating rod, as compared to the rotating shaft of the enclosed type. Double-acting compressors are built in both horizontal and vertical types. The valves are mounted in the heads or cylinder walls near the head.

In modern designs moving parts are all enclosed to prevent oil splash. Cylinder lubrication and bearing lubrication are separate. Cylinder lubrication may be by injection of oil into the suction gas, feeding oil through the cylinder wall, or a combination of both. Other design features follow those enumerated under single-acting enclosed compressors.

Design Procedure

Before starting the design of a compressor the desired capacity and operating conditions must be known and the refrigerant to be used must be selected. By assuming a reasonable volumetric efficiency, the required swept volume is computed. The type of compressor, whether single or double-acting, high-speed or low-speed, and the number of cylinders, must then be decided upon. It is a matter of simple computation to select the bore, stroke and speed. Average piston speeds are usually kept around 600 fpm.

2. Calculation of compressor swept volume is carried out as follows. The nomenclature is this—

h_s = *Enthalpy of vapor at temperature and pressure of compressor suction, Btu per lb
h_e = *Enthalpy of liquid to expansion valve, Btu per lb
M = Capacity in tons refrigeration
v = *Specific volume of vapor at compressor suction, cu ft per lb
V_s = Compressor swept volume, cfm
e_v = Volumetric efficiency, %/100

$$V_s = \frac{200\ Mv}{(h_s - h_e)e_v}$$

3. Suction and discharge valves. The limiting factors of speed are the valve area that can be accommodated and the gas velocities permissible without excessive wiredrawing losses. The actual gas velocities vary for different points in the stroke. For design purposes, gas velocity = bore area × average piston speed/valve area. The permissible gas velocity through the restricted area of the valve will depend upon the loss in volumetric efficiency and the excess power requirements the designer is willing to take. In general, with ammonia, velocities up to 12,000 fpm and with Freon-12, velocities up to 6,000 fpm result in no material loss in volumetric efficiency or increase in horsepower. Restricted valve velocities of 20,000 fpm are not unusual. The restricted valve area is not the only factor—the shape of the valve and adjacent ports determines the number of directional changes to the gas flow and affects the efficiency of the conversion of velocity head to pressure head and vice versa, which all affects the pressure drop through the valve.[1]

The ideal valve would meet the following specifications:

* The numerical values of the items starred are to be looked up in tables of thermodynamic properties of the refrigerant.

1. Largest possible restricted area, of the shortest possible length
2. Straight gas flow—no directional changes
3. Light weight combined with low lift, for quick action
4. Minimum unbalance
5. No harmful clearance
6. Rugged
7. Inexpensive
8. Tight seating.

Plate valves should have not over 0.075 in. lift. Flexing valves should have their lift restricted, to keep the stresses well below the endurance limit of the steel. Fuchs, Hofmann and Schueler tested plate valves with water and with air, finding that the friction coefficient was materially less with low lifts than with high lifts.[2]

4. The gas **pressure forces** on the piston for various positions, as well as the inertia forces, can be readily computed from **Table 1**, with the following nomenclature:

a = adiabatic factor
A = area of cylinder bore, sq in
A_c = area of cylinder bore, less rod, sq in
d = shaft diam, in.
D = belt wheel diam, ft
E = energy, ft lb
F = total force, lb
F_i = inertia force, lb
F_h = head end force, lb
F_c = crank end force, lb
F_g = net gas force, lb
I = inertia factor
K = tangential factor
k = speed fluctuation, about .02
m = bending moment
N = speed, rpm
P_s = suction pressure, psia
P_d = discharge pressure, psia
R = crank radius, ft
S = stress, psi
T = torque, lb ft
T_e = equivalent torque, lb ft
W_w = wheel rim weight, lb
W = weight of piston, rod, crosshead and connecting rod, lb

The effective connecting rod weight taken into account under W is calculated by multiplying the total rod weight by the distance from the center of gravity to the crank, and dividing by the distance from the crank to the pin. It should be noted that a double-acting piston is doing work on only one face at a time.

For double-acting compressors when

$$P_s a \leqq P_d,\ F_h = AP_s a \text{ and } F_c = A_c P_s a$$

when

$$P_s a > P_d,\ F_h = AP_d \text{ and}$$
$$F_c = A_c P_d$$
$$F_g = F_h - F_c \text{ or } F_g = F_c - F_h^*$$

For single-acting compressors when

$$P_s a \leqq P_d,\ F_g = AP_s(a-1)$$

when

$$P_s a > P_d,\ F_g = A(P_d - P_s)$$
$$F_i = 0.00034\ WN^2RI$$
$$F = F_g + F_i \qquad T = FKR$$

If T is plotted against crank angle in degrees, a curve such as **Fig. 1** results. The mean torque is the area under the curves divided by the length of the diagram.

E = ½ area a-b-c-d-e (expressed in lb deg ft) $\pi/180$

$$W_w = \frac{11{,}744E}{kD^2N^2}$$

For direct-connected motor applications, the crank effort diagram and allowable current pulsation may be submitted to the motor manufacturer.

5. Calculation of **synchronous motor** wr^2 inertia may be carried out with the following nomenclature:

W = Rotor weight, lb
P_r = Synchronizing power, kw per electrical radian
f = Frequency, in cycles per second
C = Compressor factor from **Table 2**
pf = Power factor
hp = Horsepower
e = Motor efficiency

$$\text{Motor input, kva} = \frac{0.746 \text{ hp}}{e \times \text{pf}}$$

P_r = 1.7 × motor input kva (at 100% pf)
= 2.0 × motor input, kva (at 80% pf)

$$Wr^2 = \frac{1.34P_r fC}{(N/100)^4}$$

r = Rotor radius of gyration, ft

6. The maximum **crankshaft** stress must be evaluated treating the shaft as a beam in

* Note that on suction stroke $F_h = AP_s$ and $F_c = A_c P_s$.

torsion. For each piston there will be a force F, as determined in Par. 4, acting in line with the stroke. The bending moment on the shaft is determined in both the vertical and horizontal plane from the various forces, the belt pull, wheel weight, etc. The two moments are then combined:

$$m = \sqrt{m_1^2 + m_2^2}$$

The torque is then determined at the point where m is a maximum:

$$T_e = m + \sqrt{m^2 + T^2}$$

$$S = \frac{16 T_e}{\pi d^3}$$

Sharp corners should be avoided in crankshafts to minimize stress concentrations. The stress should be kept well below the endurance limit of the shaft material. A factor of safety of not less than 3, based on the endurance limit, is suggested.

Since shaft failures are usually due to

Table 1. Constants for Constructing Crank Effort Diagrams

Crank angle	Piston position	Inertia factor I	Tangential factor K	Adiabatic factor, ammonia a	Adiabatic factor, F-12 a
0	.0000	1.200	0	1	1
10	.0091	1.173	.2079	1.012	1.01
20	.0360	1.093	.4065	1.049	1.042
30	.0796	.966	.5870	1.114	1.111
40	.1317	.801	.7421	1.201	1.173
50	.2081	.608	.8657	1.354	1.302
60	.2878	.400	.9540	1.534	1.445
70	.3735	.189	1.0052	1.837	1.696
80	.4622	−.014	1.0199	2.24	2.016
90	.5505	−.200	1.000	2.829	2.469
100	.6358	−.361	.9500	3.718	3.131
110	.7151	−.495	.8743	5.357	4.132
120	.7878	−.600	.7781	7.503	5.765
130	.8509	−.677	.6664	11.87	8.589
140	.9038	−.731	.5435	20.98	14.09
150	.9455	−.766	.4130		
160	.9757	−.786	.2776		
170	.9939	−.797	.1394		
180	1.0000	−.800	.0000		
190	.9939	.797	.1394	1.008	1.007
200	.9757	.786	.2776	1.032	1.028
210	.9455	.766	.4130	1.076	1.066
220	.9038	.731	.5435	1.141	1.121
230	.8509	.677	.6664	1.234	1.200
240	.7878	.600	.7781	1.363	1.309
250	.7151	.495	.8743	1.546	1.461
260	.6358	.361	.9500	1.802	1.668
270	.5505	.200	1.000	2.173	1.963
280	.4622	.014	1.0199	2.727	2.392
290	.3735	−.189	1.0052	3.598	3.043
300	.2878	−.400	.9540	5.049	4.085
310	.2081	−.608	.8657	7.696	5.893
320	.1317	−.801	.7421	13.95	9.882
330	.0796	−.966	.5870	26.84	17.46
340	.0360	−1.093	.4065		
350	.0091	−1.173	.2079		

Crank angles from beginning of stroke towards crankshaft; rod length/crank radius = 5; γ for ammonia = 1.3, γ for Freon-12 = 1.13.

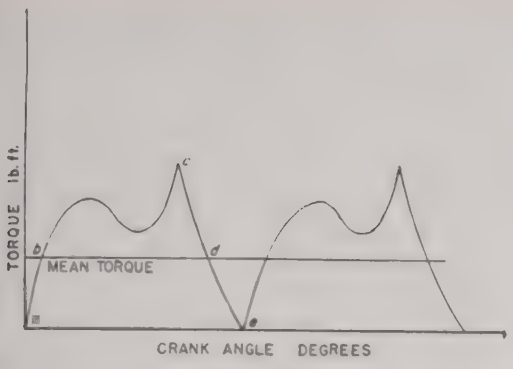

Fig. 1. Crank Effect Diagram for 2-Cylinder, Vertical s.a. Compressor

fatigue, it is suggested to design on the basis of the maximum shear theory. In most compressor designs, assuming the torque as constant at the average value and assuming the maximum bending moment as reversed introduces no significant error. For this condition Mark's Handbook gives the following equation for the shaft diameter:

$$d = \sqrt[3]{\frac{32F_s}{\pi}\sqrt{\left(\frac{Km}{S_e}\right)^2 + \frac{T^2}{Sy^2}}}$$

Where F_s = factor of safety (usually 2 to 3)
K = stress concentration factor in bending which depends upon the relation of the fillet to the height of the shoulder as well as upon the notch sensitivity of the material. With liberal fillets K should not exceed 2.5.
S_e = endurance limit of the material in reversed bending.
S_y = yield stress in tension.

Higher speeds, the elimination of massive foundations, and a more critical public combine to make balancing of crankshafts necessary. Mathematical treatment of this subject can be found in texts on the subject of vibration. (Also see Aldinger and Neubauer, *Refrig. Eng.*, Feb. 1941.)

Shafts have in general been made of forged steel. Some designs call for hardened bearing surfaces to reduce wear. Alloy cast iron such as Meehanite or Gunite is being extensively used for this purpose. It is generally lower in cost than forgings. It has lower notch sensitivity and superior damping qualities. The material has excellent wear resistance in journals but must be highly polished to avoid scratching of the bearings.

7. Bearings. Wristpin bushings are of bronze or cast iron, with pressures up to 1500 psi. The wristpins themselves are case-hardened steel. Crankpin and main bearings are generally white-metal alloys, although anti-friction bearings are also used. On enclosed compressors, thrust bearings must be provided to counteract the force due to crankcase pressure acting

Table 2. Compression Factors for Case in Design[3,4]

Application number	Description	Pulsation, %		
		66	40	20
A.	2-cyl vertical double-acting with 180° cranks for air, Freon, or ammonia. *Full load:* Both cylinders working normally. ¾ *load:* Suction valves lifted on head of one cylinder. ½ *load:* Suction valves lifted on two crank ends, head ends working normally. ¼ *load:* Suction valves lifted on 2 crank ends and one head end. *No load:* All suction valves lifted.	23.0	30.0	
B.	2-cyl vertical double-acting with 180° cranks for air, Freon, or ammonia. *Full load:* Both cylinders working normally. ½ *load:* Suction valves lifted on two crank ends, head ends working normally. *No ¾ or ¼ load.*	24.0		
C.	2-cyl vertical single-acting with ¾-load operation by opening 50% bypass on one cylinder for Freon or ammonia.	14.5	20.0	28.0

on the shaft area at the stuffing box. Table 3 gives data on white bearing metals. Practice goes considerably below the values given. In enclosed machines, crankpin pressures are generally held below 600 and main bearing pressures below 500 psi of projected area. The trend is away from bearing adjustments and toward precision bearings which require no fitting or scraping. Older practice usually made main bearings 2 to $2\frac{1}{2}$ times the diameter with bearing pressures of about 250 psi. Shorter bearings of length not much greater than diameter will carry higher unit loads.

The parts of the **cylinders and crankcase** are cast of dense high-strength iron, and are carefully tested under hydrostatic and air pressure. Pistons are given a diametral clearance of about 0.001 in. per in. of diameter with modifications for piston material. For high-speed precision-built compressors, the diametrical clearance is further modified for temperature variation of the piston, i.e., greater clearance at the top skirt than the bottom skirt. The crankcase or **frame** supports the shaft bearings and carries the cylinders. It should be checked for stiffness to prevent weaving. Cylinders and crankcase should be hydrostatically tested to two times working pressure and tested with air under water to one and a half times working pressure.

8. Shaft packing. Ammonia compressors are easy to pack, and various soft as well as metallic shaft packings are considered adequate. Freon-12, however, is very difficult to hold, and shaft seals which depend upon a ground seal joint had to be developed. The bellows type is limited to shafts about $2\frac{1}{2}$ in. in diameter and smaller, as it is difficult to secure flexibility of the bellows in large sizes. Some of the other seals can be built for large shafts. The shaft seal, born of necessity for Freon-12 compressors, is now displacing conventional packings on ammonia compressors.

9. Lubrication. Large compressors are

Table 3. Field of Usefulness for Various Bearing Metals[5]

Description of bearing metal	Maximum permissible unit pressure	Minimum permissible Zn/P_{max}	Maximum $P_{max}\,V$	Oil reservoir temp, F	Minimum crankshaft hardness	Affected by corrosion?
Tin Base Babbitt						
Copper 3.50%						
Antimony 7.50%	1,000	20	35,000	235	Not important	No
Tin 89.00%						
Lead (max) 0.25%		Standard quality bearings				
Tin Base Babbitt						
Same composition as above	1,500	15	42,500	235	Not important	No
		Alpha process quality bearings				
High Lead Babbitt						
Tin 5 to 7%						
Antimony 9 to 11%	1,800	10	40,000	225	Not important	No
Lead 82 to 86%						
Copper (max) 0.25%						
Cadmium-Silver						
Silver 0.75%	Over 1,800					
Copper 0.50%	and up to	3.75	90,000 and upwards	260	250 Brinell	Not likely if temperature is maintained as specified and proper lubricating oil is used
Cadmium 98.75%	3,500					
Copper-Lead						
Copper 60%	Over 1,800	3.75	90,000 and upwards	260	300 Brinell	
Lead 40%						

Z = Absolute viscosity of the lubricating oil in centipoises. *Data by Albert B. Willi, Federal Mogul Corporation.*
n = Revolutions per minute.
P_{max} = Maximum load in psi of projected area.
V = Rubbing speed, fps.

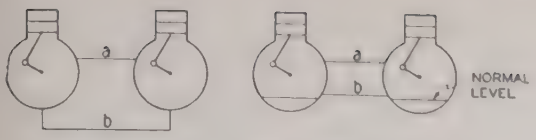

Fig. 2 (Left) and Fig. 3 (Right). Crank Case Oil Equalizing

today all provided with force-feed lubrication; small ones are splash lubricated. The dividing line is usually at about 15 hp. Oil pumps are of the positive displacement type, driven from the crankshaft. Some designs place the pump on the end of the shaft; others submerge it below the oil level. Oil circulating systems are provided with filters.

Enclosed compressors pump some oil. With Freon-12 and methyl chloride the quantity pumped is quite large; it is therefore necessary that it be returned to the crankcase automatically. With a number of compressors in parallel, the returned oil must be properly apportioned between the machines. **Figs. 2** to **5** illustrate various methods of apportioning the oil.

Fig. 2 shows the simplest kind of crankcase oil equalizing system. Line a connects the top of the crankcases to equalize the pressures, and line b connects the bottoms of the crankcases. The oil should have a common level in the two machines. This method is unreliable. A pressure difference of 0.02 psi between the two machines will cause difference in level of about 1 in. Even on the large compressors, a fluctuation of only a few inches is permissible. The very close pressure equalization is difficult to maintain. **Fig. 3** shows a modification of Fig. 2 in that the line b is now at or above the normal oil level. The oil cannot all be forced out of one crankcase through the equalizing connection as only an excess above line b can get across. If, however, the oil is pumped out of the crankcase at the higher pressure, oil cannot

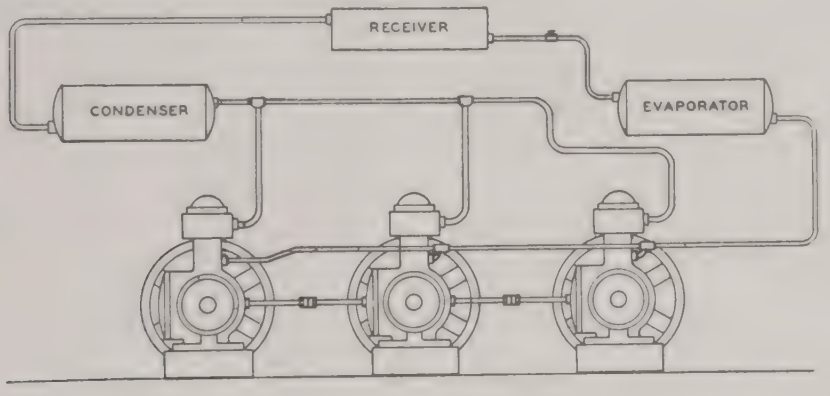

Fig. 4

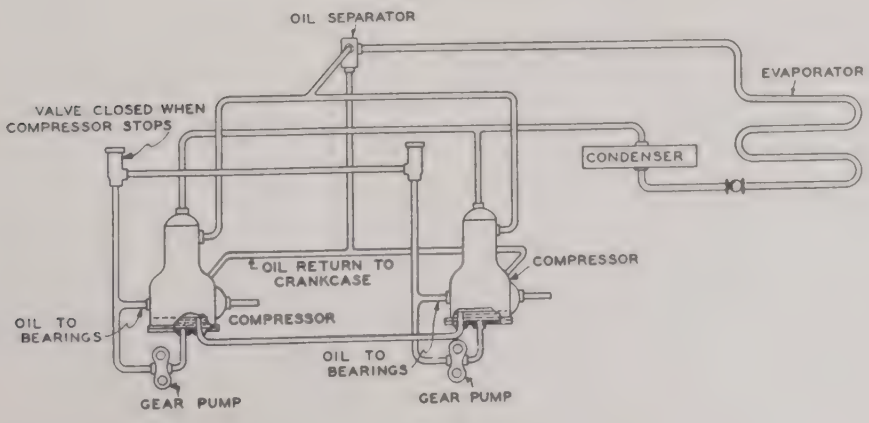

Fig. 5

flow from the other against that pressure, so this system is not certainly reliable.

Fig. 4 is a modification of Fig. 3 in that a double-acting check valve prevents flow through the oil equalizing line when a material pressure difference exists between machines. Fig. 5 has equalizing connections as in Fig. 3. In addition, the lubricating pump discharges are cross-connected so that if one pump ceases to function due to oil level failure, the pump from the other compressor supplies oil to the compressor whose pump has failed. The electrically operated check valves prevent oil from being pumped into a compressor not operating. **Fig. 6** shows a central reservoir for collecting return oil, and level control means to feed oil to the various crankcases.

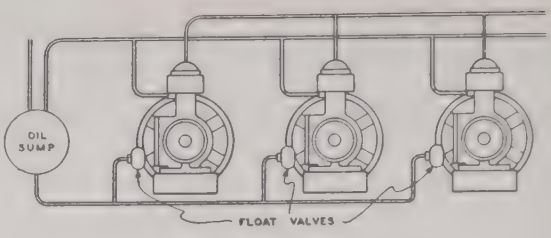

Fig. 6

Starting

10. Ammonia, Freon-12, and methyl chloride compressors that are to start under load are driven by motors having about 225% starting torque. Ammonia compressors, provided with starting bypasses having not more than 20,000 fpm gas velocity at full speed, are successfully started by synchronous motors having 40% starting and 40% pull-in torque.

Freon-12 compressors when unloaded require from 40 to 60% starting and 50 to 75% pull-in torque. Tests on a two-cylinder single-acting compressor, $7\frac{3}{4}$-in. bore $\times$ 5-in. stroke, equalized at 90 psig at the start with a $2\frac{1}{2}$-in. bypass, required 55 lb ft starting torque, 52 lb ft at 400 rpm, and 60 at 450 rpm, whereas with a $1\frac{1}{4}$-in. bypass the values were 55, 80, and 97.

11. The **foundation** should extend down to a good solid footing. It may serve to keep various elements, such as bearings, in line, and has the added function, by virtue of its mass, of keeping the amplitude of vibrations small. The heavier the foundation, the smoother will be the operation of the compressor. The vibrational forces are a function of the unbalanced masses and the speed. The size of foundation needed, therefore, depends upon the degree of unbalance. Well-balanced machines can be operated without even being fastened to the floor and with almost no perceptible motion. Only on recently developed high-speed compressors has there been any attempt at balancing. **Fig. 7** gives an average value of concrete foundation volumes for enclosed ammonia compressors ranging from 3×3-in. to 12×12-in. at speeds of from 500 rpm for the 3×3-in. to 300 rpm for the 12×12-in. These foundations are for machines without balance weights.

To avoid transmitting vibrations, foundations should not be tied into building or column footings. To absorb vibrations, machinery may be mounted on cork, rubber, etc., or on springs. For machines having considerable unbalance, better results will be obtained by mounting the machine on a heavy foundation and carrying the foundation on a resilient mounting

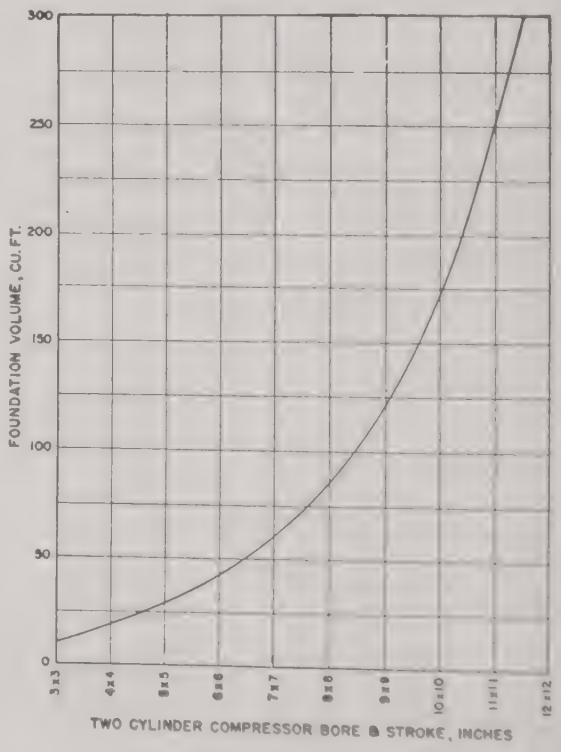

Fig. 7. Foundation Size

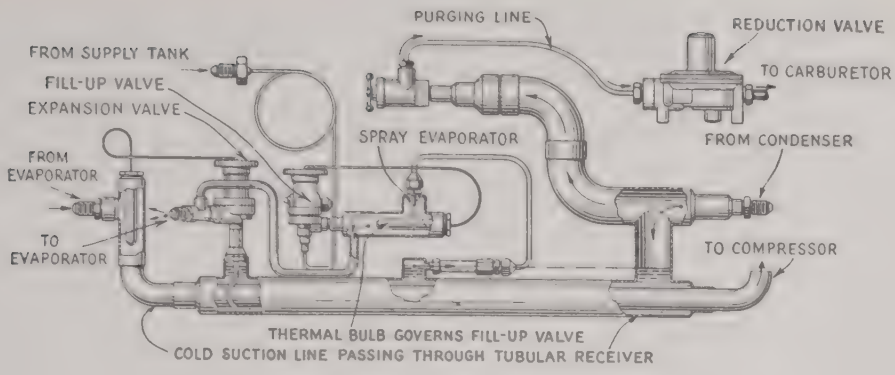

Fig. 8. Propane Unit Equipment

One such mounting can be made by pouring the foundation concrete into a pit lined with corkboard.

Resilient mountings should have a natural frequency much slower than the frequency of the disturbing force (the vibrational forces of the machine). The ratio of disturbing force to natural frequency of the mounting should be at least 1.41, and preferably 4 or more.

Natural **frequency** of the resilient support, in cycles per minute $= 188(1/S_d)^{1/2}$ where S_d is the static deflection, in inches.[9] This represents free motion without damping, such as would result from a spring rubbing in its guide. Damping minimizes the bad effect of operating close to natural frequency and somewhat reduces the isolation efficiency for larger frequency ratios. It may be noted when bringing machines mounted on vibration isolation up to speed, they will shake violently in passing through a certain speed range. That is when the impressed vibrations are close to the natural frequency of the support.

The suction and discharge pressures, or unloading, have no effect on the machine vibration (except the vibration caused by the speed fluctuation of the rotating masses), because the gas pressure forces are balanced out, within the machine. The pressure on the piston imposes a force through the rod to the shaft and bearings into the machine frame. This force is balanced by one of like magnitude on the cylinder head, which is carried back to the frame through the cylinder walls. Unbalanced moving parts are the principal cause of vibration.

The completed design of a compressor is a summation of compromises. Valve area may be sacrificed to keep the clearance within practical limits, the cylinder block design is modified to simplify cleaning, the crankcase is made bigger to give a larger oil reservoir, piping is changed to improve appearance, and weaknesses shown on test must be corrected. The finished result is more than likely the composite product of the best minds of the manufacturer's staff in its effort to provide the best efficiency, reliability and appearance in a compact form, to sell at a reasonable price. If there were one proved best possible compressor others would not be built.

On vertical single-acting machines the clearance between the piston and cylinder head is ordinarily held at about 1/32 in. on all sizes above 4-in. stroke. On a $1\frac{3}{4}$-in. stroke machine the clearance may be as little as .010 in. Horizontal double-acting machines may be set with a little greater clearance. These values may have to be modified where the design is such as to give unusually great rod expansion under operating temperatures. The clearance should be held as close as practical and yet avoid the possibility of striking the head under any operating condition.

Propane Unit

12. A special application of the refrigeration principle in unison with the use of the refrigerant as a motor fuel has been developed by Schlumbohm, using propane.*

* U. S. Patents 1,935,749; 2,082,850; 2,195,387-388-389. There are foreign patents in more than 14 countries.

The cycle is called transitory as the refrigerant passes through the evaporator only on its way to being used in the driving motor, along lines indicated in **Fig. 8.** Propane makes both an excellent fuel and refrigerant.

The carburetor of the engine is connected to the condenser side of the refrigeration circuit. Fueling the engine with condenser gas has the advantage of purging the condenser of non-condensible gases. The condenser is open and connected by the purging line with the carburetor of the propane combustion engine. This purging line is characterized by an ascending tube branching off from the condenser receiver and equipped with baffles to retain oil. There is also a pressure reduction valve, and a fuel regulator, which opens only in response to the vacuum of the engine-intake and which shuts off, if the engine stops.

The low-side is connected with a propane bottle by the fill-up line, which is controlled by two valves: a shut-off valve, which is kept open only if the generator supplies the current and closes if the motor stops, and the "fill-up valve," designed to control the level of liquid in the receiver and to respond to presence of liquid propane at that level. A spray line begins at that chosen level and ends in a secondary evaporator with a very small opening, through which a spray of liquid propane will expand into the secondary evaporator and will produce a cold effect if there is liquid propane at that level. If this is not the case, the spray will run dry and no cold effect will be produced. This cold effect and its cessation are utilized to close or to open respectively the fill-up valve. This valve is a standard expansion valve with the one side of the diaphragm exposed to the evaporator pressure, while the other side is exposed to the pressure of the thermal bulb. The bulb in turn is cooled by the secondary evaporator if there is sufficient liquid propane in the receiver. The fill-up valve is kept closed by this cooling effect. Consequently, lack of liquid propane in the receiver will open the fill-up valve, so that the compressor may take in vapors from the propane bottle until the level in the receiver is filled up by the fresh condensate, which then delivers the spray to produce the cold effect and to close the fill-up valve. A pint of propane is sufficient to circulate in a unit of 1.5 ton capacity.

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Part II. Centrifugal Compressors

1. The centrifugal refrigeration cycle. The centrifugal refrigeration machine, as commercially available in standard sizes. consists of a centrifugal **compressor,** horizontal shell-and-tube **condenser,** horizontal shell-and-tube **evaporator** (water or brine cooler), all interconnected with a high-pressure refrigerant flow **control valve** or other similar means of regulating the flow of refrigerant from the condenser to the evaporator. They are complete unit refrigerating machines arranged for direct drive to either a steam turbine or through speed increasing gears to motor, gas engine or Diesel drives. The refrigeration cycle is the compression cycle with the exception that interstage liquid cooling is incorporated as standard with the machines, and is simply accomplished by flashing liquid refrigerant at condensing temperature to one or more of the intermediate stages, thereby subcooling the liquid before it is supplied to the evaporator. Single-stage cooling is common to most water cooling units, and one- or two-stage cooling may be applied for low-temperature work.

2. The compressor. Centrifugal compressors are similar in many characteristics in design and construction to centrifugal pumps. Erosion or wear does not occur at points of clearance of the impeller and the casing labyrinth as in the centrifugal pumps. No valves, piston rings, or cylinder walls are required as in the reciprocating compressors. Efficiency of the centrifugal compressor is sustained for the life of the equipment. There are no practical pressure limits for the centrifugal compressor, except the limitation of mechanical design and volume flow. The pressure produced depends on the peripheral speed of the wheel, the density of the refrigerant handled, and the number of wheels. It is essential that the centrifugal compressor run at high speeds to compress relatively light fluids such as a gaseous refrigerant as compared with a centrifugal liquid pump. The maximum peripheral speed of the compressor is fixed by the mechanical strength of the materials and the sonic speed found in the refrigerant. Centrifugal compressors are essentially for heavy-duty work, continuous operation and where requirements are for large horsepower. The centrifugal compressor is inherently a constant-pressure machine, designed to give capacity variations and operates with little or no vibration.

Table 4 illustrates an important feature of the centrifugal. Its partial load performance is exceptionally good, which is unusual for machinery in general.

Direct shaft connection is made through a suitable coupling. Speed increasing gears are utilized with electric motors and engines. Constant speed motors, syncronous or induction, may be combined with magnetic or hydraulic slip coupling for variation of compressor speed, either manually or automatically controlled.

3. Refrigerants. Normally, refrigerants in the centrifugal compressor have high molecular weights; however, centrifugal compressors have employed the following refrigerants ("Carrene No. 2" (Freon-11) is a common refrigerant used because of its low pressure and high cycle efficiency):

Ammonia (NH_3)
Dielene (CH_2Cl)
"Carrene No. 1" (CH_2Cl_2)
"Carrene No. 2" (CCl_3F)
Freon-12 (CCl_2F_2)
Freon-113 ($C_2Cl_3F_3$)
Isobutane (C_4H_{10})
Freon-114 ($CClF_2$)
Methyl chloride (CH_3Cl)
Ethylene (C_2H_4)
Butane (C_4H_{10})
Propane (C_3H_{10})
Sulfur dioxide (SO_2)
Water vapor (H_2O)

4. Performance. Centrifugal refrigerating compressors have been built with efficiencies of well over 80%, but most commercial refrigerating compressors of today operate with efficiencies of between 70 and 80%. Centrifugal refrigerating compressors used for water cooling for industrial

Table 4. Performance of Centrifugal Refrigerating Machines Applied to Varying Industrial Loads

Load	%	100	90	80	70	60	50
*Temperature head	%	100	96	92	88	84	81
Speed	%	100	94	90	88	86	84
Bhp	%	100	80	68	59	49	51
Bhp/ton	%	100	89	85	84	82	82
Kw input	%	100	86	76	67	57	49
Kw input/ton	%	100	95	95	95	96	98

* Leaving chilled water and entering condenser water temperatures are constant; gpm of chilled water and condenser water are also constant.

purposes or water cooling for air conditioning applications usually consist of two stages using two single-inlet impeller designs, although some are designed for three-stage operation with three angle-inlet impellers for this type of application. Temperature of chilled water leaving the evaporator is as low as 34 F, and these machines are capable of operating with good efficiencies to high condensing temperatures; thus the differential between evaporating temperature of 25 F (for 34 F water) and 110 F condensing temperature (ordinarily obtained with as high as 95 F entering condenser water) is 85 degrees. This differential of 85 degrees has no particular significance other than to give an indication of the temperature lift at ordinary levels. The ratio of compression, however, for Freon-11 (CCl_3F) between these limits is expressed as follows:

P_2 at 110 F = 28.09 psia
P_1 at 25 F = 4.92 psia

$$\frac{28.09}{4.92} = 5.7$$

$\sqrt{5.7} = 2.40$ per wheel (approx) for two wheels.

For normal water cooling to 45 F (35 F evaporating temp) leaving chilled water temperature, and with 100 F condensing temperature obtained with 85 F entering condenser water the compression ratios are:

P_2 at 100 F = 23.60 psia
P_1 at 35 F = 6.26 psia

$$\frac{P_2}{P_1} = \frac{23.6}{6.26} = 3.8 \text{ or } \sqrt{3.8} = 1.95$$ (approx) per wheel for two wheels

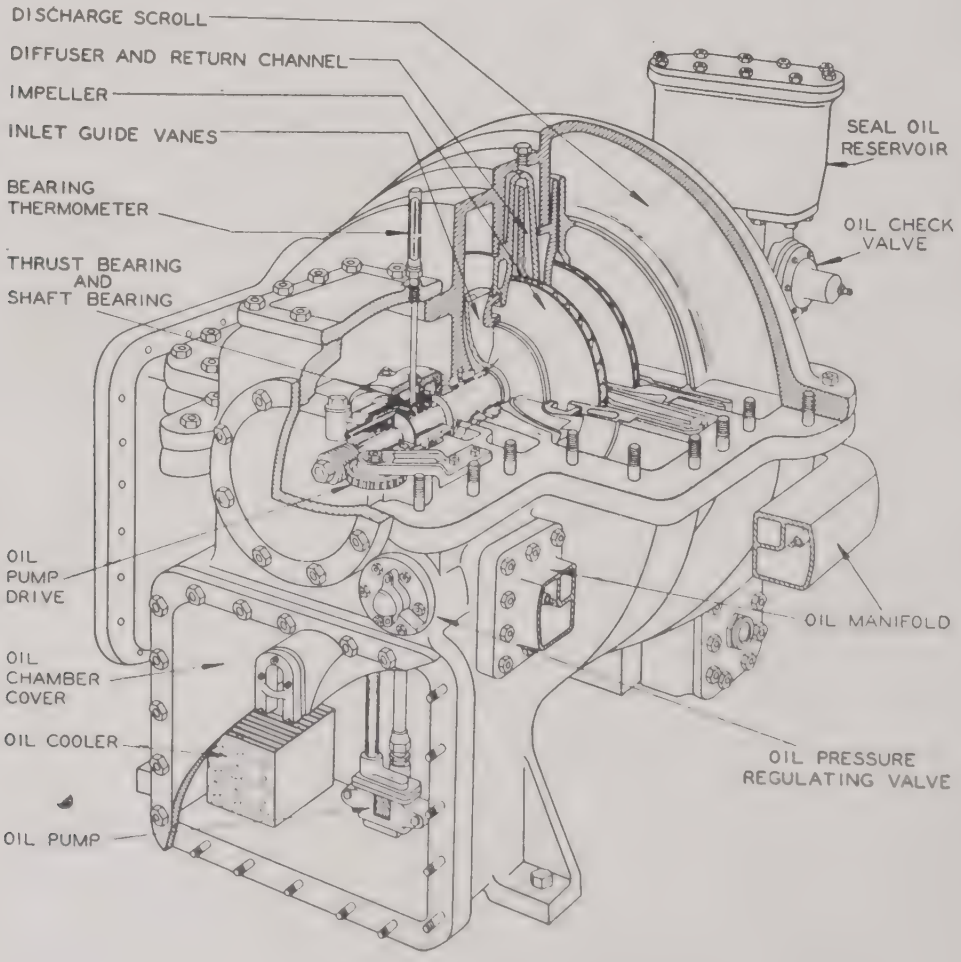

Fig. 9. Centrifugal Compressor

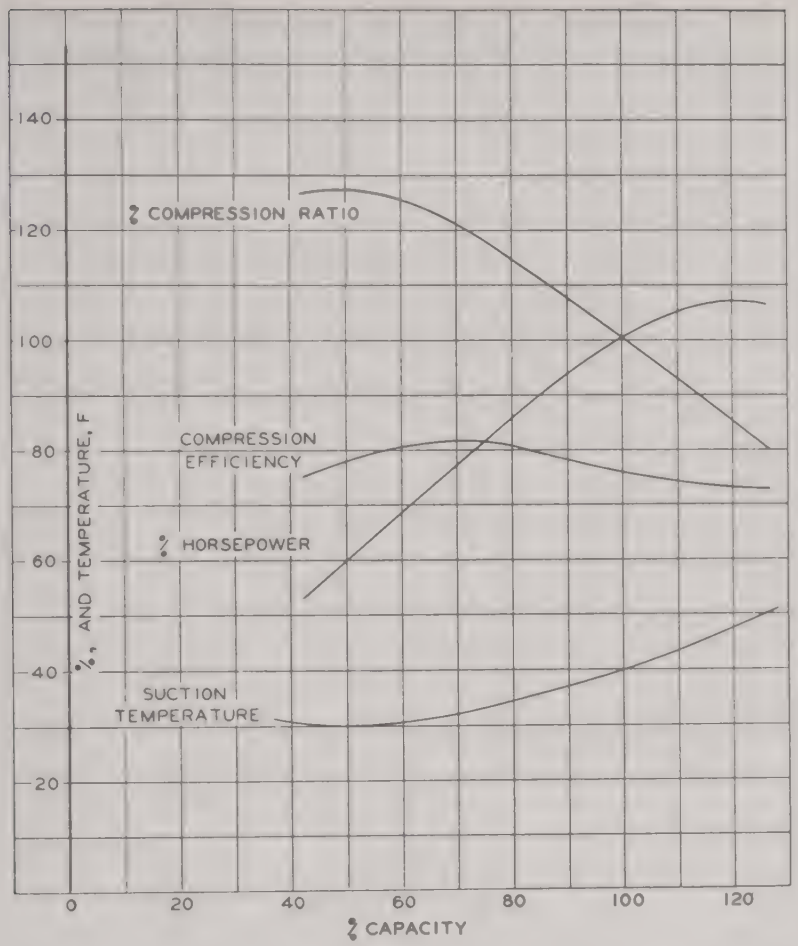

Fig. 10. Performance of Centrifugal Compressor at Constant Speed and Fixed Condensing Temperature

Fig. 10 shows typical performance curves for a centrifugal compressor operating at constant speed and condensing temperature.

Figs. 11 and **12** show performance characteristics of centrifugal machines with different refrigerants and are plotted against varying condensing temperatures. It will be noted that there is an appreciable drop in efficiency and capacity as the condensing temperature increases. This effect can be utilized to govern a machine for reducing capacity, as it is only necessary to raise the condenser pressure; this can easily be done by decreasing the quantity of water flow through the condenser.

Table 7 indicates typical capacities and sizes of the machines of one manufacturer. The capacity in this table is expressed in tons in the three left-hand columns; it will be noted how much the capacity increases in terms of higher suction temperatures.

5. Control. Ordinary safety devices consisting of a **safety relief valve** or valves on the condenser or evaporator are required, along with an **oil-pressure failure switch** for lubrication failure, **high-pressure cutout**, low-temperature water or **brine cutout thermostat,** and sometimes low-liquor **refrigerant temperature** thermostat **cutout.** The operation of the refrigerating compressor to balance with the refrigeration load is not difficult nor critical, and can be accomplished by means of manually varying the quantity of gas entering the impeller of the compressor by suction-damper control or by variable inlet guide vanes. With either motor drive using a wound-rotor motor or with a turbine, variable speed can be accomplished. For the electric

Table 5. Typical Performance of Turbine with a Centrifugal Compressor

Type turbine	Inlet pressure, psig	Superheat, deg F	Exhaust pressure	Steam consumption in lb/bhp-hr
Condensing (high-pressure)	600	300	4 in. Hg abs	8–10
	400	150	4 in. Hg abs	10–12
	250	100	4 in. Hg abs	11–13
Condensing (low-pressure)	1	0	4 in. Hg abs	28–32
Non-condensing	600	300	25 psig	20–22
	400	150	25 psig	27–29
	250	100	25 psig	34x36

drive a sufficient number of steps of resistance to reduce the speed approximately 25%, from full-load to no-load rating, is used. With turbine drive an infinite number of speed ranges can be obtained automatically or manually as desired, by regulating flow of steam by the turbine throttle valve. The centrifugal compressor, like all centrifugal pumping apparatus, can be throttled in capacity simply by increasing the head, which in the refrigeration cycle is accomplished by regulation of condenser water quantity. The regulation of condenser water to increase the head may be accomplished by means of a thermostat located in the brine or in the liquid refrigerant; this was one of the earliest types of capacity control for the centrifugal machine. Although suitable capacity regulation can be obtained by this means, it is more economical from a power and operation cost standpoint to employ speed regu-

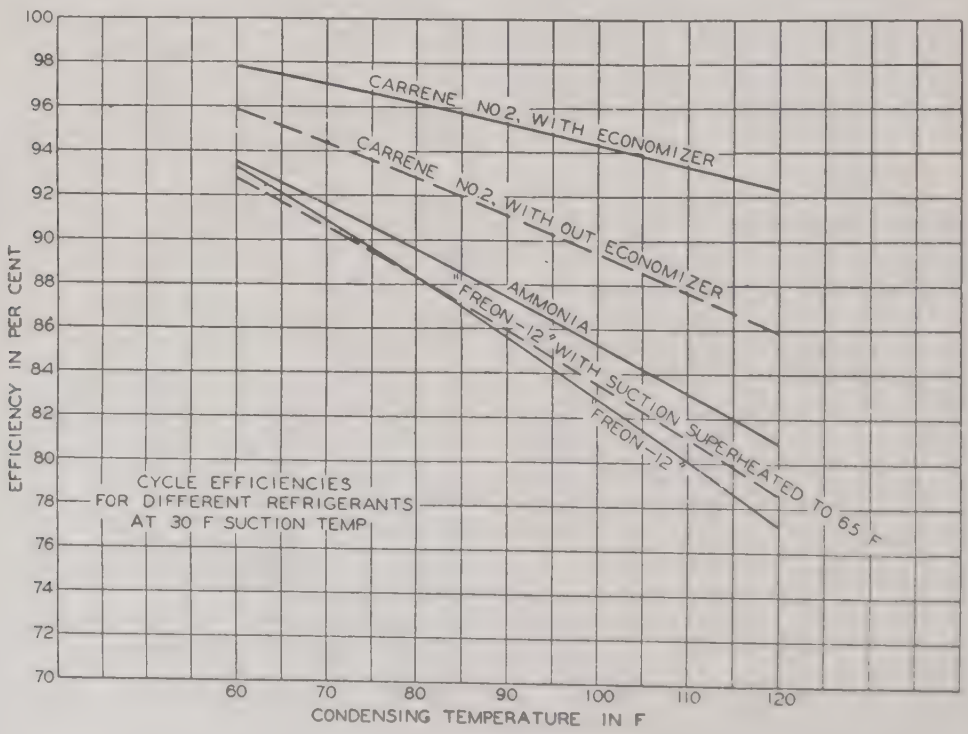

Fig. 11. Cycle Efficiencies for Different Refrigerants under Varying Condensing Temperatures

Table 6. Characteristics of Common Refrigerants

At 5 F sat evaporation and 86 F sat condensing	Gage pressure corresponding to sat evap	Gage pressure corresponding to sat cond	Specific volume of vapor at sat evap	Latent heat of evaporation	Liquid circulation, lb/min ton	Theoretical displacement, cu ft/min ton	Refrigerating effect, Btu/lb	Compression ratio Cond pres psia / Suc pres psia	Theoretical hp/ton	Coefficient of performance 200 / Theoretical hp ×42.4	Efficiency % Carnot cycle
Ammonia	19.6	154.5	8.15	565.0	0.42	3.44	474.4	4.93	.989	4.77	83.2
Freon-12	11.8	93.2	1.49	69.47	3.92	5.82	51.1	4.07	1.007	4.69	81.6
Methyl chloride	6.46	80.00	4.47	180.70	1.33	5.95	150.25	4.48	.963	4.90	85.4
Freon-114	16.14	21.99	4.221	61.98	4.64	19.60	43.10	5.42	1.233	3.83	66.6
Carbon dioxide	317.5	1028.3	0.266	117.5	3.53	0.94	56.7	3.14	1.84	2.56	44.6
Carrene No. 1 (Methylene chloride)	27.55" Vac	8.35" Vac	49.90	162.10	1.49	74.5	134.1	8.59	.965	4.89	85.2
Carrene No. 2 (Freon-11)	23.95" Vac	3.58	12.27	84.00	2.96	36.4	67.5	6.24	.936	5.04	87.8
Propane without economizer	26.98	141.5	2.52	171.35	1.625	4.1	123.	3.75	1.03	4.59	80.0
Butane without economizer	—	26.9	9.98	169.5	1.622	16.2	123.5	5.09	.982	4.81	84.0
Carnot cycle	—	—	—	—	—	—	—	—	.822	5.74	100.
At 40 F sat evaporation and 100 F sat condensing											
Ammonia	58.6	197.2	3.97	536.2	0.43	1.70	467.8	2.89	.649	7.28	87.3
Freon-12	36.98	116.9	0.792	65.7	3.88	3.70	51.6	2.54	.667	7.07	84.9
Freon 114	0.52	31.69	1.982	58.86	4.47	8.86	44.67	3.04	.878	5.04	64.7
Carrene No. 2 (Freon-11)	15.61" Vac	8.90	5.45	81.2	2.90	15.8	68.8	3.36	.625	7.55	90.9
Water	29.75" Vac	28.07" Vac	2444.0	1068.8	0.198	482.0	1008.9	7.78	.665	7.08	85.0
*Carbon dioxide	553.1	1054.7	0.1444	95.0	5.04	0.727	39.7	1.885	.867	5.45	52.0
Propane without economizer	63.10	174.0	1.33	159.6	1.63	2.17	122.4	2.405	.678	6.98	83.8
Butane without economizer	3.0	37.5	4.88	163.5	1.563	7.64	128.0	2.95	.687	6.89	82.8
Carnot cycle	—	—	—	—	—	—	—	—	.566	8.34	100.0

* For sat evaporation at 40 F and sat condensing at critical point = 87.8 F.

Table 7. Typical Capacity, in Tons, of Centrifugal Machines at Varying Suction Temperatures and Representative Sizes

(Conditions 80 F condensing water, 8 fps water velocity in cooler and condenser)

Tons at 35 F	Tons at 40 F	Tons at 45 F	Compressor wheels	Length	Width	Height	Compressor head room	Approximate shipping weight in pounds without drive
130	145	153	2	14′6″	7′3″	6′4″	6′2″	17,580
145	163	171	2	14′6″	7′3″	6′4″	6′2″	17,900
160	178	188	2	14′6″	7′6″	6′4″	6′2″	18,950
220	245	260	2	14′6″	8′8″	6′9″	7′0″	23,600
246	274	290	2	14′8″	8′8″	6′11″	7′0″	24,200
272	300	320	2	14′9	9′2	7′3″	7′0″	25,600
290	324	340	2	14′9″	9′2″	7′8″	7′0″	25,600
376	420	440	2	14′10″	10′9″	8′1″	8′3″	34,400
420	465	495	2	14′10″	10′9″	8′5″	8′3″	35,800
460	515	545	2	15′0″	11′0″	8′9″	8′7″	38,400
500	555	590	2	15′0″	11′0″	8′9″	8′7″	38,400
640	710	750	2	15′8″	12′8″	9′8″	10′6″	55,900
710	790	840	2	15′8″	13′0″	10′0″	10′9″	57,400
780	870	925	2	15′9″	14′0″	10′1″	10′9″	67,000
925	1000	1120	2	15′9″	14′0″	10′1″	10′9″	67,000

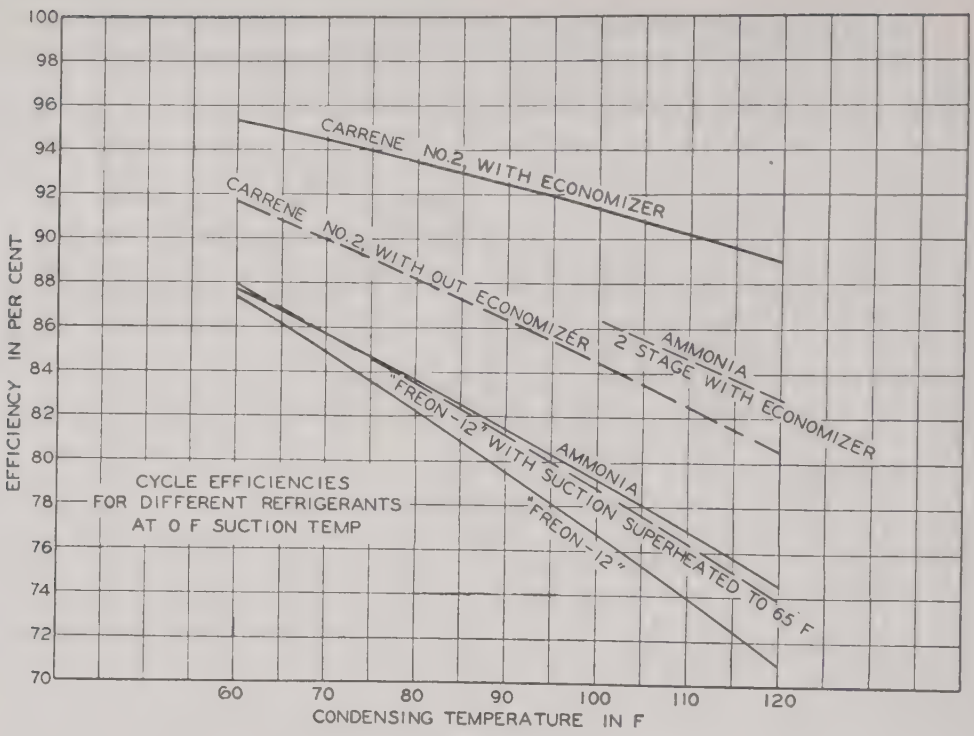

Fig. 12. Cycle Efficiencies for Different Refrigerants Under Varying Condensing Temperatures

lation or regulation of suction gas by means of suction damper control or inlet guide vanes or a combination of both.

Basically a centrifugal machine properly designed does not require extensive control to be capable of **varying with the refrigerating load.** This is accomplished in the design of the impeller and results in a flat characteristic curve for a wide variation in capacity. With this flat characteristic a small change in speed over the entire range of capacity, say from 25% to more than 100%, will have little effect in changing the ratio of compression, and therefore only minor modifications in speed or by suction damper control are necessary for most economical and stable operation.

As the capacity loading of a centrifugal compressor is reduced, a point is finally reached, usually somewhere below 50% of basic selection, where surging occurs. This is a natural characteristic of all centrifugal compressors whether compressing refrigerant vapors, gases or air. The surge point, which may be instantly noted by a change in noise level or hum of the machine, is caused by the compressor removing refrigerant vapor from the evaporator at a rate in excess of its formation in the evaporator. The flow of gas momentarily stops and reverses as the compressor reaches the limit point, and this results in a change of noise level. No appreciable loss of economy and no untoward effects on the mechanical function of the machine are caused by this surging. The point of surging can be lowered by suction damper control, and in severe cases of a low-load condition, usually less than 15%, a bypass can be installed to eliminate surging, but only with some loss in efficiency, and with somewhat higher horsepower per ton required at the reduced load.

6. Ammonia condensing cycle. The complete centrifugal refrigerating unit, consisting of compressor, condenser, evaporator and drive, has been successfully applied to industrial refrigeration not only for cooling of standard brines, such as calcium chloride and sodium chloride, but for all refrigerated solutions. Within the last 15 years, the application of the centrifugal

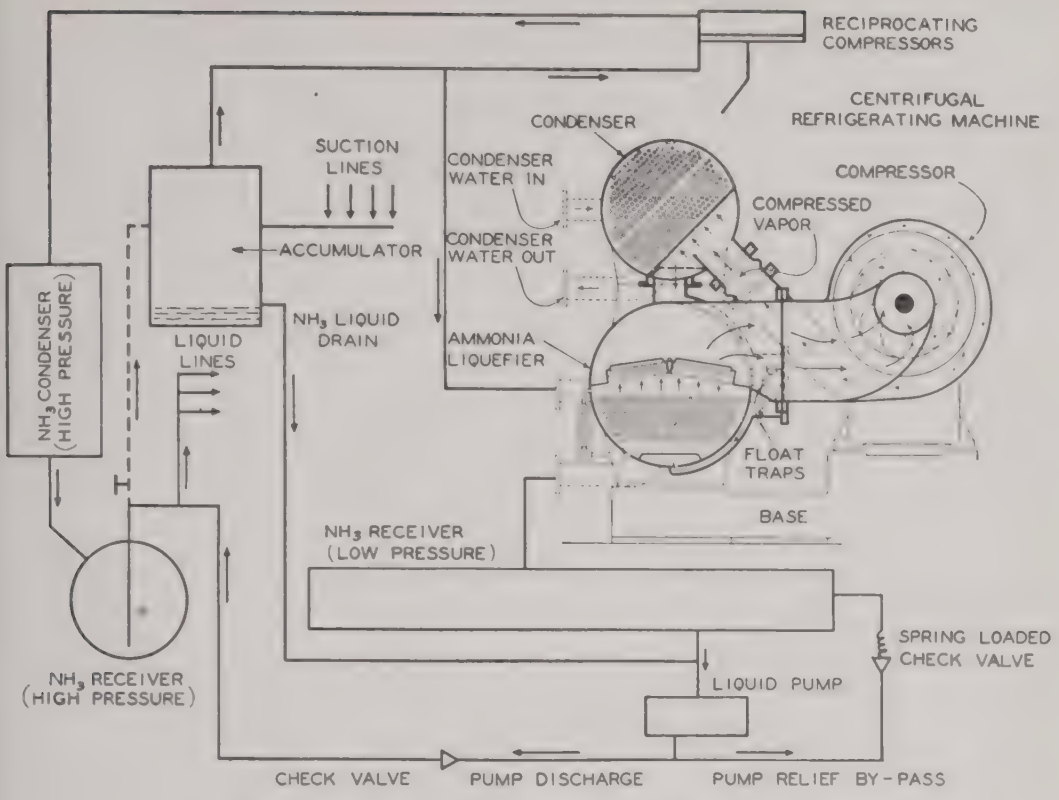

Fig. 13. Ammonia Condensing Cycle Using Centrifugal and Reciprocating Compressors

unit has been broadened to include the condensing cycle for the condensing of various vapors such as chlorine, hydrocarbon gases, but more especially ammonia (see **Fig. 13**).

In 1934 the first application of the centrifugal unit for ammonia condensing was made with success, a design which allowed the cross connecting of a centrifugal refrigerating machine with standard ammonia compression cycles. The merits of the centrifugal machine were thus realized in those plants using ammonia as the refrigerant and having extensive ammonia piping systems.

The ammonia suction line is connected to the evaporator or cooler of the centrifugal unit in a manner identical to that used in connecting it to a reciprocating machine. The ammonia vapor passes through the evaporator tubes, and condenses to liquid by means of heat transfer rather than by compression and heat transfer, as is the case in the standard ammonia condenser. The liquid is taken off at the several passes in the cooler and drained into a conventional receiver, the only difference being that the ammonia is at low pressure and temperature as established by the refrigerating plant requirements. It is obvious that some differential in temperature between the refrigerant evaporating in the cooler and in the liquid ammonia must exist, and coolers are designed for approximately 7.5° to 10° differential for this purpose.

The advantage of this type of ammonia refrigerating system is the elimination of flash gas in the ammonia cycle, with subsequent increase of efficiency of ammonia pipe coils when compared to dry expansion or unit coolers or other evaporators in the plant itself. The lubricating oil ordinarily circulated through the evaporators in a reciprocating cycle is completely eliminated in this type of system, and not of least

importance is the safety feature of having the ammonia at low pressure and circulated through the system by means of a conventional centrifugal pump. Ammonia circulating pumps require a 5 to 15-hp motor for supplying approximately 200 to 500 tons refrigerating capacity of ammonia liquid.

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24. CONDENSERS

THE condenser in a refrigerating system converts to liquid form the gas delivered by the compressor, thereby removing the heat of compression and the latent heat of vaporization to provide for a continuous supply of liquid refrigerant for expansion in the evaporator.

The **common forms** of **condensers** used at the present time may be classified as follows: shell-and-tube, double-pipe, atmospheric and the evaporative type. Principal attention in this chapter is given to condensers as applied in industrial systems. Domestic and commercial-type condensers are also mentioned briefly. In larger installations the shell-and-tube design, built in either vertical open type or horizontal closed type, is more prominent. The horizontal type is exclusively multipass with closed water heads. In the vertical type water flow is by gravity with individual distributors for each vertical tube, thus assuring proper water distribution to each tube.

The type of condenser varies with the size of installation and operating conditions. Shell-and-tube condensers have been replacing the double-pipe or atmospheric type for industrial applications. Evaporative condensers have a wide range of application, particularly where water is scarce or expensive. For commercial refrigeration applications, the shell-and-tube, shell-and-coil and double-pipe types are generally used. Domestic and small commercial refrigerating machines use air-cooled condensers which are generally limited to 3 hp condensing unit sizes.

Heat Removed

1. The heat in Btu per min removed in the condenser for each ton of refrigeration (200 Btu per min) produced in the evaporator, may be estimated for various refrigerants from **Fig. 1** for ammonia, **Fig. 2** for Freon-12 and **Fig. 2A** for Freon-22. The values shown are based on saturated vapor at the compressor inlet and adiabatic compression.

The **amount of condensing water** required in pounds per minute may be calculated by dividing the total heat removed per minute in the condenser by the product of the specific heat of water and the temperature rise. To convert pounds of water to gallons, divide by 8.33.

Example: Ammonia condensing pressure = 180 psig; suction pressure = 20 psig; initial temperature of condensing water, $t_1 = 85$ F; and final temperature, $t_2 = 90$ F. The quantity of condensing water required, gpm per ton of refrigerating capacity, is

$$\frac{H \times 1 \text{ ton}}{8.33(t_2 - t_1)}$$

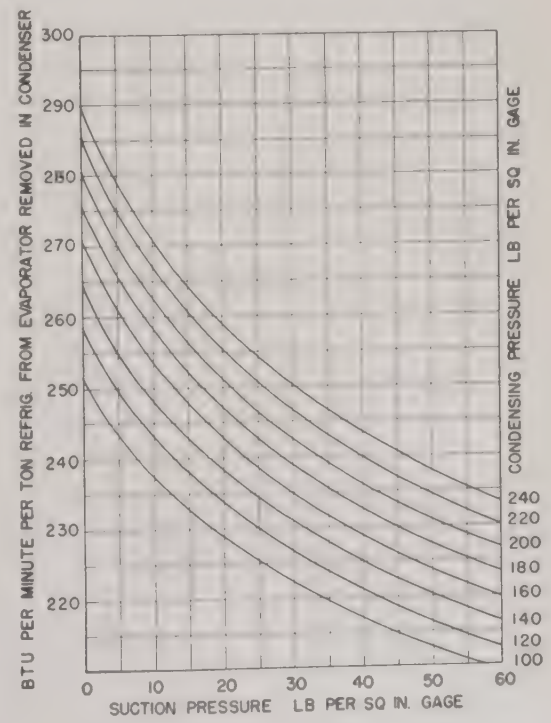

Fig. 1. Btu per Minute Removed in Ammonia Condenser Per Ton of Refrigerating Effect Occurring in Evaporator

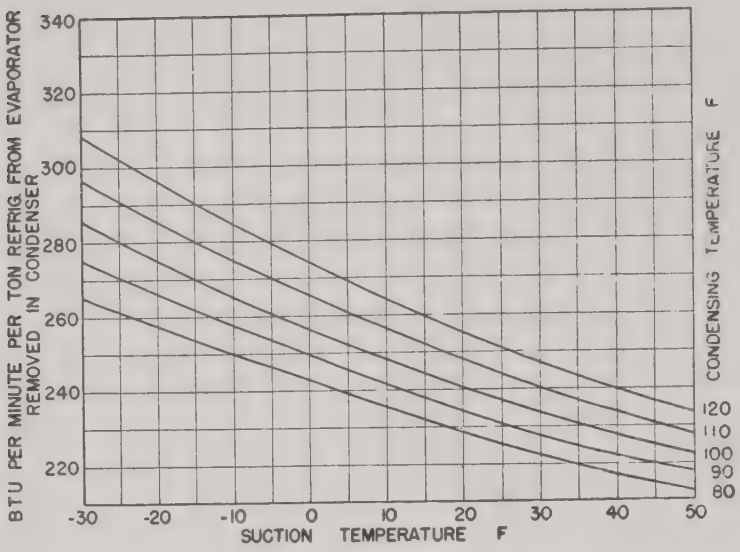

Fig. 2. Btu per Minute Removed in Freon-12 Condenser per Ton of Refrigerating Effect Occurring in Evaporator

where $H = 247$ Btu per min taken from curve, Fig. 1; hence

$$\frac{247 \times 1 \text{ ton}}{8.33(90-85)} = 5.9 \text{ gpm per ton.}$$

Vertical Open Shell-and-Tube Ammonia Condenser

2. The characteristics of this type of condenser are: (a) large condenser capacity may be installed in small floor space; (b) complete installation cost is low; (c) water distribution is simplified; (d) gas binding difficulties are eliminated; (e) large space is available in the condenser for storing liquid ammonia; (f) it is adaptable to simple purging connections; (g) tubes may be easily cleaned without shutting down; (h) condensed ammonia flows down the tubes and is quickly carried away from the heat-transmitting surfaces while still keeping them wet; (i) condenser can be made to carry large overloads by increasing the quantity of water circulated through the tubes, at the same time avoiding a heavy increase in friction head; (j) condenser is adaptable to any location, inside or outside of buildings, but inside installations have a tendency to "steam" in winter; and (k) it can be used effectively with condensing water from a cooling tower or spray pond.

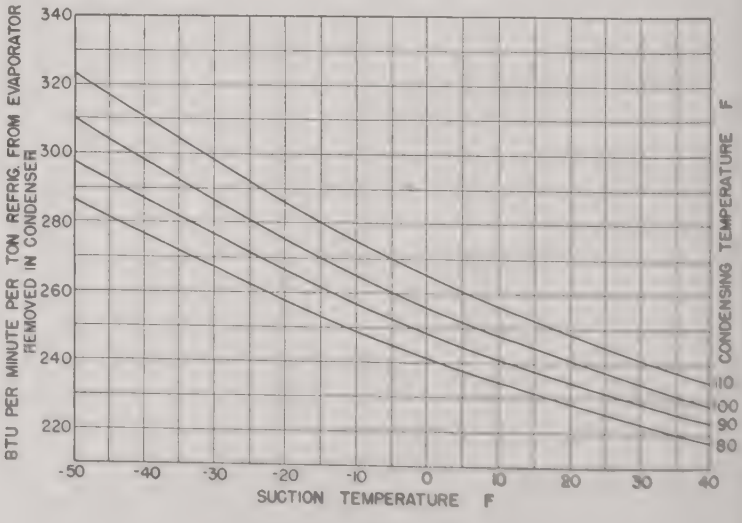

Fig. 2A. Btu per Minute Removed in Freon-22 Condenser per Ton of Refrigerating Effect Occurring in Evaporator

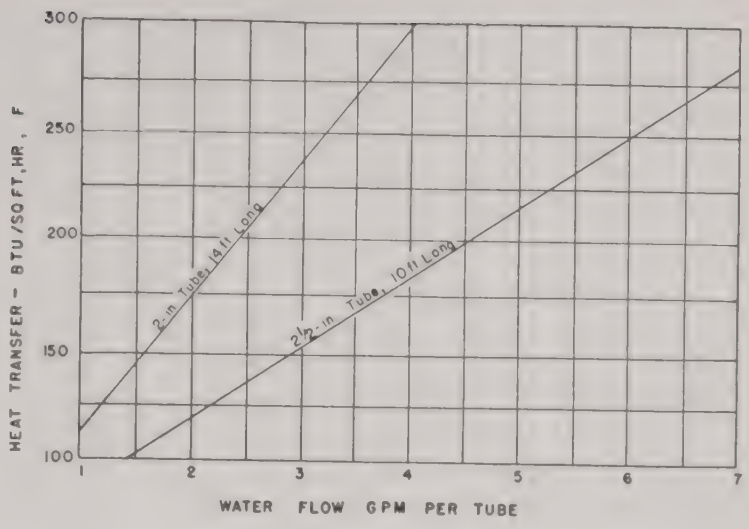

Fig. 3. Heat Transfer Coefficient for Vertical Shell-and-Tube Ammonia Condensers

The laboratory data available on shell-and-tube condensers are by Kratz, Macintire and Gould[2,3] on a 20-in. condenser with 30 2-in. tubes of varied length. Heat transfer coefficients ran from 150 to 350 as the water varied up to 2.5 gpm per tube. Raising the level of the liquid, and thus making the area smaller, increased the unit transfer of heat through the condenser surface.

In computing **transmittance,** values may be based upon logarithmic or arithmetic temperature differences and upon internal or external area. The exact basis used to arrived at a *U* value is stated here if mentioned in the original source.

In **Fig. 3,** the 2-in. tube curve is presented from the data of Zumbro[4] obtained in tests of a vertical shell-and-tube condenser. Ophuls and Horne[5] gave the results of a certain test made on a vertical shell-and-tube condenser, 10 ft long, having $2\frac{1}{2}$-in. tubes, also in Fig. 3.

Kratz et al call attention to the fact that in their tests, the coefficient of heat transfer at constant water rates was found to decrease with increasing values of the mean temperature difference between the condensing ammonia and the water in the tubes of the shell-and-tube condenser.

The data given in curves and tables by Sloan and Panlener in **Fig. 4** estimate the **performance** of vertical ammonia shell-and-tube condensers. In actual practice they give very close agreement to the data obtained in the field after the condensers have been in service for some length of time, and thus lend themselves to convenient practical use.

The construction of a typical vertical ammonia shell-and-tube condenser is shown in **Fig. 5.** It is built in **sizes** varying from 16 to 60 in. in diameter and in **heights** ranging from 10 to 16 ft, with 14 ft a recommended normal height. As a rule the seamless steel tubes are 2-in. O.D. No. 11 B.W.G. average wall thickness. The top of each tube is fitted with a water distributor made of cast iron, brass, plastic, glass or porcelain to give the water a swirling motion as it enters the tube.

Example: Required, the surface in a 16-ft vertical ammonia condenser having 139 tons capacity at 10 psi suction and 175 psi condensing pressure, when using 3 gpm per tube at 75 F water temperature. Take 139 tons $\times$ 7.26 sq ft per ton $\times$ 1.05 $\times$ .863 = 916 sq ft of surface. This requires a shell having 109 2-in. O.D. tubes, 16 ft long, using 327 gpm of condensing water. Tables under Fig. 4 permit correction due to varying tube lengths, gallons per minute water per tube and suction pressures.

Table 1 gives approximately the number of 2-in. tubes which might be placed in various shell diameters.

Numerous **variations** of the **vertical tube** principle have been worked out in ammonia condenser practice. The condenser shell may be made of standard pipe in the smaller sizes and fusion welded shells in the larger. The tube sheets, usually 1 to $1\frac{1}{4}$-in. thick, are welded to the shell. One or more grooves are machined in the

Table 1. Condenser Tubes per Shell*

Shell diam, in.	No of 2-in. tubes	Shell diam, in.	No of 2-in. tubes
16	21	38	144
20	33	42	191
24	51	45	220
28	76	48	244
30	85	50	291
34	115	54	330
36	132	60	395

* The number of tubes in each shell varies with individual designs and may vary up to 20% above those shown here.

drilled and reamed tube hole in the tube sheet. Tubes with ground or polished ends are then inserted through their respective tube sheet holes and rolled into the grooves to form a gas-tight joint. This same procedure is followed on horizontal shell-and-tube ammonia condensers.

Fig. 4A shows typical values of leaving terminal difference (temperature corresponding to ammonia pressure minus leaving water temperature) as a function of water flow in gpm per tube for loadings from 5 to 18 sq ft of external tube surface per ton and for condensing water quantities from 1 to 5 gpm per ton. The values are for commercially clean 2-in. OD steel tubes with a nominal length of 14 ft, and are based on a condenser free of non-condensible gases. The rise in the condensing water temperature is considered 28 gal deg per min per ton and the removal of heat from the ammonia vapor as 250 Btu per min per ton.

Condenser selection. As an example, assume that 60 tons of refrigeration are required in the evaporator, and that there are available 240 gpm of 80 F condensing water. The water quantity is then 4 gpm per ton, which fixes the degree rise at 28 gal deg per min per ton, divided by 4 gpm per ton, or 7 F, and the leaving water temperature at 87 F. Fig. 4A shows that 4 gpm per ton at a loading of 10 sq ft per ton would give a leaving terminal difference of approximately 5 F, and that the water flow would be about 2.94 gpm per tube. As the decrease in terminal difference would probably not warrant an increase in surface, and as a decrease in surface would result in too high a water flow per tube, the condensing surface in this case would be figured at 10 sq ft per ton, or 600 sq ft.

For a commercially clean condenser, free of non-condensible gases, the condensing temperature would be 87 F (the leaving water temperature) plus 5 F (the leaving terminal difference), or 92 F. For the usual installation, it is considered good practice to allow 1.5 F for non-condensible gases, giving a condensing pressure corresponding to 93.5 F, or 176.4 psig.

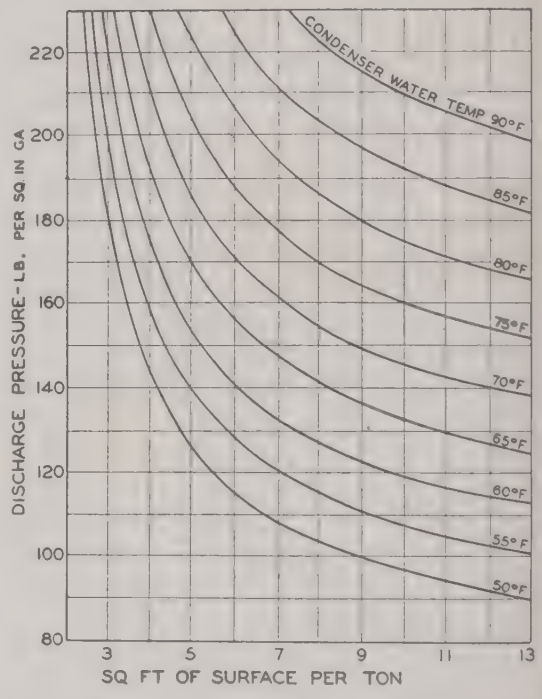

Fig. 4. Typical Rating Curves for Vertical Shell-and-Tube Ammonia Condensers

gpm per tube	Surface Correction Multiplier		
	12-ft	14-ft	16-ft
0.50	3.483	3.636	3.752
1.00	1.914	2.045	2.149
1.50	1.353	1.460	1.548
2.00	1.053	1.142	1.216
2.33	.921	1.000	1.068
2.50	.865	.941	1.005
2.67	.816	.889	.951
3.00	.738	.806	.863
3.50	.657	.720	.774
4.00	.605	.666	.719

Suction pressure	Multiplier	Suction pressure	Multiplier
0	1.12	25	0.98
5	1.08	30	0.97
10	1.05	40	0.95
20	1.00	50	0.93

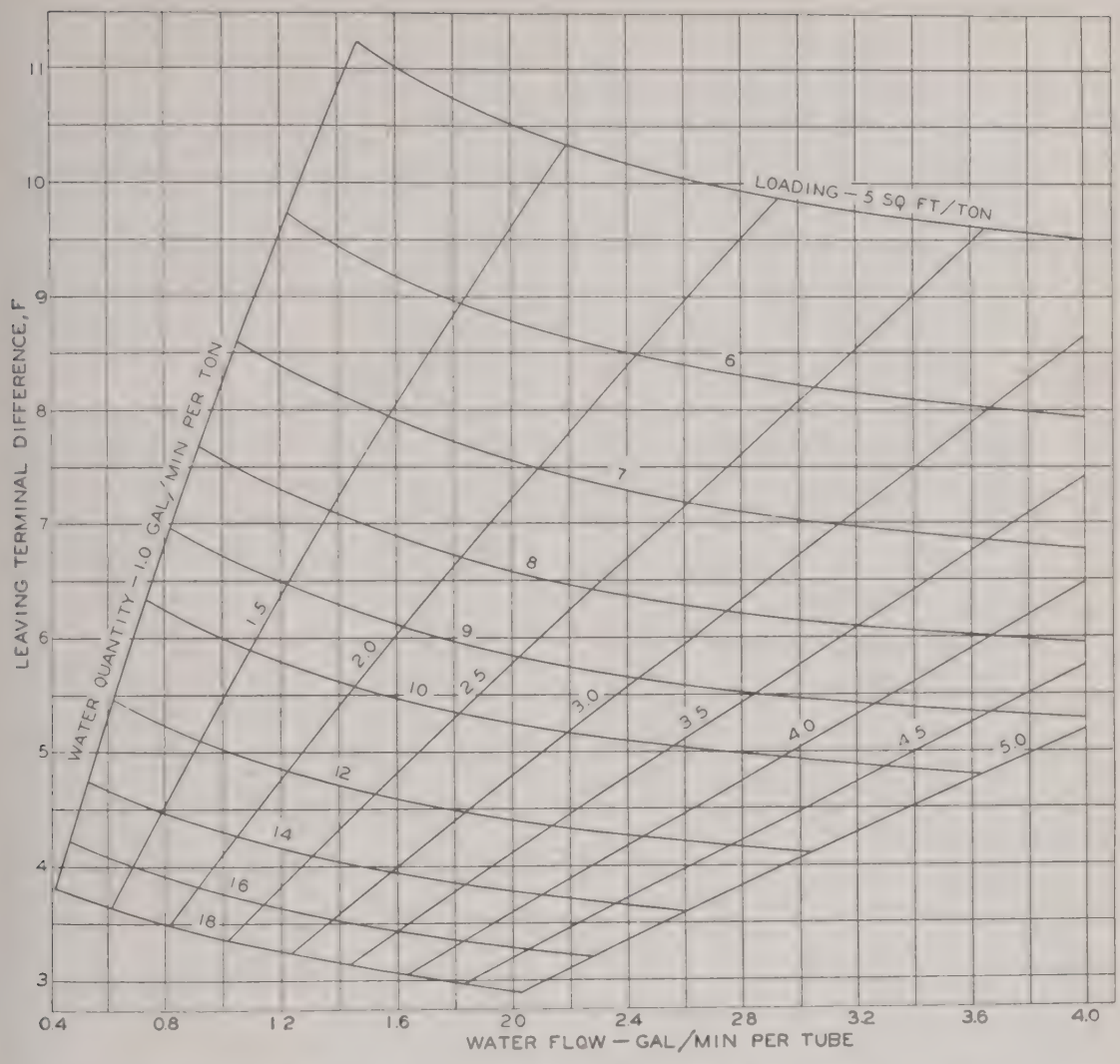

Fig. 4A. Performance of Ammonia Condensers Vertical Open Shell-and-Tube Type for Commercially Clean 2″ O.D. Steel Tubes, 14′ Long Condenser Free of Non-Condensible Gases[1]

Horizontal Closed Shell-and-Tube Ammonia Condenser

3. The horizontal shell-and-tube ammonia condenser offers the following features: (a) large condenser surface may be installed in comparatively small floor space; (b) total installed cost is low, although not so low as the vertical condenser; (c) it is adaptable to any location in the building or engine room; (d) water piping connections are relatively simple; (e) it must be shut down to remove scale; (f) it can be overloaded but only with a heavy increase in water friction; (g) it can be arranged in batteries for multiple installation, being mounted either side by side or one above the other.

This type of condenser is the standard in many fields of industrial application, especially where location of other types is difficult on account of architectural limitations.

Horizontal condensers are built in sizes varying from $8\frac{5}{8}$ in. to 60 in. in diameter and in lengths varying from 9 to 18 ft in condensers 16 in. and under, and 12 to 18 ft in those larger than 16 in.[1] Seamless steel tubes are usually $1\frac{1}{4}$-in. O.D. or $1\frac{1}{2}$-in. O.D. No. 13 B.W.G. average wall thickness, or 2-in. O.D. No. 11 B.W.G. average wall thickness.

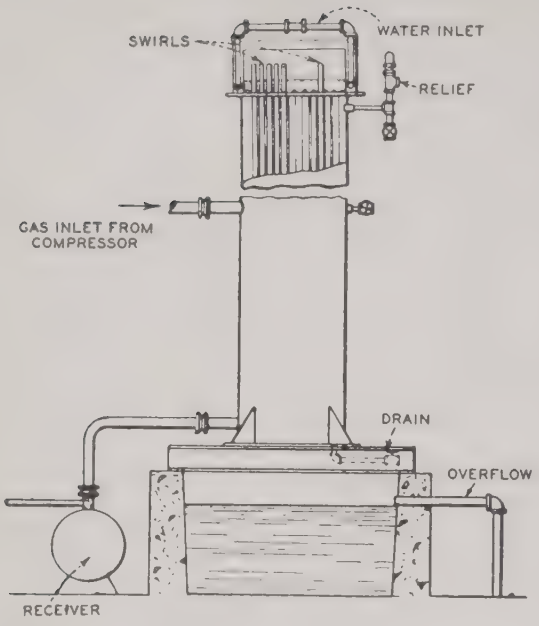

Fig. 5. Construction of Typical Shell-and-Tube Condenser

Figs. 8A and **8B** illustrate a method of rating horizontal ammonia condensers with $1\frac{1}{4}$-in. and 2-in. O.D. tubes. The lower two quadrants of Fig. 8A for $1\frac{1}{4}$-in. O.D. steel tubes and Fig. 8B for 2-in. O.D. steel tubes show typical values of leaving terminal difference (temperature corresponding to ammonia pressure minus leaving water temperature) as a function of length of travel (tube length times number of passes), water quantity, and loading (square feet of external tube surface per ton). The curves of the lower left-hand quadrant form a complete picture of heat transfer at a loading of 8 sq ft per ton, the curves of the lower right-hand quadrant offering a correction for other loadings. The values are for commercially-clean tubes in a condenser free of non-condensible gases and are based on a temperature rise in the condensing water of 30 gal deg per min per ton and on a removal of heat from the ammonia vapor of 250 Btu per min per ton.[1]

The condensing temperature is the sum of the entering water temperature, the degree rise, and the leaving terminal difference. It is, therefore, determined by the selection of a water quantity, length of travel, and loading that will give a relatively low condensing temperature at a reasonable pressure drop. As the terminal differences in the accompanying curves are based on a condenser free of non-condensible gases, it is recommended that, for the usual installation, an allowance of 1.5 F be added to the leaving terminal difference for non-condensibles. In refrigerating systems operating below atmospheric pressure, or where unusually high percentages of non-condensible gases are present, this allowance should be increased accordingly.

The upper two quadrants show values of pressure drop as a function of length of travel, water quantity, and loading. The right-hand quadrant need be used only for values of loading other than 8 sq ft per ton. Values in Fig. 8A are based on $1\frac{1}{4}$-in. O.D. steel tubes with an average wall thickness of 13 BWG and a nominal tube length of 14 ft; and Fig. 8B on 2-in. O.D. steel tubes with an average wall thickness of 11 BWG and a nominal tube length of 16 ft. Pressure loss in the water heads is based on Fig. C-5 of the 1939 edition of the Condenser Section, Standards of Heat Exchange Institute, and can be adjusted for other tube lengths if desired.

As an example, assume that 75 tons of refrigeration are required in the evaporator, that the entering water temperature is 80 F, and that the water costs are quite reasonable and the power costs average. For 2-in. tubes (see Fig. 8B), with 5 gpm per ton, the leaving terminal difference (lower left-hand quadrant) is reasonable at a length of travel of 128 ft. For a 128-ft length of travel, the terminal difference would be 5.3 F at 8 sq ft per ton and 4.5 F

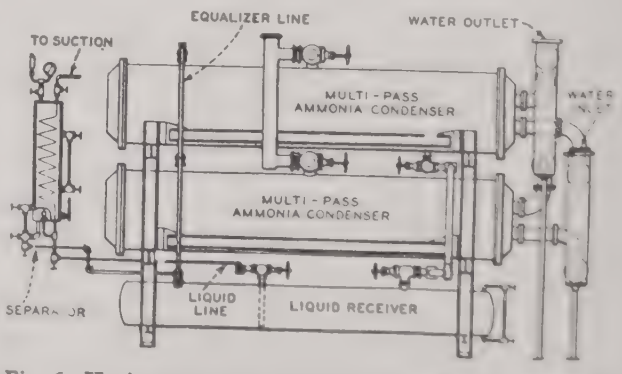

Fig. 6. Horizontal Shell-and-Tube Ammonia Condenser and Receiver

(lower right-hand quadrant) for 10 sq ft per ton. The resulting pressure drop for a 128-ft length and 5 gpm per ton would be 10.9 psi at 8 sq ft per ton (upper left-hand quadrant) and 7.5 psi at 10 sq ft per ton (upper right-hand quadrant) for a 16-ft condenser. Selecting the lower pressure drop, the condenser would require about 750 sq ft of external tube surface and would have a leaving terminal difference of approximately 4.5 F plus 1.5 F for noncondensible gases, or 6.0 F. The condensing temperature would equal the sum of the entering water temperature, the degree rise (30 gal deg per min per ton divided by gpm per ton), and the leaving terminal difference, or 80 F plus 6 F plus 6 F, or 92 F. This corresponds to a condensing pressure of 172 psig.

Although the optimum selection depends upon the conditions of each installation, the prevailing power and water rates, and, frequently, the quantity of water available, Fig. 8B shows that the above selection would be a reasonable one for average water and operating conditions. For cases of heavily-incrusted water, the condenser should be selected for low values of terminal difference and on the flatter portion of the curves, where a change in heat transfer will not produce as large a change in condensing pressure.

Figs. 8A and 8B show that at low water quantities, low lengths of travel, and low values of square feet per ton, the leaving terminal difference becomes excessive. At such conditions, a small change in heat transfer would mean a large change in condensing pressure. At the same time, the curves show that, at high water quantities for long lengths of travel and relatively small values of square feet per ton, the pressure drop becomes excessive. The curves therefore afford a direct comparison and a balance of conditions to give a practical and complete analysis of condenser performance.[1]

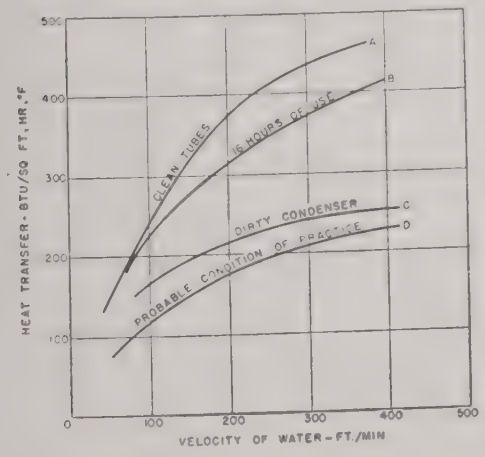

Fig. 7. Heat Transfer Coefficients in Horizontal Shell-and-Tube Ammonia Condensers

The most comprehensive data available relating to **heat transfer** of the horizontal multipass shell-and-tube ammonia condenser are given by Kratz, Macintire and Gould.[3] Tests were performed on a bank of two shells, each containing seven 2-in. No. 13 BWG tubes, each 12.82 ft long. The superheat or heat of compression in the gas as it came from the compressor was removed in a gas precooler.

In **Fig. 7** the results have been condensed to give the heat transfer as a function of the water velocity in curve A. This curve refers to the tests made with water circulating parallel through the two shells when the tubes were clean. Curve B refers to data taken in the same manner as those of curve A with tubes which had been in service 16 hr. Curve C refers to dirty tubes; the length of time the condenser had been in service was not stated.

The variation in heat transfer as a function of the length of time the condenser has been in service is clearly indicated by the progressively lower values as indicated by curves A, B and C. Ferrous tubes seem to have an increasing scale resistance because of the continuous oxidation of inside tube surfaces, even with normal waters. High rates of hear transfer, often given, are frequently traceable to the fact that such measurements are made on condensers having clean tubes.

Curve D represents the heat transfer that may be expected in practice after the condenser has been in service a reasonable length of time. These curves are based on external tube surface and real temperature differences measured at various points and averaged.[3] The logarithmic mean temperature difference may be used with curve D with sufficient accuracy. This checks with the data of Pownall.[11]

The **water friction** pressure loss, important on horizontal condensers, is composed

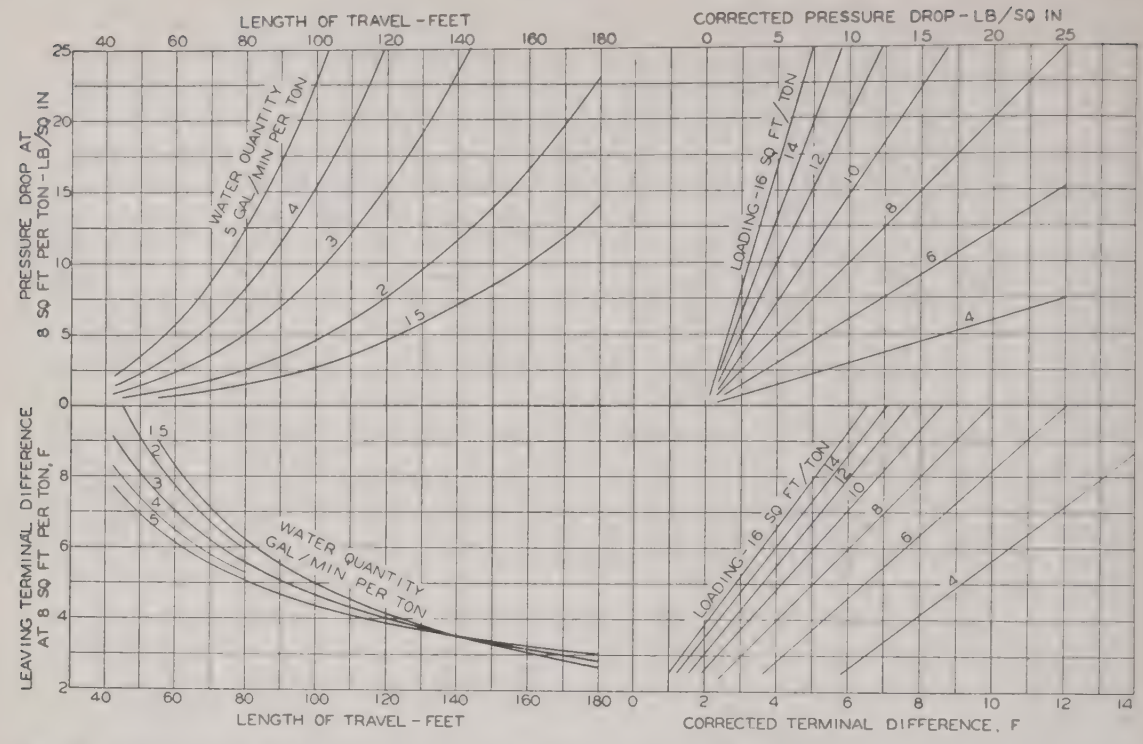

Fig. 8A. Performance of Ammonia Condensers Horizontal Closed Shell-and-Tube Type for Commercially Clean $1\frac{1}{4}$" O.D. Steel Tubes Condenser Free of Non-Condensible Cases[1,11]

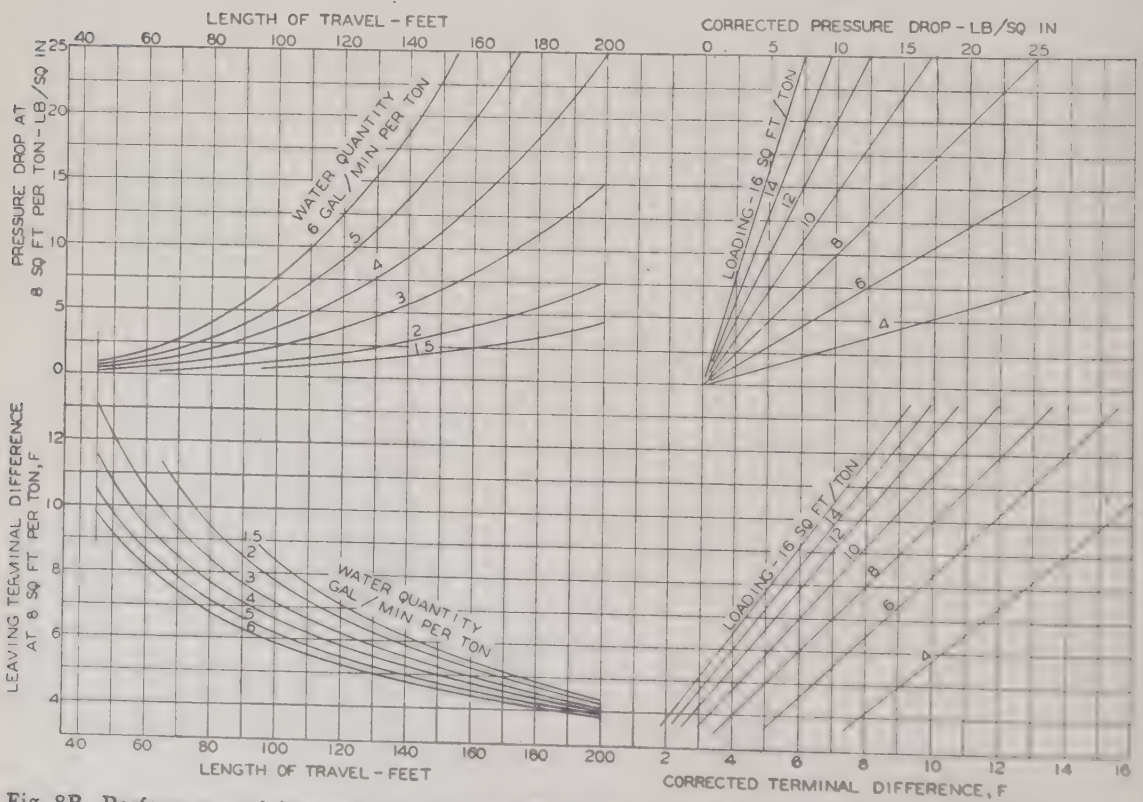

Fig. 8B. Performance of Ammonia Condensers Horizontal Closed Shell-and-Tube Type for Commercially Clean 2" O.D. Steel Tubes Condenser Free of Non-Condensible Gases[1]

of the loss through the tubes plus the loss resulting from entry conditions, return bends, and changes in cross sectional area. This is clearly illustrated by **Fig. 9,** taken from the tests of Kratz et al.[3] Note that the loss through the return bends forms a large proportion of the total friction loss, being more than half in this cited case. In order to evaluate this loss, it is necessary to have the return bends resolved into equivalent length of tubing; this value varies principally with design. The values may be estimated in seamless steel tube condensers by using 16 ft of equivalent length of tubing for each return bend, the Williams and Hazen friction factor with an accompanying multiplier of .55 for smooth, clean tubing.

The method of fastening tubes to the tube sheet is the same as on vertical shell-and-tube ammonia condensers.

4. The **double-pipe ammonia condenser** is usually built of $1\frac{1}{4}$ and 2-in. pipe from 4 to 12 pipes high in lengths 10 to 20 ft. Gas comes in at the top, water at the bottom, the action thus being counter-flow. The gas passes between the pipes, the water through the inner pipe. The condenser may be built up with fittings or may be of welded construction.

Numerous references to the heat transfer obtained in double-pipe condensers are available. **Fig. 10** shows data taken from tests by Kratz et al[2] on a standard $1\frac{1}{4}$-in. by 2-in. pipe, 12 pipes high, 18 ft long condenser. The condenser was new, and hence probably not corroded or scaled. The transfer recorded referred to that part of the condenser surface effective in liquefying the dry saturated ammonia and surfaces ranged from 18.4 to 4.6 sq ft per ton capacity. The results show that the heat transfer varies not only with velocity of the water within the pipes, but with the mean temperature difference between the water and the condensing ammonia. (Note that the water is measured in lb per min.)

Stewart and Holland[6] tested a double-pipe condenser, consisting of $1\frac{1}{4}$-in. and 2-in. pipes, 8 pipes high, 10 ft long. The heat transfer recorded by these experimenters refers to the overall heat transfer including the removal of superheat. **Table 2** shows the heat transfer through clean surface with a 10 F mean temperature difference for varying water velocities. Actual values should be reduced in accordance with the water fouling effect.

Table 2. Heat Transfer Coefficients in Double-Pipe Condensers

(10 F mean temperature difference)

Water velocity, fpm	Heat transfer, Btu/sq ft hr F
50	185
100	240
150	285
200	330

In these tests the mean temperature difference was obtained by graphical solution of the difference in temperatures between the ammonia and water at various points in the condenser. Both of the foregoing investigators conclude from their tests that in double-pipe condensers the actual heat transfer is a function of both the velocity of the water within the tubes and the mean temperature difference between the condensing ammonia and water, with the heat transfer decreasing with increasing values of the mean temperature difference. **Fig. 11** indicates this variation and may be used to predict double-pipe condenser performance in actual practice.

The **friction head** in a double-pipe condenser is shown in **Table 3.** These are val-

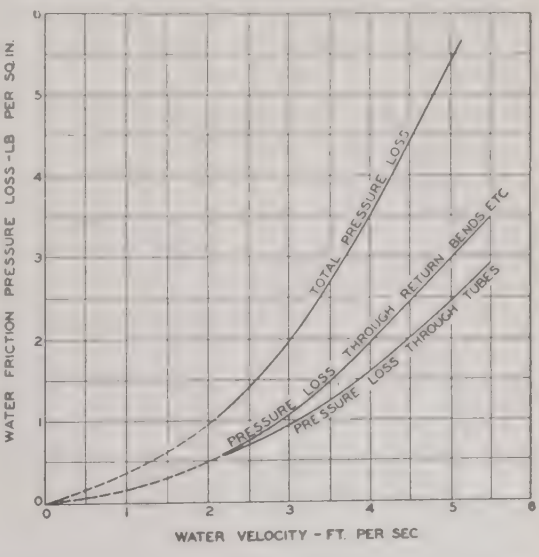

Fig. 9. Water Friction Pressure Loss in Horizontal Ammonia Shell-and-Tube Condensers

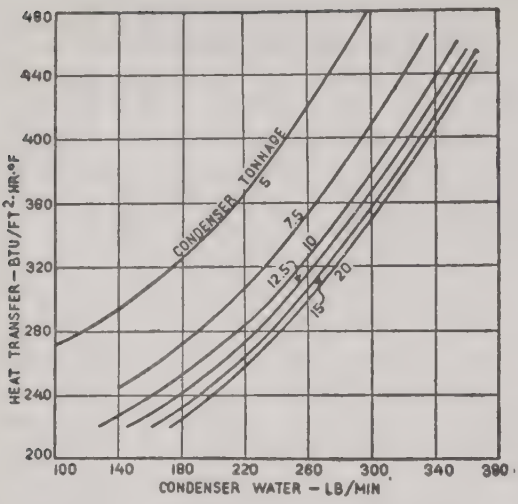

Fig. 10. Performance Curves of a Double-Pipe Ammonia Condenser

ues upon which the pumping equipment may be conservatively estimated.

The large double-pipe ammonia condenser with 2-in. and $1\frac{1}{4}$-in. pipes is rapidly becoming obsolete in the refrigerating industry; however, special designs still have their economic applications.

5. The **atmospheric** ammonia condenser was used extensively years ago in various forms, such as atmospheric drip type, the straight atmospheric type, with and without drip strips, and the Block type condenser. **Table 4** gives overall values of heat transfer including superheat and conforms to those obtained in practice. These values are taken from the data of Torrance and others.

Table 3. Water Friction Head in Double-Pipe Condensers, ft

(Pipes $1\frac{1}{4} \times 2$-in., 20 ft long)

Water, gpm	10-pipe	12-pipe
10	8	9
20	24	27
30	50	59

Table 4. Heat Transfer in Atmospheric Bleeder-Type Condenser

Water supply, gpm/ft of length	Transmittance, Btu/sq ft hr F
1	78
2	124
3	153
4	170
5	180

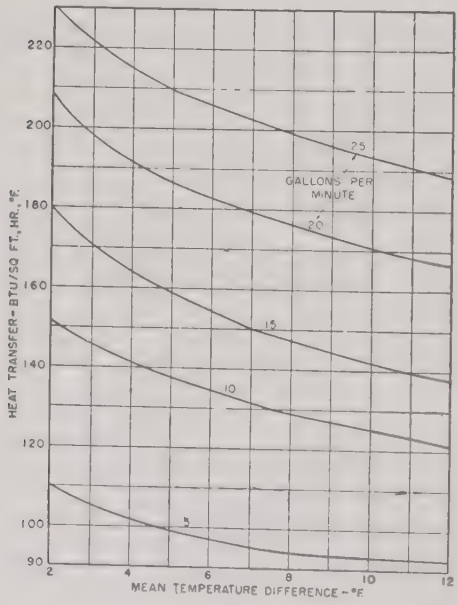

Fig. 11. Heat Transfer Values for Double-Pipe Condensers ($1\frac{1}{4} \times 2$)

Kratz et al find that in this type the overall heat transfer varies directly with the quantity of water distributed over the condenser, and that the heat transfer is a function of the velocity, independent of the mean temperature difference. The curve shown in **Fig. 12** is taken from their data as giving the percentage of splash loss at different water rates for an atmospheric bleeder condenser provided with drip strips between the pipes. The usual practice in atmospheric condensers is to circulate not more than 2 gpm of water per ft of condenser length; however, with redistributing or splash boards as much as 6 gpm per ft can be used.

For rough calculations of heat transfer, it has been customary to figure 60 to 100 Btu per sq ft hr F as the heat transfer obtained in straight atmospheric type condensers with the gas entering at the top and the liquid removed from the bottom.

and 125 for the atmospheric drip, bleeder, or flooded double-pipe type.

The atmospheric type of condenser is today practically obsolete. Several factors have contributed to this end, the foremost being the large floor space necessary for its installation and its higher first cost as compared to the shell-and-tube type condenser. It has, however, the advantage that in winter the water may often be entirely dispensed with and condensation effected by low air temperature. The present evaporative condenser is an outgrowth of this cooling tower and atmospheric condenser combination.

6. Practically all smaller ammonia units use the shell-and-tube condenser. Some of the smaller sizes previously used a shell-and-coil condenser where the water circulates through the coil, but this type of construction causes trouble where bad water conditions are encountered and necessitates a complete condenser replacement, since single tube or coil replacement is impossible. These condensers may be either vertical or horizontal and are sized on the same design data as the larger condensers. Frequently water velocities are rather low, resulting in lower heat transfer coefficients and much larger amount of surface per ton of refrigeration.

Condensing units operating with the low-pressure refrigerants are sometimes equipped with non-ferrous, double-pipe condensers up to the 2 to 5-hp range; above this shell-and-coil and shell-and-tube condensers are used. (See Chapter 31.)

7. Research on **performance** of ammonia condensers of various types by Linge[7] has brought to light a review and comparison of existing tests on **ammonia condensers.**

Condensation takes place in the superheated part of the condenser as long as its walls are cooled below the condensing temperature. The product of the coefficient of heat transfer and the mean temperature difference between the vapor and the wall is practically the same for superheated vapor and saturated vapor. When the superheat increases, the transmission coefficient decreases to about the same extent as the temperature difference increases; thus the quantity of heat transmission remains about the same. Therefore, an average overall heat transfer coefficient and the mean temperature difference calculated by using condensing temperature corresponding to the saturated condensing pressure and the entering and leaving water temperatures, will give accurate condenser ratings.

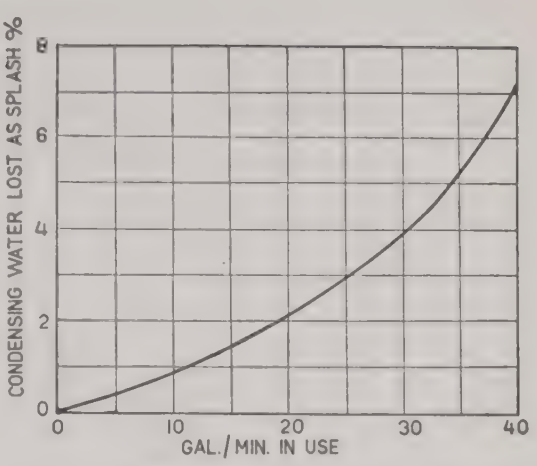

Fig. 12. Water Loss in Atmospheric Condenser, 70F Water, 12 2-in. Pipes High, 20 ft Long

It has not been possible to determine with accuracy the coefficients of heat transfer in the **subcooling regions** of the different condensers, because the temperature differences and heat quantities are so small that their measurement is subject to considerable error. These values range from 30 to 60 Btu per sq ft hr F. Hence there is no advantage with big condensers in using a portion of the surface for subcooling which may, however, be advantageously obtained by using a separate cooler.

The main factors affecting the performance of all types of condensers are the **velocity** of the water and the **thickness of the scale** on the tubes. The curves obtained from the observations were always found to be located between those calculated for clean and for scaled condenser surface.

A tabulation of all the test results shows that atmospheric condensers have the slowest water speed and the lowest coefficient of transmission, and that the shell-and-coil condensers with the water through the coil have the highest water speed and the highest coefficient of transmission.

Table 5 from the same source contains factors for conversion of test results ob-

Table 5. Relative Heat Transfer with Various Refrigerants under Identical Conditions

NH_3	SO_2	CO_2
100	98	93
200	192	174
400	370	315

tained with ammonia into values pertaining to other refrigerants. These values were derived by theoretical analysis based on Nusselt's theory of condensation.

Evaporative Condenser

8. The evaporative condenser combines into one piece of equipment the functions of the condenser and cooling tower and thus takes the place of the water-cooled condenser, the water circulating system and the cooling tower.

Water is sprayed over the outside of the condenser coil in which the discharge gas from the compressor is condensed, and in so doing absorbs heat. Simultaneously, air is drawn through both the coil and the spray water, thus cooling the water by evaporation to a point between the entering air wet-bulb temperature and the refrigerant condensing temperature. Hence the water acts as a heat transfer medium, absorbing heat of condensation which is transmitted through the metal coil and in turn giving up this heat to the air. The total heat surrendered by the condensing refrigerant is ultimately transferred to the air and results in an increase in the total heat content of the air. While the principle of evaporative cooling is not new, the application as found in the evaporative condenser is a comparatively recent outgrowth of the air conditioning industry. Its general use and acceptance have been hastened because of the restrictions on water supply and sewage systems, the high cost of water, or where space limitations made it impossible to install cooling towers.

Axial or **radial fans** draw or blow air through the water-sprayed condenser coil. These fans are selected to handle relatively large volumes of air at high velocities. The evaporated water is picked up by the air stream and discharged from the unit.

The smaller sizes usually have only one water **spray nozzle** and sometimes the unvaporized water is wasted to the sewer, thus eliminating a recirculating water pump and piping. Larger sizes **(Fig. 13)** utilize low-pressure pumps as part of the assembly, to recirculate the water through the spray nozzles and keep the condenser coil thoroughly wetted. Evaporative condensers are manufactured in factory-assembled units in capacities ranging from 1 to 100 tons of refrigeration; the larger units are usually made in sections for ease of installation and assembly. Larger size installations are practical and are obtained by multiple units.

The water which is vaporized absorbs its heat from the condensing refrigerant in the coil. Each pound of water vaporized in this process absorbs approximately 1,000 Btu. Each pound of water used in a water-cooled condenser absorbs 15 to 30 Btu, depending on the temperature of the water available and the condensing temperature maintained in the refrigeration system. Thus the theoretical water consumption of an evaporative condenser is from 1.5 to 3% of that required by a water-cooled condenser. Because evaporation of the water causes a progressive concentration of salts and deposits in the recirculated water, it is

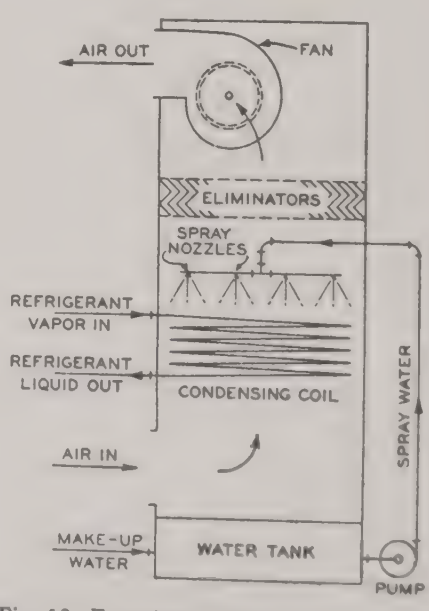

Fig. 13. Functional View of Evaporative Condensers

advisable to waste an additional amount of water so that the actual water consumption is from 5 to 10% for air conditioning installations operating during the summer months only, and from 10 to 15% for a refrigeration installation operating the year 'round. Make-up water is usually controlled by means of a float control in the water tank. Frequently water treatment is necessary to prevent excessive salt concentration and scaling on the condenser coil. If the coil becomes scaled, chemical removal is imperative for economical operation.

In general, the **water consumption** of evaporative condensers is comparable to that of cooling towers since both operate on the same principle of evaporation. Because of their completely enclosed construction and use of eliminators to reduce entrained moisture carry-over, evaporative condensers will have a lower windage loss than atmospheric cooling towers. In addition, the evaporative condenser, handling outdoor air, can produce an additional water saving by operating without water as an air-cooled condenser during comparatively cold winter weather. This may not always be advisable.

Goodman[10] shows a very intimate relationship between **coil surface** per ton of capacity and air quantity. **Fig. 14** indicates that a decrease in air quantity beyond point *A* results in an increase in surface which is out of proportion to the decrease in air quantity. Similarly, a reduction in coil surface beyond point *B* requires a disproportionate increase in air quantity which would result in excessive fan motor horsepower and moisture carry-over. Consequently, evaporative condensers are designed with a coil surface and air quantities in the range of points *A* and *B*, and are usually rated at a fixed standard air quantity.

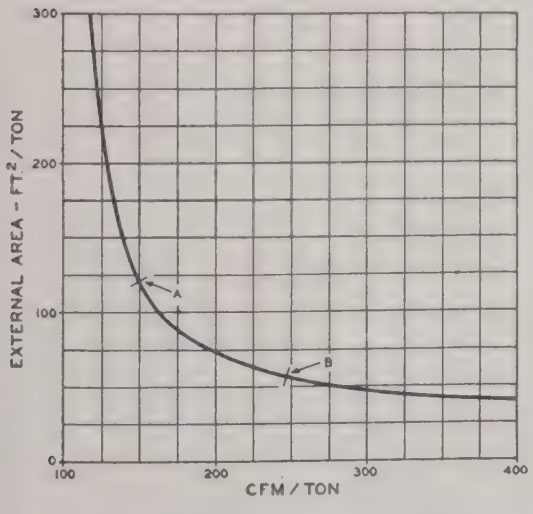

Fig. 14. Air Requirements and Area on Evaporative Condenser for 100F Condensing Temperature and 80F Initial Wet Bulb

The practice among manufacturers of evaporative condensers is to set a limit between 180 to 600 cu ft of air per min per ton. It is considered good practice to select evaporative condensers for Freon-12 having a 20 to 35 F temperature difference between the condensing temperature (saturated temperature of dry refrigerant vapor entering evaporative condenser) and the wet-bulb temperature of the entering air. For ammonia these values are from 15 to 25 F.

The total amount of heat transferred from the refrigerant to the air is equal to the refrigerating load plus the heat of compression. It is more convenient, however, when selecting a compressor and an evaporative condenser to satisfy a specific refrigeration load, to have both units rated in terms of refrigerating effect.[8] When this is done, the capacity of the evaporative condenser operating on a given refrigerant becomes dependent upon (a) suction temperature, (b) condensing temperature, and (c) entering air wet-bulb temperature.

James[9] presents **Fig. 15** showing this relationship in a typical rating curve, and indicates that this curve relationship would become a straight line by plotting capacity percentage versus total heat of the entering air. These rating curves are based upon a constant suction temperature; for other suction temperatures, correction multipliers are indicated as in **Fig. 16.** It is considered good practice to select evaporative condensers for a 20 to 35 F temperature difference between the condensing temperature of the saturated refrigerant vapor and the entering air wet-bulb temperature.

James[9] bases his **overall heat transfer** on external surface and a wet-bulb mean temperature difference, that is, he uses the en-

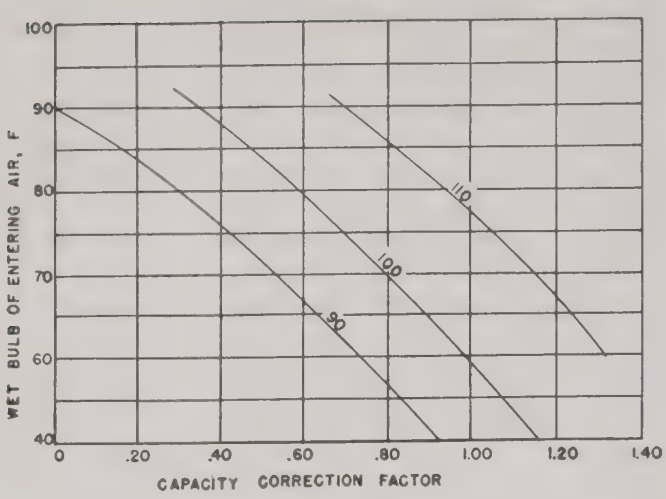

Fig. 15. Typical Rating Curve for Freon-12 Evaporative Condensers at Various Condensing Temperatures, F

tering and leaving air wet-bulb temperature and the saturated condensing temperature for figuring mtd. This permits the use of one simple equation. Goodman[10] states that the transfer of heat from the condensing refrigerant to the air is determined by two completely different mechanisms: (a) the transfer of heat from the condensing refrigerant to the spray water, which depends upon actual temperature differences; and (b) the transfer of heat from the spray water to the air, which depends upon the difference in heat content. This means that one simple formula cannot be used to compute transfer of heat from the refrigerant to the air; two separate steps are needed. Before using this method, it is necessary to find the temperature of the spray water. The transfer of heat from the condensing refrigerant to the spray water and from the spray water to the air can then be computed.

Standard **rating conditions**[3] are based on the following conditions: (a) barometric pressure, 29.92 in. of mercury; (b) entering air dry bulb, 90 F; (c) entering air wet bulb, 75 F; (d) ambient air dry bulb, 90 F; (e) water entering unit, 75 F; (f) saturated temperature of dry refrigerant vapor entering unit, 105 F; (g) actual temperature of dry refrigerant vapor entering unit, not less than 125 F; (h) ratio of net refrigerant heat rejection effect to the net refrigeration effect, assumed as 1.20.

In regard to location, several items must be considered:

(a) **For indoor installation, summer operation only,** the evaporative condenser can be used to aid in ventilating the enclosure in which it is placed. The unit will remove air and create air movement in proportion to its size. If ductwork is required for the exhaust or intake, the fan motor should be adjusted in accordance with the new static pressure.

(b) **For indoor installation, year 'round operation,** the unit may serve the ventilation purposes in summer mentioned under (a), but in winter it should be supplied with outside air by means of ducts to prevent imposing an overload on the heating system by exhausting large amounts of heated air. Care should be exercised in locating the inlet and outlet so that the air does not short circuit and recirculate. Motor sizes should be ad-

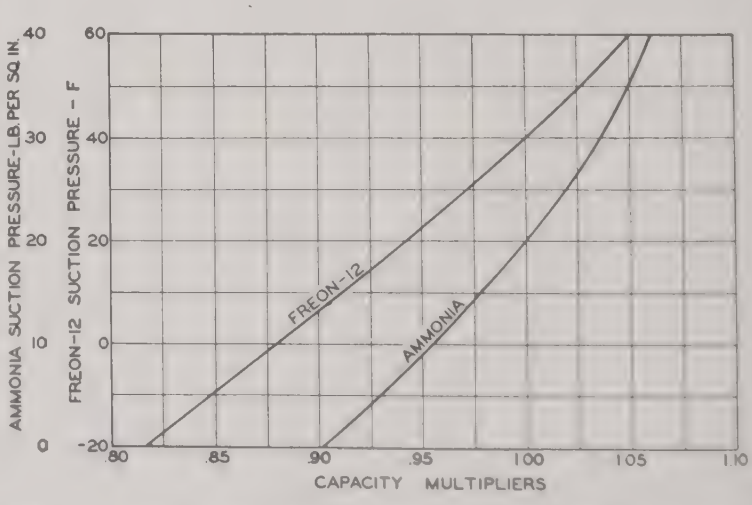

Fig. 16. Suction Pressures and Temperature Correction Factor for Ammonia and Freon-12 Evaporative Condensers

justed to the new static pressure.

(c) **For outdoor installation, summer operation only,** the unit should be treated with special reflective and corrosion-retardant paint, and the liquid receiver, if mounted outside, should be shielded from the direct rays of the sun. Care should also be exercised to protect the unit from neighboring corrosive fumes.

(d) **For outdoor installation, year 'round operation,** the unit should receive the considerations under (c), plus the advisability of having a drain tank and pump mounted inside the building to prevent freezing in winter. For year 'round operation, indoor installations with intake and exhaust ducts are preferred.

Evaporative condensers are made for ammonia as well as the low-pressure refrigerants. The coils in an ammonia unit are made of steel pipe or tubing, whereas in low-pressure refrigerant units they are made of non-ferrous or steel tubing, hot dipped galvanized after fabrication. Either finned or all prime surface coils are available for the common applications. Liquid subcooling coils and desuperheating coils are also included in some units.

Metal parts exposed to the action of the recirculating water should be made of such material, or the material properly coated, as to provide good corrosion resistance. All painting should be made accessible for repainting.

Horizontal Closed Shell-and-Tube Freon Condenser

9. Horizontal closed shell-and-tube Freon-12 condensers are built in sizes from about five tons up to those required on the largest installations; however, on these large jobs, greater operating economy is obtained by a multiple condenser installation.

Freon-12 condensers have been made with many types of material specifications, including all prime surface ferrous tubes, and all prime surface or finned, non-ferrous tubes. With the background of performance experience on horizontal ammonia condensers with steel and with the multiplicity of tube types on Freon condensers, efforts were made to resolve the performance of Freon condensers down to their fundamental characteristics.

The computation of **overall heat transfer**[1] in a condenser may be made from test data of heat transfer coefficients of the water and refrigerant sides, physical measurements of the condenser, assumed fouling factors and the following equation:

$$U = \frac{1}{\frac{R}{h_i} + \frac{R}{h_f} + \frac{1}{h_o} + \frac{L}{k}}$$

where

U = overall heat transfer coefficient, Btu per sq ft hr F, based on the external surface and the mean temperature difference between the external and internal surfaces.
R = ratio of external surface to internal surface.
h_i = internal or water side film coefficient, Btu per sq ft hr F
h_f = fouling allowance on water side, Btu per sq ft hr F
h_o = external or refrigerant side coefficient, Btu per sq ft hr F
L = thickness of tube wall, in.
k = thermal conductivity of tube material, Btu in. per sq ft hr F

Note: $\frac{L}{k}$ applies to prime surface. When extended surface is used, this factor should be modified.

Values of h_i are reliably and accurately reported by McAdams.[12] For turbulent flow DG/μ exceeding 10,000 in horizontal tubes and using average water temperatures, he cites the general correlation for condensing:

$$\frac{h_i D}{k} = 0.0225 \left(\frac{DG}{\mu}\right)^{0.8} \left(\frac{c_p \mu}{k}\right)^{0.4}$$

where:

D = inside tube diameter, ft
k = thermal conductivity, Btu per sq ft hr F
G = weight velocity, lb per hr sq ft of cross section
μ = viscosity, lb per hr ft, based on arithmetic mean temperature of the water = 2.42 × viscosity-centipoises
c_p = specific heat, Btu per lb F

For a given condenser tube and a given range of operating conditions, this equation may be resolved into $h_i = cv^{0.8}$, where c is calculated from the above equation with G resolved into velocity v fps.

Pownall[11] reports that after the oxide film on non-ferrous tubes has been built up with normal waters, its resistance becomes constant and gives a scale factor, h_f, of 4,000, equal to that of non-ferrous tube condensers operating for three years with normal water. Ferrous tubes, however, seem to have an increasing scale resistance because of the continuous oxidation of inside tube surfaces, even with normal water.

For general use, the values of **fouling factors** h_f have been recommended[1] as 2,000 for copper tube and 1,000 for steel tube.

The values of external or refrigerant-side coefficient, h_o, must be determined from test data, since they vary with the physical construction of the tube. Jones[13] cites tests with Freon-11 and $\frac{5}{8}$-in. O.D. (0.545-in. I.D.) copper tubes, having sixteen $\frac{1}{16}$-in. high fins per inch, in which the refrigerant film varied markedly and directly with the rate of loading of the surface. The same tests indicated practically no evidence of important changes in refrigerant film coefficient with respect to mean temperature differences or water velocities. These tests were run with high water velocities, 8.1 to 9.2 fps, but because of the improved heat transfer, two passes of water flow are sufficient, where four are usually necessary for plain tubes. Thus the total water friction loss remains about the same. **Fig. 17** is a typical rating. See Paragraph 3 for a general description of the method used for this type of rating curve.

Non-Condensible Gases

10. Non-condensible gases are a combination or mixture of nitrogen, hydrogen, oxygen, chlorine, oil vapors, water vapor, etc. inside the refrigerating system. These gases circulate with the refrigerant, but unlike the refrigerant, they do not liquefy in the condenser, where they collect and on certain types of systems settle in the receiver (See also Chap. 34).

When these gases are present, they will raise the condensing pressure above that corresponding to the condensing temperature by forming a film over the condensing surface or some part of it, thus acting as a resistance to efficient heat transfer.

Due to the continuous flow of the gases entering the condenser, the non-condensible gas film formed around the condensing surface is wiped away and these gases then accumulate in the quiet parts of the condenser, tending to blanket the condensing surface.

This reduction of the effectiveness of the condenser not only causes higher head

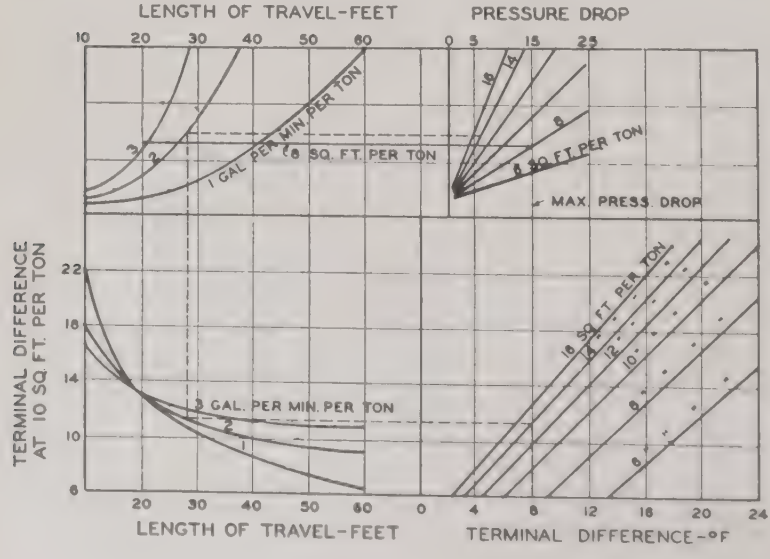

Fig. 17. Rating Curve for Horizontal Freon-12 Condenser with $\frac{5}{8}$-in. O.D. Tubes Having 8 Fins per Inch, $\frac{1}{4}$-in. High[1, 11]

Example: For 8F difference, 14 sq ft per ton and 2 gpm per ton, find length of water travel and pressure drop. Read vertically on 8° temperature difference line to intersection with 14 sq ft per ton, then horisontally to 2 gpm per ton line. Read 29-ft travel to intersection with 2 gpm and then horisontally to 14 sq ft per ton, finding 6.5 psi pressure loss.

pressures, but also increases brake horsepower, reduces capacity and promotes oxidation of the oil, especially at the valves due to the presence of oxygen at this point of high discharge temperatures. Operating experience with all refrigerating systems substantiates this. According to Badger,[17] who presents data for steam and air, the film resistance coefficient between steam and a metal wall is as much as six times as great with 4.5% air as with no air present.

As reported by Linge,[7] the presence of these gases according to Dalton's law of partial pressures raises the condensing pressure P by the amount of the partial pressure, P'. The increase of the work of compression is in relation to the ratio $P+P'/P$. This is greater with higher partial pressures and lower condensing pressures.

Non-condensible gases come (a) from outside the system by being drawn into the crankcase of the compressor along with the oil and through leaks in piping, stuffing boxes, etc., when the system is being operated under a vacuum and (b) from inside the system by improperly evacuating a new system before charging, by decomposition of the lubricant under the heat of compression, and by miscellaneous chemical reactions within the system.

A test for non-condensible gases is to shut down the refrigeration system, but to allow the condenser water to flow for a sufficient length of time so that the refrigerant is at the same temperature as the water. If the condenser pressure is higher than that pressure corresponding to the refrigerant temperature, there are non-condensible gases present. Experimental data have shown that with these gases present in the condenser under the above conditions with a 10 F difference between refrigerant temperature and the temperature corresponding to the condensing pressure, there will be as much as 20 to 25 F increase in condensing temperature during operation.

Non-condensible gases can be eliminated to a great degree by **purging.** This may be accomplished by shutting down the system as mentioned above, and by blowing off directly into the atmosphere or by using devices for this purpose which work while the system is in operation. Purging connections should be located in the system where non-condensible gases will accumulate. It has been found on long horizontal condensers with the discharge gas inlet in the center of the shell, that these gases will accumulate at the extreme ends and will not work their way down into the receiver. The purging connections should therefore be on the top two extreme ends of this type of condenser.

On any coil type condenser, such as evaporative condensers, where the actual refrigerant gas velocity through the coil is high, the non-condensible gases are pushed down into the receiver and will remain there during normal operation. This type of condenser can be purged very readily from the top of the receiver. However, on ammonia condensers using a number of refrigerant passes in parallel, these gases have been known to pocket in one or more of the passes, rendering them ineffective.

Purging devices employ a refrigerated surface within a container into which is introduced a mixture of refrigerant gas and non-condensible gases. The low temperature surface condenses a great amount of the refrigerant; the remaining mixture is allowed to escape to the atmosphere. **Fig. 18** shows the refrigerant loss when purging ammonia and Freon-12 at various temperatures and pressures, and indicates the desirability of purging devices. Even though Freon-12 losses are considerably greater than ammonia, it is economical to purge these systems as thoroughly as ammonia systems. This can be determined by balancing the savings in compressor power requirements and increased wear on machinery against the loss of refrigerant.

Although it is desirable to operate without non-condensible gases, exceedingly low condensing pressures are also to be avoided. In the endeavor to reduce condensing pressures, the minimum pressure must be taken into account; i.e., the pressure below which the system will not operate satisfactorily because of existing static heads and resistances of mains and control apparatus to the flow of the refrigerant. Condensing pressures are best increased for this purpose by decreasing the water flow.

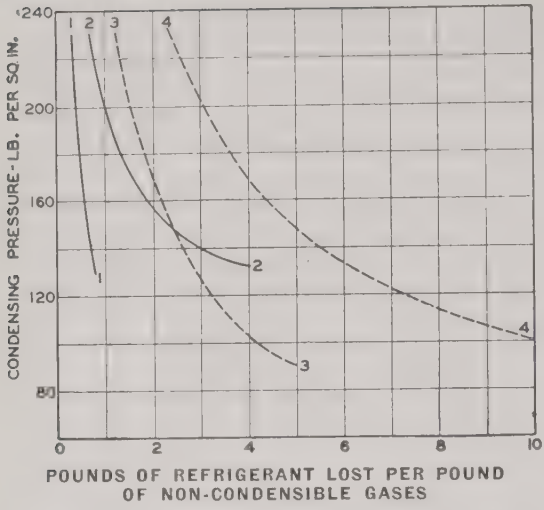

Fig. 18. Ammonia and Freon-12 Loss When Purging at Different Gas Temperatures and Pressures

1—Ammonia, 45F; 2—ammonia, 70F; 3—Freon-12, 45F; 4—Freon-12, 70F

Fouling Factors

11. Frequently the progressive decrease in capacity of shell-and-tube or shell-and-coil condensers is due to the fouling on the water side of the tube. In order that condensers will have sufficient excess tube surface to maintain satisfactory performance in normal operation, with reasonable service time between cleanings, it is recommended that standard minimum fouling factors be used. For conditions of extreme fouling, poor maintenance and the like, higher fouling factors should be used. These factors are the reciprocal of the heat transfer rate of the layer of foreign substances deposited on the surface of the heat transfer tubing, and enter into the equation as explained under Freon condensers. In order to obtain the total resistance to heat transfer, the sum of fouling resistance as selected is added to the resistance of the fluid film on the inside of the tubes and on the outside of the tubes and the resistance of the tube wall. The effect of fouling resistances in reducing clean coefficients is shown in **Fig. 19. Table 6**[18] lists the fouling factors for various types of water.

12. Design water temperatures for condensers are tabulated in **Table 7** along with the source of the water for most cities in the United States over 20,000 population, according to a 1937 survey.[14] These values are the highest design water temperatures occurring from June through September; however, this temperature occurred in August in the majority of the localities. The source of water supply is indicated by *S* for surface water and *W* for well; cities having both surface and well water have the temperature of the surface water tabulated. The **design wet-bulb temperatures** for selection of evaporative condensers or cooling towers also have been included.[15,16] It may be deisrable to check local conditions for water and wet-bulb temperatures.

13. Where state or municipal codes do not take precedence, **design pressures, materials, welding, tests,** and **relief devices** are recommended to be in accordance with ASME Unfired Pressure Vessels Section of the Boiler Construction Code and ASRE Standard 15, which is the American Standards Association Code B9, Safety Code for Mechanical Refrigeration.

14. The condenser, being a part of the refrigeration cycle high-side, is necessarily used with other more or less essential accessories; of these the important items are the **liquid receivers** and **oil separators.**

In some cases the condenser may also be used as the liquid receiver, but a separate receiver is more frequently advisable and necessary, since liquid accumulation in a condenser blocks off condensing surface with a consequent reduction in capacity or an abnormal increase in condensing pres-

Table 6. Fouling Factors for Various Types of Condenser Water

Types of water	Fouling factor
Sea water	.0005
Brackish water	.001
Cooling tower and artificial spray pond:	
Treated make-up	.001
Untreated	.003
City or well water	.001
Great Lakes	.001
River water	
Minimum	.001
Mississippi	.002
Delaware, Schuylkill	.002
East River and N.Y. Bay	.002
Chicago Sanitary Canal	.006
Muddy or silty	.002
Hard (over 15 gr per gal)	.003
Engine jacket	.001
Distilled	.0005

sures. It is desirable to have the liquid receiver of such size that it can hold the entire refrigerant charge, but this is impossible at times, especially on systems with large flooded evaporators. In all cases, the minimum receiver volume should be such that it can absorb the seasonal fluctuations in liquid refrigerant requirements; the liquid seal in the receiver must never be broken, and the liquid level should never build up into the condenser. Although the receiver is a part of the high-side, its size is predicated upon the design and operation of the low-side.

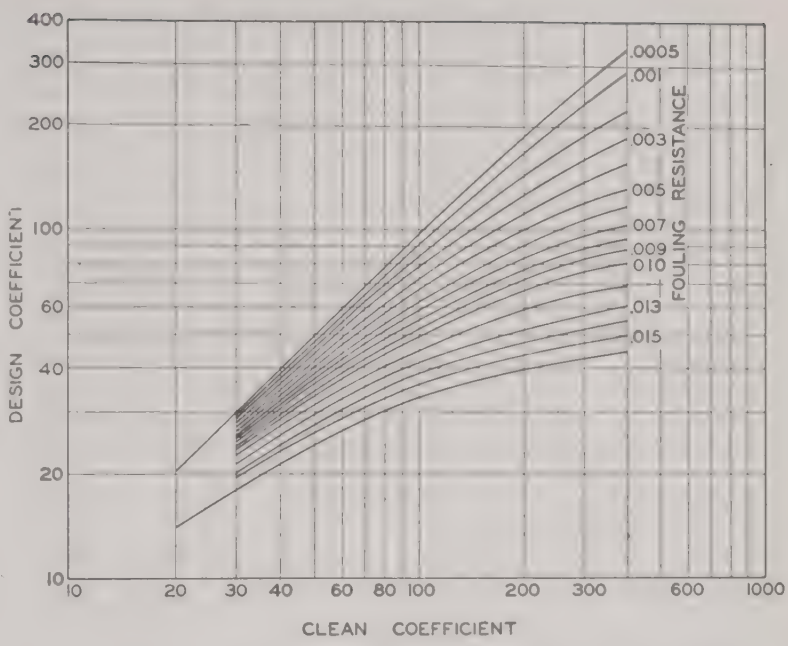

Fig. 19. Effect of Fouling Resistance on Heat Transfer Coefficients

Sizes recommended[1] for ammonia liquid receivers are from 8 in. to 36 in. in diameter, with lengths varying from 6 to 16 ft, depending upon diameter. Freon-12 liquid receivers are built in the same sizes with connections suitable for Freon-12 mains. Many special receivers are made for evaporative condensers, and condensing units with design efforts made to balance the size of the receiver with the capacity of the unit so that the receiver will fit the majority of installations.

Discharge oil separators are traps installed between the compressor and condenser to remove the major portion of the compressor oil carried along by the superheated discharge gas. These separators introduce large volumes into the discharge main to reduce gas velocities, thus permitting the oil particles to be separated by filter, centrifugal, or impingement type separators. The oil removed in the separator prevents fouling of the condenser and evaporator and results in an increased plant efficiency. These separators should be installed close to the condenser so that the pulsating effect of the compressor is reduced as much as possible and also to permit the superheated vaporized oil to form into droplets.

The separated oil on ammonia or Freon-12 installations may be drained automatically back into the compressor crankcase; however, on ammonia plants it is customary to discard this oil or reclaim it by acceptable methods.

Mean Temperature Difference

15. When there is any great change in the temperature difference between two media at different parts of the surface of heat transfer apparatus, it is customary to use the *logarithmic*, rather than the *arithmetic*, **mean temperature difference** in computing heat transfer. This is based on the assumption that the coefficient of heat transfer for all parts of the surface is constant, and that therefore the rate at which heat is transferred through any portion of the surface is proportional to the temperature difference existing there. The formula is

$$D = \frac{\Delta_1 - \Delta_2}{\text{Log}_e \frac{\Delta_1}{\Delta_2}}$$

where D = logarithmic mean temperature difference

Table 7. City Water Temperatures and Design Wet-Bulb Temperatures

City	Water source*	Water temp, F	Design wet bulb, F	City	Water source*	Water temp, F	Design wet bulb, F
Abilene, Tex.	S-W	80.0	74	Chattanooga, Tenn.	S	79.5	77
Akron, Ohio	S	74.7	74	Chester, Pa.	S	79.0	77
Alameda, Calif.	S	64.0	65	Cheyenne, Wyo.	—	—	65
Albany, N.Y.	S	66.0	75	Chicago, Ill.	S	69.4	75
Albuquerque, N.M.	W	72.0	66	Chicago Heights, Ill.	W	55.0	75
Alexandria, La.	W	86.0	80	Chicopee, Mass.	S	66.7	74
Alexandria, Va.	S	74.0	78	Cicero, Ill.	S	72.0	75
Alhambra, Calif.	W	68.0	70	Cincinnati, Ohio	S	82.0	78
Allentown, Pa.	S-W	70.0	75	Clarksburg, W.Va.	S	77.0	73
Alliance, Ohio	S-W	68.0	74	Cleveland, Ohio	S	73.5	75
Amarillo, Tex.	W	65.0	71	Clifton, N.J.	S	55.0	76
Ambridge, Pa.	W	66.0	73	Colorado Springs, Colo.	S	62.0	64
Amsterdam, N.Y.	S	57.0	73	Columbus, Ga.	S	80.0	79
Anderson, Ind.	S-W	87.0	75	Columbus, Ohio	S	76.0	75
Ann Arbor, Mich.	W	62.5	73	Concord, N.H.	S-W	73.0	72
Anniston, Ala.	W	67.0	77	Council Bluffs, Ia.	S	81.5	75
Appleton, Wis.	S	76.0	73	Covington, Ky.	S-W	82.0	77
Asheville, N.C.	S	76.0	75	Cranston, R.I.	—	65.0	74
Ashland, Ky.	S	82.4	76	Cumberland, Md.	S	65.0	73
Atlanta, Ga.	S	83.5	75	Dallas, Tex.	S-W	82.5	78
Atlantic City, N.J.	S-W	72.5	77	Danville, Ill.	S	84.0	76
Attleboro, Mass.	S-W	68.7	74	Davenport, Ia.	S	83.9	76
Auburn, N.Y.	S	69.0	73	Dearborn, Mich.	—	75.0	73
Augusta, Ga.	S	85.0	78	Decatur, Ill.	S	83.3	76
Aurora, Ill.	W	60.0	75	Denver, Colo.	S-W	70.9	64
Austin, Tex.	S	85.0	77	Des Moines, Ia.	S	72.9	77
Baltimore, Md.	S	70.0	78	Detroit, Mich.	S	75.0	75
Bangor, Me.	S	72.0	71	Dubuque, Ia.	W	60.0	76
Battle Creek, Mich.	S-W	52.0	73	Duluth, Minn.	S	70.6	71
Bay City, Mich.	S	78.0	72	Durham, N.C.	S	76.2	76
Beaumont, Tex.	S	88.0	79	East Cleveland, Ohio	—	73.5	75
Belleville, Ill.	S	90.0	77	East Providence, R.I.	S	60.0	74
Beloit, Wis.	W	57.0	76	East St. Louis, Ill.	S	90.0	77
Berkeley, Calif.	S	64.0	65	Easton, Pa.	S	65.0	76
Berlin, N.H.	S	66.0	71	Eau Claire, Wis.	—	53.0	75
Bethlehem, Pa.	S-W	75.0	75	Elgin, Ill.	—	56.0	75
Birmingham, Ala.	S	82.0	78	Elizabeth, N.J.	S-W	66.7	76
Bismarck, N.D.	—	—	73	Elkhart, Ind.	W	60.0	74
Bloomington, Ill.	S	78.8	76	El Paso, Tex.	W	85.0	69
Bluefield, W.Va.	S	70.6	74	Elyria, Ohio	S	72.9	73
Boise, Idaho	—	—	65	Erie, Pa.	S	72.9	73
Boston, Mass.	S	74.3	75	Evanston, Ill.	S	69.0	75
Braddock, Pa.	S	80.0	73	Evansville, Ind.	S	85.0	78
Bradford, Pa.	S-W	60.0	73	Everett, Wash.	S	62.0	63
Bridgeport, Conn.	S	64.0	75	Fargo, N.D.	S	85.1	70
Brockton, Mass.	S	62.0	73	Findlay, Ohio	S-W	63.0	74
Brookline, Mass.	W	67.0	73	Flint, Mich.	S	72.0	72
Brownsville, Tex.	S	87.0	79	Fond du Lac, Wis.	W	56.0	73
Buffalo, N.Y.	S	73.0	75	Fort Dodge, Ia.	W	52.0	76
Burlington, Ia.	S	87.0	76	Fort Smith, Ark.	S	80.0	76
Burlington, Vt.	S	66.0	73	Fort Wayne, Ind.	S	79.0	74
Butler, Pa.	S	70.0	73	Fort Worth, Tex.	S	83.0	76
Cambridge, Mass.	S	74.0	73	Freeport, Ill.	W	56.0	76
Camden, N.J.	W	58.0	76	Fresno, Calif.	W	72.0	70
Carbondale, Pa.	S	70.0	74	Gadsden, Ala.	S	82.0	77
Cedar Rapids, Ia.	S	82.5	76	Galesburg, Ill.	W	67.0	76
Central Falls, R.I.	—	74.0	74	Galveston, Tex.	—	—	80
Charleston, S.C.	S	85.0	80	Gary, Ind.	S	70.0	75
Charlestown, W.Va.	S	83.0	74	Glendale, Calif.	S-W	68.2	68
Charlotte, N.C.	S	81.0	73	Grand Rapids, Mich.	S	74.0	73

* S = spring; W = well.

Table 7. City Water Temperatures and Design Wet-Bulb Temperatures (Continued)

City	Water source*	Water temp, F	Design wet bulb, F	City	Water source*	Water temp, F	Design wet bulb, F
Granite City, Ill.	—	90.0	77	Mount Vernon, N.Y.	S	68.0	75
Greenville, S.C.	S	76.0	76	Muncie, Ind.	S-W	75.5	75
Hagerstown, Md.	S	83.0	74	Muskegon, Mich.	S	68.0	72
Hamilton, Ohio	W	65.0	76	Muskogee, Okla.	S	90.7	77
Hammond, Ind.	S	82.0	74	Nashua, N. H.	W	67.0	72
Hannibal, Mo.	S	80.0	77	Nashville, Tenn.	S	88.0	78
Hartford, Conn.	S	70.0	74	New Albany, Ind.	S	83.0	78
Hazelton, Pa.	S	65.0	74	Newark, N.J.	S	71.5	76
Helena, Mont.	—	—	67	Newark, Ohio	S	70.0	76
Highland Park, Mich.	S	74.0	72	New Bedford, Mass.	S	71.0	73
High Point, N.C.	S	78.0	75	New Brunswick, N.J.	S	71.0	76
Holyoke, Mass.	S	74.0	73	Newburgh, N.Y.	S	54.0	75
Houston, Tex.	W	84.0	78	New Haven, Conn.	S	70.0	75
Hutchinson, Kan.	W	60.0	74	New Orleans, La.	S	90.0	79
Indianapolis, Ind.	S-W	82.0	76	Newport, R.I.	S	70.2	74
Ithaca, N.Y.	S	71.6	73	Newport News, Va.	S-W	82.0	78
Jackson, Mich.	W	52.0	73	New York, N.Y.	S-W	70.0	75
Jackson, Miss.	S	82.0	79	Niagara Falls, N.Y.	S	69.8	72
Jackson, Tenn.	W	70.0	79	Norfolk, Va.	S	83.0	78
Jacksonville, Fla.	W	86.7	78	Norristown, Pa.	S	84.0	75
Janesville, Wis.	W	52.0	75	North Little Rock, Ark.	S	85.0	78
Jefferson City, Mo.	S	80.0	76	North Tonawanda, N.Y.	S	78.0	72
Johnstown, Pa.	S	59.5	72	Norwood, Ohio	W	56.0	77
Kalamazoo, Mich.	W	52.0	73	Oakland, Calif.	S	64.0	70
Kansas City, Kan.	S	93.0	76	Oak Park, Ill.	S	75.0	75
Kansas City, Mo.	—	—	76	Ogden, Utah	S-W	60.0	65
Kenosha, Wis.	S	67.0	74	Oil City, Pa.	W	65.0	73
Knoxville, Tenn.	S	84.3	75	Oklahoma City, Okla.	S	77.2	76
LaCrosse, Wis.	W	52.0	75	Omaha, Nebr.	S	80.9	75
Lafayette, Ind.	W	53.0	76	Orlando, Fla.	S	87.0	79
Lancaster, Pa.	S	77.0	75	Oswego, N.Y.	S	58.0	72
Lansing, Mich.	W	59.0	72	Ottumwa, Ia.	S	77.5	76
Laredo, Tex.	S	84.6	77	Paducah, Ky.	S	89.0	78
Leominster, Mass.	S	74.0	73	Parkersburg, W.Va.	—	—	75
Lima, Ohio	S	77.4	74	Pasadena, Calif.	S-W	74.0	68
Lincoln, Nebr.	W	59.0	75	Passaic, N.J.	S	55.0	76
Little Rock, Ark.	S-W	88.0	78	Paterson, N.J.	S	64.0	76
Los Angeles, Calif.	S-W	76.0	70	Pawtucket, R.I.	S	74.0	74
Louisville, Ky.	S	82.0	76	Pensacola, Fla.	W	70.0	79
Lowell, Mass.	W	50.0	72	Peoria, Ill.	W	56.0	76
Lubbock, Tex.	W	67.0	72	Perth Amboy, N.J.	W	54.0	76
Lynchburg, Va.	S	73.0	75	Petersburg, Va.	S	78.0	78
McKeesport, Pa.	S	80.6	73	Philadelphia, Pa.	S	79.0	78
Madison, Wis.	W	53.0	75	Phoenix, Ariz.	—	—	76
Manchester, N.H.	—	—	73	Pine Bluff, Ark.	W	80.0	79
Manitowoc, Wis.	W	58.0	73	Pittsburgh, Pa.	S	80.6	75
Mansfield, Ohio	W	50.0	74	Pontiac, Mich.	W	55.0	72
Marion, Ind.	W	55.0	75	Portland, Me.	S	66.0	73
Mason City, Ia.	W	59.0	76	Portland, Ore.	S	62.0	65
Massillon, Ohio	S-W	55.0	74	Portsmouth, Ohio	S	80.6	76
Maywood, Ill.	W	60.0	75	Poughkeepsie, N.Y.	S-W	76.6	74
Memphis, Tenn.	—	—	78	Providence, R.I.	S	65.0	75
Meridian, Miss.	S-W	86.0	79	Pueblo, Colo.	S	74.0	64
Michigan City, Ind.	S	71.0	74	Quincy, Ill.	S	82.2	76
Milwaukee, Wis.	S	60.6	75	Quincy, Mass.	S	74.0	73
Minneapolis, Minn.	S	80.2	75	Racine, Wis.	S	68.5	74
Mishawaka, Ind.	W	57.0	74	Raleigh, N.C.	S	83.4	78
Mobile, Ala.	—	—	80	Reading, Pa.	S	78.8	75
Moline, Ill.	S	85.9	76	Reno, Nev.	—	—	65
Montgomery, Ala.	W	71.0	78	Richmond, Va.	S	80.4	78

* S = spring; W = well.

Table 7. City Water Temperatures and Design Wet-Bulb Temperatures (Concluded)

City	Water source*	Water temp, F	Design wet bulb, F	City	Water source*	Water temp, F	Design wet bulb, F
Riverside, Calif.	W	74.0	70	Superior, Wis.	W	64.4	71
Rochester, Minn.	W	58.0	75	Syracuse, N.Y.	S	70.4	73
Rochester, N.Y.	S	68.9	72	Tacoma, Wash.	S-W	61.0	64
Rockford, Ill.	W	55.0	76	Tampa, Fla.	S	87.0	79
Rock Island, Ill.	S	80.0	76	Taunton, Mass.	S	70.0	73
Rome, Ga.	S	78.0	77	Texarkana, Tex.	S-W	86.0	79
Sacramento, Calif.	S	80.6	71	Toledo, Ohio	S	84.0	73
Saginaw, Mich.	S	78.0	72	Topeka, Kan.	S-W	86.0	75
St. Cloud, Minn.	S	77.0	72	Trenton, N.J.	—	—	78
St. Louis, Mo.	S	85.0	78	Troy, N.Y.	S	72.5	73
St. Louis County, Mo.	S	84.0	78	Tucson, Ariz.	W	80.0	70
Salina, Kan.	W	62.0	75	Tulsa, Okla.	S	81.8	76
Salt Lake City, Utah	S-W	58.0	67	Tuscaloosa, Ala.	S	71.0	78
San Angelo, Tex.	S	80.0	74	Uniontown, Pa.	S-W	68.0	73
San Antonio, Tex.	W	76.0	78	Utica, N.Y.	S	70.2	73
Sandusky, Ohio	S	75.2	73	Vicksburg, Miss.	—	—	78
San Francisco, Calif.	S-W	60.0	65	Waco, Tex.	S	84.6	78
San Jose, Calif.	S-W	73.0	68	Warren, Ohio	S	78.6	74
Santa Ana, Calif.	W	69.0	67	Washington, D.C.	S	75.0	78
Santa Barbara, Calif.	S	70.0	65	Waterbury, Conn.	S	74.0	75
Santa Fe, N.M.	—	—	65	Watertown, N.Y.	S	72.0	72
Savannah, Ga.	—	—	78	Waukegan, Ill.	S	69.8	74
Schenectady, N.Y.	W	57.0	73	Wausau, Wis.	W	60.0	72
Scranton, Pa.	S	70.9	74	Westfield, Mass.	S	63.0	74
Seattle, Wash.	S	62.0	65	West Palm Beach, Fla.	S	86.2	79
Shamokin, Pa.	S	65.0	74	Weymouth, Mass.	S	75.2	73
Sharon, Pa.	S	73.0	73	Wheeling, W.Va.	S	80.9	73
Shawnee, Okla.	S	55.0	77	Wichita, Kan.	—	—	75
Sheboygan, Wis.	S	65.0	73	Wichita Falls, Tex.	S	90.0	76
Shreveport, La.	S	91.0	79	Wilkinsburg, Pa.	S	79.0	73
Sioux Falls, S.D.	W	55.0	75	Williamsport, Pa.	S-W	66.0	73
Spartanburg, S.C.	S	78.8	78	Wilmington, Del.	S	79.0	79
Spokane, Wash.	—	—	65	Wilmington, N.C.	S	86.0	79
Springfield, Ill.	S	80.6	77	Winona, Minn.	W	52.0	75
Springfield, Mass.	S	55.0	74	Woodbridge, N.J.	S	51.8	76
Springfield, Mo.	S-W	76.0	76	Woonsocket, R.I.	S	72.0	74
Stamford, Conn.	S	76.0	75	Worcester, Mass.	S	73.0	73
Steubenville, Ohio	S	68.0	74	Yakima, Wash.	S	65.0	63
Stockton, Calif.	W	70.0	71	Yonkers, N.Y.	S	79.0	75
Stratford, Conn.	S	64.0	75	Youngstown, Ohio	S	68.5	74

* S = spring; W = well.

Δ_1 = difference between condensing temperature and entering water temperature

Δ_2 = difference between condensing temperature and leaving water temperature

Solution requires reference to a table of natural logarithms, which is not always at hand. The formula may also be written

$$D = \frac{\Delta_1 - \Delta_2}{2.3026 \log_{10} \frac{\Delta_1}{\Delta_2}}$$

in which case a table of common logarithms is suitable.

Also, it frequently happens that either Δ_1 or Δ_2 is not known, but D is known; or that neither Δ_1 nor Δ_2 is known, but that D and $(\Delta_1 - \Delta_2)$ are known; or that $\left(\frac{\Delta_1 - \Delta_2}{D}\right)$ is known, together with Δ_1 or Δ_2. In any of these cases a trial and error solution is required.

Fig. 20 was prepared by A. B. Stickney[19] to eliminate the need of a table of logarithms, to facilitate computation, and to

obviate the need of trial and error solutions where possible and otherwise to simplify them. By its use, knowing any two of the five quantities D, Δ_1, Δ_2 $(\Delta_1-\Delta_2)$, $\left(\frac{\Delta_1-\Delta_2}{D}\right)$, the other three can be read directly.

In refrigerating engineering, the most frequent cases involve either condensing a refrigerant by heating a fluid, or cooling a fluid by evaporating a refrigerant. In these cases the formulae to be solved take the form

$$sUD = wc(t_1-t_2) = w_r r \text{ in which}$$

s = area of heat transfer surface, sq ft

U = coefficient of heat transfer, $\frac{\text{Btu}}{\text{sq ft hr F}}$

D = logarithmic mean temperature difference.

w = weight of fluid, lb per hr

c = specific heat of fluid

t_1 = temperature of fluid being heated leaving apparatus, *or* temperature of fluid being cooled entering apparatus.

t_2 = temperature of fluid being heated entering apparatus, *or* temperature of fluid being cooled leaving apparatus.

w_r = weight of refrigerant evaporated or condensed, lb per hr

r = the change in heat content in Btu per lb of the refrigerant.

Since the condensation or evaporation takes place at constant temperature,

$$(t_1-t_2) = (\Delta_1-\Delta_2)$$

Examples:

A. Suppose a condenser in which the temperature of the "water on" is 70 F, of the "water off" is 80 F, and of the condensing refrigerant is 85 F. Then $\Delta_1 = 85-70 = 15$; $\Delta_2 = 85-80 = 5$. To find D, read 15 at the bottom of the chart, and follow diagonal upward to the left. Read 5 at the bottom of the chart, and follow diagonal upward to the right. From where these intersect, follow solid curve down to bottom of chart and read $D = 9.1$.

B. A condenser having 1,000 sq ft of surface must handle a load of 100 tons, using 300 gpm of water coming on at 70 F. The coefficient of heat transfer is 150. What is the condensing temperature of the refrigerant? Taking 30 gal deg per ton, the temperature rise of the water is $(100\times30)/300 = 10° = (\Delta_1-\Delta_2)$. 300 gpm is 150,000 lb per hr, the specific heat of water is 1, and therefore

$$D = \frac{150{,}000\times1\times10}{1{,}000\times150} = 10 \text{ F mtd}$$

To solve for Δ_1 and Δ_2, read in $(\Delta_1-\Delta_2) = 10$ at left of chart, and follow dashed curve to right. Read in $D = 10$ at the bottom of chart, and up along solid curve. From where these intersect, read down diagonally to left and right to find, respectively, $\Delta_2 = 5.8$ F and $\Delta_1 = 15.8$ F. The temperature of the condensing refrigerant is $70+15.8 = 85.8$ F.

C. Given the same condenser, the same quantity and temperature of water, and the same coefficient of heat transfer, what tonnage will give a terminal temperature difference of 3 F? And what will be the temperature of the condensing refrigerant?

$$\left(\frac{\Delta_1-\Delta_2}{D}\right) = \frac{sU}{wc} = \frac{1{,}000\times150}{150{,}000\times1} = 1$$

Read $\left(\frac{\Delta_1-\Delta_2}{D}\right) = 1$ at right of chart, and read to left along horizontal line. Read $\Delta_2 = 3$ at bottom of chart, and read upward to right. From point of intersection, read downward to right to find $\Delta_1 = 8.2$, and read to left along dashed curve to find $(\Delta_1-\Delta_2) = 5.2$ F. Taking 30 gal deg per ton, the tonnage is $(300\times5.2)/30 = 52$ tons, and the temperature of the condensing refrigerant is $70+8.2 = 78.2$ F.

Although the above examples have been confined to condensers, cooler problems with an evaporating refrigerant can be solved by the use of the chart along exactly the same lines. In addition, the chart will be found useful in the solution of problems involving heat exchange between two fluids each changing in temperature, providing two of the five quantities are known. In case they are not and a trial and error solution is necessary, the chart can be used in solving the various trial values.

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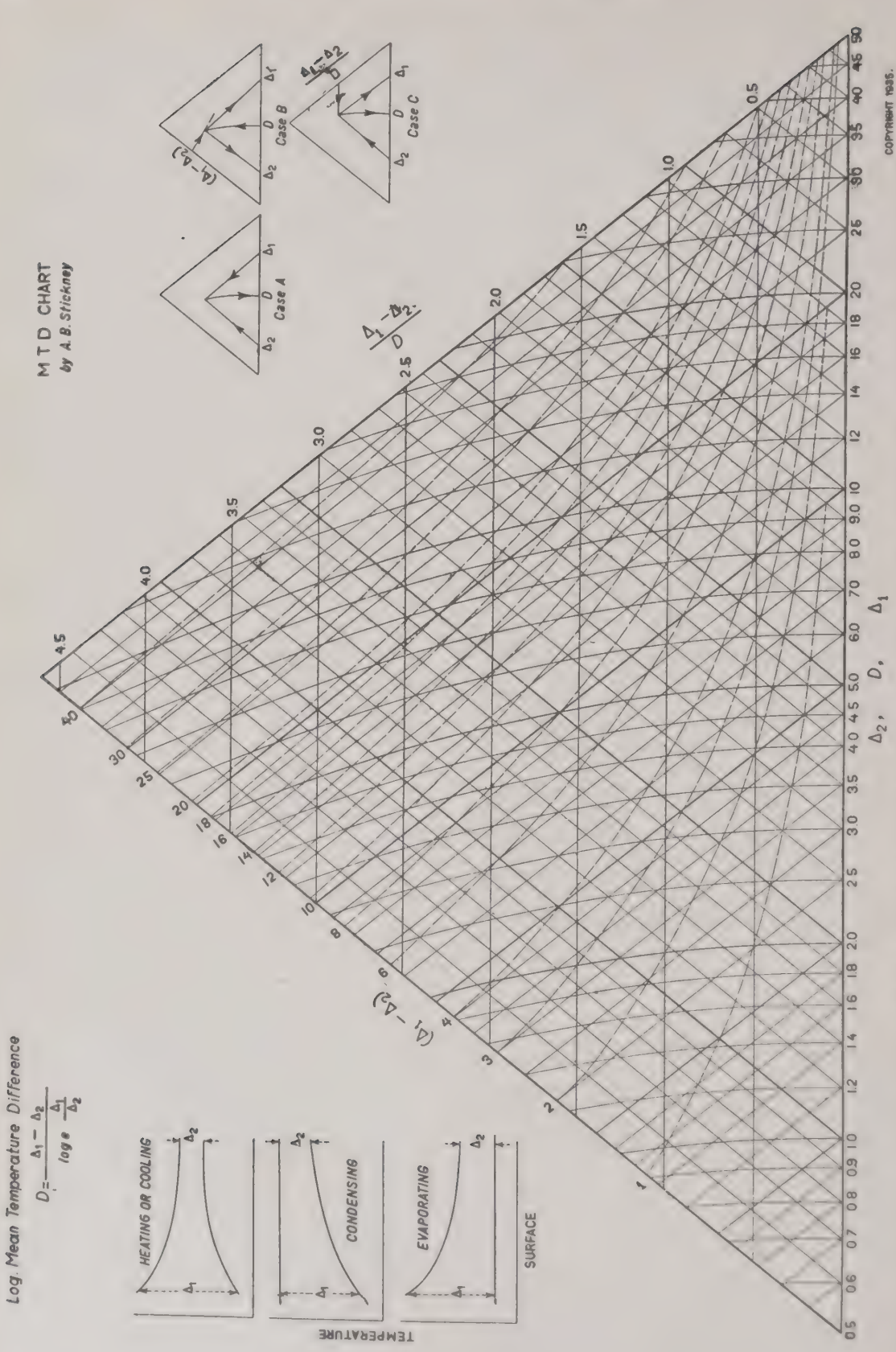

Fig. 20. Chart for Determination of Logarithmic Mean Temperature Differences

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25. EVAPORATORS

THE cooling effect of a refrigerating system takes place in the evaporator, where heat from the medium to be cooled (air, water or brine) vaporizes the liquid refrigerant. As the refrigerant boils away, it is constantly replaced by more liquid from the receiver, while the vapor is returned through the suction piping to the compressor.

This chapter covers various types of ammonia and Freon evaporators in common use for industrial and air conditioning applications.

Good evaporator design is concerned with (a) the production of high **heat transfer,** and (b) sound **practical design** to give long life and freedom from functional troubles such as liquid carry-over, oil logging, and scaling.

The amount of heat absorbed in an evaporator is shown by the following two equations:

$$Q = wc(t_1 - t_2) \qquad (1)$$

where

Q = Btu per hr, given up by the medium
w = lb of medium circulated, lb per hr
c = specific heat of medium, Btu per lb deg F (0.24 for air, 1.0 for water, 0.7 for C_aCl_2 brine of 1.2 sp gr)
t_1 = entering temperature, deg F of medium being cooled
t_2 = leaving temperature, deg F of medium being cooled

$$Q = UAD \qquad (2)$$

where

Q = heat transferred or given up, Btu per hr
U = overall heat transfer coefficient, Btu per sq ft hr deg F, based on outside surface
D = logarithmic mean temperature difference, deg F
A = area of heat transfer surface, sq ft (outside)

Equations (1) and (2) may be combined as below:

$$UAD = wc(t_1 - t_2)$$

$$A = \frac{wc(t_1 - t_2)}{UD}$$

Once the factor U is established, the performance of the cooler may be predicted. U varies materially, however, with design of evaporator, the fluid and its velocity, D, and with fouling of tube surface.

Heat Transfer

Overall heat transfer through a tube is a function of (a) film coefficient on refrigerant side, (b) tube-wall conductivity, (c) film coefficient on fluid side, and (d) ratio of outside to inside tube surface, and is evaluated in the following expression:

$$\frac{1}{U} = \frac{1}{h_f} + \frac{1}{h_{fs}} + \frac{L}{K} + \frac{R}{h_{ws}} + \frac{R}{h_w} \qquad (3)$$

where

U = overall heat transfer factor, Btu per sq ft hr deg F
h_f = film transfer coefficient, Freon side, Btu per sq ft hr deg F
h_{fs} = scale coefficient; Freon side, Btu per sq ft hr deg F
L/K = tube-wall resistance, wall thickness divided by conductivity; wall thickness in feet conductivity in Btu ft per sq ft hr deg F
R = Ratio of outside total surface to inside tube surface
h_{ws} = scale coefficient, water side
h_w = film coefficient, water side

The **refrigerant side coefficient** depends on thorough wetting of the surface, rapidity of boiling and rate of escape of gas bubbles from tube surface. As gas bubbles

form from the boiling action, they tend to cling to the surface of the tube, reducing the effective surface exposed to liquid. The rougher the surface, the smaller the bubbles, and the more quickly they are released. Thus, in flooded coolers, heat transfer is greatest for small tubes and those with rough surfaces **(Fig. 1)**.

An increase in mean temperature difference, D, will cause greater turbulence in boiling, sweeping away of the gas bubbles, and improved heat transfer, as shown in Fig. 1. Close spacing of tubes will also increase boiling turbulence. A special type of low extruded fin on a small diameter tube has given very high heat transfer rates; due to the closely nested surface, the turbulence actually becomes foaming and results in high percentage of wetted surface with relatively small amounts of refrigerant. The refrigerant side coefficient is better when:

1. Tubes are in liquid refrigerant contact, or submerged
2. Tubes are small
3. Tubes have roughened or extruded surface to refrigerant
4. Tubes are closely spaced for maximum turbulence of boiling
5. D is high, say 12 F or more, and
6. Oil is absent.

The **resistance** of a clean **tube-wall** to **heat flow** is usually so low as to have little effect on the overall heat transfer. Fouling of the tube-wall surface, particularly on the side of the fluid, can materially decrease transfer. Steel tubes foul more readily than copper. Higher fluid velocities decrease fouling. For small copper tubes, good design practice is to use a fouling factor of 2,000, h_{ws} in Equation (3).

For a given fluid, the **film coefficient on the fluid side** is higher when:

1. Fluid velocity or turbulence is high
2. Viscosity is low and
3. Tubes are small.

Most applications deal with turbulent flow where the Reynolds number is above 3,000 (see Chap. 6) as differentiated from

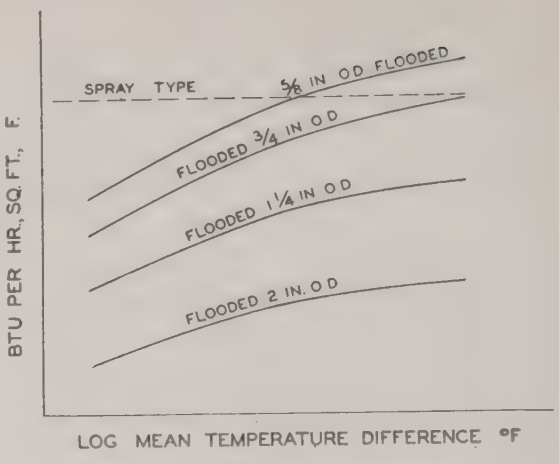

Fig. 1. Heat Transfer Affected by Tube Size and Surface

streamline flow (Reynolds number below 2,500).

The effect of varying the **ratio of outside to inside tube surface** can be seen by referring to Equation (3), where the overall factor U is based on outside surface. If fluid side and refrigerant side coefficients are about equal, bare tubes are normally used. With high fluid velocities up to 8 fps, the fluid side coefficient may become much higher than the refrigerant side coefficient. In this case, it is economical to utilize a finned tube to increase outer tube surface. The finning ratio should be only sufficient to derive maximum benefit from the high fluid side coefficient. For example, in Equation (3), if $h_f = 225$; $L/k = .0000183$; $h_{fs} = 5{,}000$; $h_{ws} = 2{,}000$ and $h_w = 1{,}000$, first take R as 10 to 1, secondly as 4 to 1. For $R = 10$ to 1, $U = 51$. For $R = 4$ to 1, $U = 95$. It will be seen that in the foregoing conditions, the U factor almost doubles when ratio of outside to inside surface is reduced from 10 to 1 to 4 to 1. Therefore, the mention of total surface is no measure of cooler performance unless both quality of surface and ratio of outside to inside surface are known.

Table 1 shows the range of values of overall heat transfer for some of the commonly used types of evaporators. The top limits are for high D values of 12 F or more and for turbulent fluid flow. Fluid velocities for bare tubes range up to 5 fps and for finned tubes up to 8fps.

Table 1. Overall Heat Transfer

Current conservative design values of heat transfer coefficient U for bare tube coolers, unless mentioned otherwise, are as follows:

	Min	Max
Flooded shell-and-tube cooler (water to ammonia or Freon)	50	150
Flooded shell-and-finned tube high velocity Freon water cooler	30	150
Flooded shell-and-tube cooler (brine to ammonia)	45	100
Flooded shell-and-tube cooler (brine to Freon)	30	90
Dry expansion shell-and-tube cooler, Freon in tubes, water in shell	50	115
Baudelot cooler, flooded (ammonia or Freon to water)	100	200
Baudelot cooler, dry expansion (ammonia to water)	60	150
Baudelot cooler, dry expansion (Freon to water)	60	120
Double-pipe cooler (water to ammonia)	50	150
Double-pipe cooler (brine to ammonia)	50	125
Shell-and-coil cooler (water to ammonia)	10	25
Shell-and-coil cooler (water to Freon)	10	25
Spray type shell-and-tube water coolers (ammonia or Freon)	150	250
Tank-and-agitator, coil type water cooler, ammonia, flooded	80	125
Tank-and-agitator, coil type water cooler, Freon flooded	60	100
Tank, ammonia, brine cooling, coils between can in ice tank	15	40
Tank, high velocity raceway type, brine to ammonia	80	110

Actual **design values** of heat transfer, accepted by the industry as a basis for guaranteeing brine coolers, are given in **Figs. 2** and **3.** These charts clearly indicate the effect on heat transfer of tube size, mean temperature difference D, brine velocity, and brine viscosity with lower brine temperatures and heavier brine densities. Figs. 2 and 3 are based on a brine concentration which will freeze at a temperature approximately 20 deg F below the average brine temperature. These curves may be used conservatively for Freon-12 by applying a correction factor of 0.75 to the ammonia performance, based on $1\frac{1}{4}$-in. or smaller tubes for Freon.

Static head correction. In rating flooded brine coolers, the effect of static head is often neglected. The pressure at the bottom of the shell may be a psi or more greater than at the top, with a resultant higher refrigerant boiling temperature and less capacity. For low mean temperature difference and low evaporating temperatures in the larger size coolers, the capacity penalty is appreciable. In **Table 2** is given the liquid static head correction in degrees F to be applied to log MTD in Figs. 2 and 3.

Table 3, for flooded bare copper tube Freon-12 water coolers, gives one manufacturer's performance guarantee. Experience has shown it to be conservative for both large and small tonnage applications and it is representative of normal practice for bare tubes. Heat transfer rates up to 200 Btu per sq ft/hr deg F could be realized with higher water velocities and greater tube submergence.

Flooded Shell-and-Tube Ammonia Brine Cooler

This type of cooler, **Fig. 4,** has long been the standard for industrial brine cooling applications. The construction is all-steel with either $1\frac{1}{4}$-in. or 2-in. tubes expanded into the tube sheets. Typical specifications are given in **Table 4.**

Tube heads are arranged for multipass brine flow through the tubes. The perform-

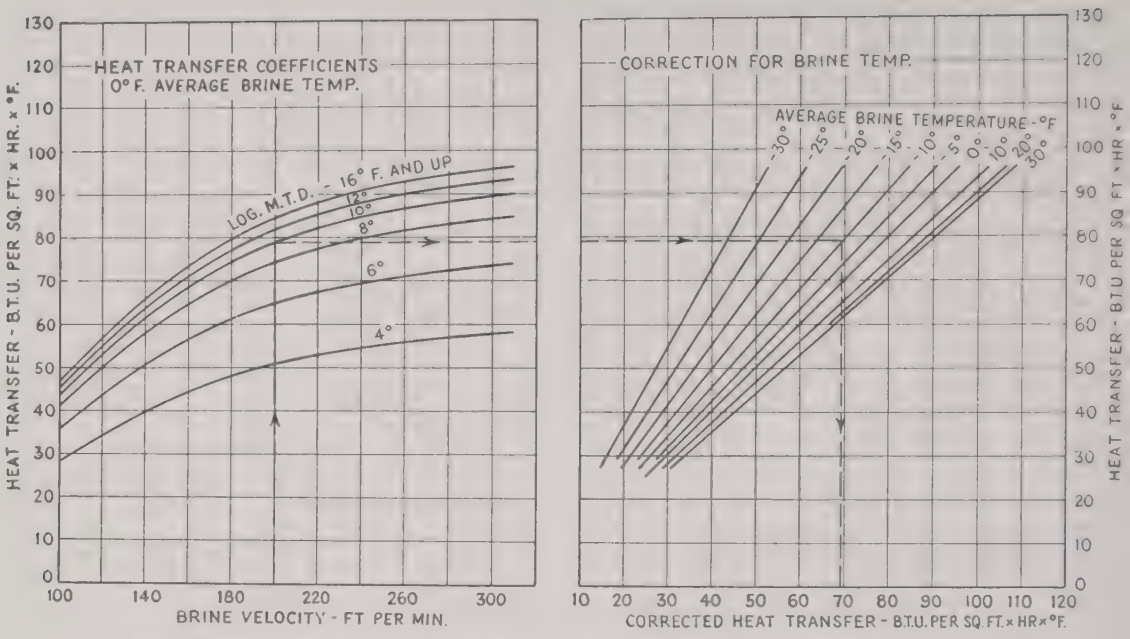

Fig. 2. Heat Transfer from Ammonia to Brine in Flooded Coolers—2-in. tubes

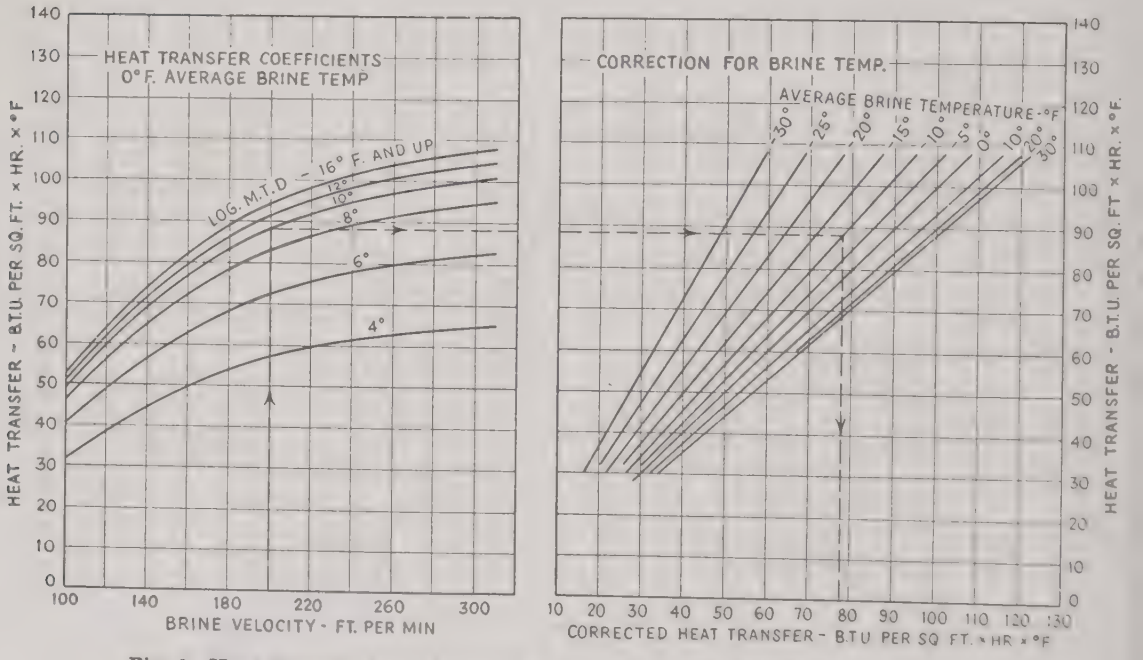

Fig. 3. Heat Transfer from Ammonia to Brine in Flooded Coolers—$1\frac{1}{4}$-in. tubes

Example: For average field operating conditions with a flooded cooler having $1\frac{1}{4}$-in. plain steel tubes, find the heat transfer coefficient when −10 F average brine is circulated at a velocity of 200 fpm and the *D* is 10 F. Following the dashed line and arrow, proceed vertically up from the brine velocity of 200 fpm to the transfer curve for 10 F *D*, go horizontally to the right to the correction curve for −10 F average brine, and, reading vertically down, obtain the corrected heat transfer coefficient of 77.7 Btu per sq ft, hr, deg F.

For identical conditions, except with 2-in. tubes, the transfer coefficient is 69.5 Btu per sq ft, hr, deg F.

Table 2. Liquid Static Head Correction in deg F for Various Evaporator Temperatures, Horizontal Flooded-Type Ammonia Brine Coolers

Shell diam, in.	10 in. vac −42.1 F	5 in. vac −34.5 F	0 −28 F	5 psi −17.2 F	10 psi −8.4 F	15 psi −1.0 F	20 psi 5.5 F	25 psi 11.3 F	30 psi 16.6 F	40 psi 25.8 F	45 psi 30.0 F
16	0.2	0.2	0.2	0.2	0.2	0.2	0.2	0.15	0.15	0.15	0.15
20	0.4	0.4	0.3	0.3	0.3	0.3	0.3	0.2	0.2	0.2	0.2
24	0.7	0.7	0.5	0.4	0.4	0.4	0.35	0.25	0.25	0.2	0.2
30	0.9	0.9	0.7	0.5	0.5	0.5	0.4	0.3	0.3	0.25	0.25
34	1.1	1.0	0.8	0.6	0.6	0.55	0.45	0.35	0.35	0.3	0.3
38	1.3	1.1	0.9	0.7	0.7	0.6	0.5	0.4	0.4	0.3	0.3
42	1.4	1.2	1.0	0.75	0.75	0.65	0.55	0.45	0.45	0.35	0.35
45	1.5	1.3	1.0	0.8	0.8	0.65	0.55	0.45	0.45	0.35	0.35
50	1.6	1.4	1.1	0.9	0.9	0.7	0.6	0.5	0.5	0.4	0.4
54	1.7	1.5	1.2	1.0	1.0	0.75	0.65	0.55	0.55	0.4	0.4
60	1.9	1.7	1.4	1.1	1.1	0.85	0.75	0.6	0.6	0.5	0.5

ance of this cooler, Figs. 2 and 3, is based on use of calcium chloride brine with the shell flooded with liquid ammonia to a static height of 0.8 of the internal shell diameter. Brine temperature differences, entering minus leaving, in practice run from about 4 to 15 deg F. **Table 5** compares flow of water to $CaCl_2$ brine of 1.2 sp gr for various temperature differences.

Consley[5] has discussed the use of swirl strips in the tubes to increase brine turbulence at low velocity. Although the gain in heat transfer is somewhat balanced by increased pressure drop, swirls may be used to increase capacity of existing equipment in the field, as in **Fig. 5.**

Flooded Shell-and-Tube Freon Coolers

This cooler has found its greatest field of application in chilling water for air conditioning service. Recent installations have been made, however, in large tonnage capacities for industrial brine cooling. Usr ual construction is steel shell and coppe-tubes. Latest trend is to the use of finned tubes and higher fluid velocities.

For $CaCl_2$ brine cooling, construction is generally of steel throughout, although in plants where brine is chemically treated and kept under supervision, non-ferrous alloys are coming into use. Usual practice is to groove or serrate the tube holes in the tube sheet, of either steel or non-ferous alloy, then expand the tube to a tight fit in the tube hole. Some coolers are made with light copper or alloy tube sheets with tubes silver soldered to the tube sheet.

A typical design of bare tube cooler is shown in **Fig. 6** and its hook-up with float control in **Fig. 7.**

In this cooler the suction is taken off the top at several points into a main header containing superheater surface with the compressor connection in the center. This type of design uses a somewhat smaller shell, the shell itself being almost filled with tubes on close spacing. The need for distributing headers on the liquid side is mainly to cause uniform turbulence through the length of the shell. For typical specifications of this type cooler, see **Table 6.**

A typical design of finned tube cooler is shown in **Fig. 8** and its hook-up with pilot thermal valve control in **Fig. 9.** Here the super-heater surface is located within the shell. The close nesting of finned tubes

Table 3. Overall Heat Transfer Factors in Flooded Coolers

(Btu per sq ft hr deg F)

Water velocity, fpm	Mean temperature diff, deg F				
	6	8	10	12	15
150	67	76	83	90	97
200	83	95	103	110	118
250	97	109	115	122	529
300	103	115	123	130	138

These factors are based upon the actual temperature of refrigerant in the shell, and a Freon level of approximately ⅔ of the total height, with commercially pure, oil-free Freon and ¾-in O.D. copper tubes.

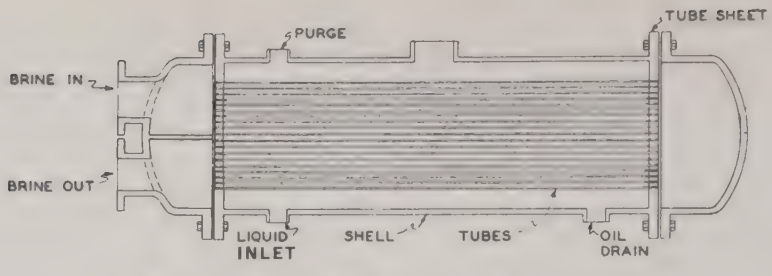

Fig. 4. Ammonia Shell-and-Tube Brine Cooler

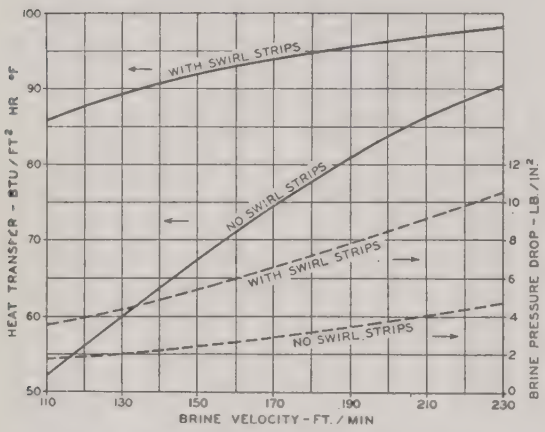

Fig. 5. Effect of Swirl Strips—Ammonia Brine Coolers

concentrates heat transfer surface to produce violent turbulence and foaming, which effect uniform wetting of the entire surface. Thermal valve action causes some carry-over of foam containing oil. This foam is dried out by the superheater and the oil returned to the compressor via the suction line. Specifications for this cooler are as in **Table 7.** The ratio of outside to inside surface is approximately 4 to 1.

Spray-Type Coolers

This type is being largely superseded by flooded coolers except for applications at very low temperatures where the static head penalty on flooded coolers is too great or at low MTD's where the performance of flooded coolers falls off rapidly. Since the wetting of the tube surface by sprays is uniform regardless of heat load, heat transfer of spray-type coolers is independent of MTD.

Construction is similar to flooded coolers except for location of a spray header with nozzles above the tube bundle. The Freon or ammonia is collected in a sump at the bottom of the cooler and recirculated by a low-head pump through the spray system, flowing down over the tubes. Recirculating flow ratio is high compared to evaporated refrigerant flow, assuring thorough wetting of tubes. Refrigerant charge is low and is much less affected by load variations than the flooded coolers. Greatest disadvantage of spray coolers is necessity of mechanical pump and high installed cost.

Table 4. Specifications of Typical Ammonia Flooded Brine Coolers with 1¼-in. Tubes

Shell diam, in.	Length, ft	No. tubes	Surface area, sq ft	Weight, lb	Ammonia charge, lb
16	16	60	315	2,700	420
20	16	103	538	4,400	642
24	16	163	853	6,500	868
30	16	247	1,292	9,200	1,427
34	16	341	1,784	12,200	1,880
38	16	426	2,229	15,400	2,320
42	16	530	2,773	18,500	2,795
45	16	612	3,202	21,500	3,140
50	16	764	3,997	26,500	3,830
54	16	868	4,541	30,200	4,340
60	16	1,082	5,661	37,700	5,450

Table 5. Rate of Flow of Fluid at Differences in Temperature to Equal One Ton of Refrigeration

Temperature change, deg F	Water circulated, gpm/ton	$CaCl_2$ circulated, gpm/ton
1	24	28.6
2	12	14.3
4	6	7.15
6	4	4.8
8	3	3.6
10	2.4	2.86
12	2	2.4
14	1.72	2.04
16	1.5	1.8
18	1.33	1.6
20	1.2	1.43

Non-Flooded Shell-and-Tube Freon Coolers

The so-called dry expansion cooler (Fig. 10) in which the refrigerant is in the tube, is becoming more popular for small and medium tonnage applications, say up to 150 tons per unit. Thermal valve is used for Freon control. Since the valve requires superheat for its operation, this cooler is not good for applications requiring close temperature splits between Freon and fluid, unless used in conjunction with heat exchanger. There are two main types of design, first the U-tube type, in which the entire tube bundle is removable from the shell and, second, the fixed-tube type which is like an ordinary cooler except that

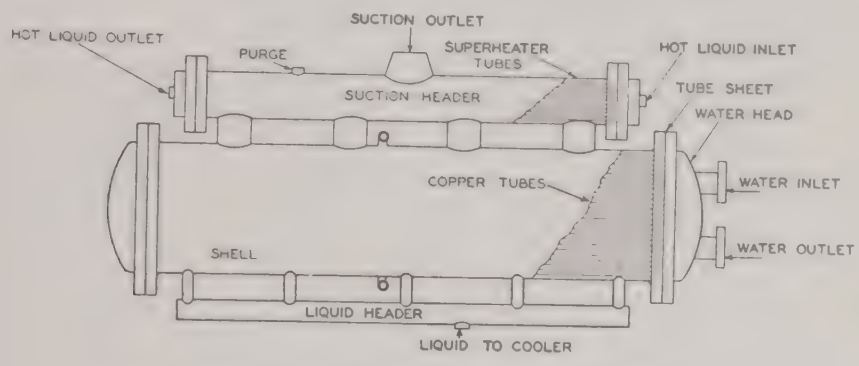

Fig. 6. Typical Freon Flooded Water Cooler

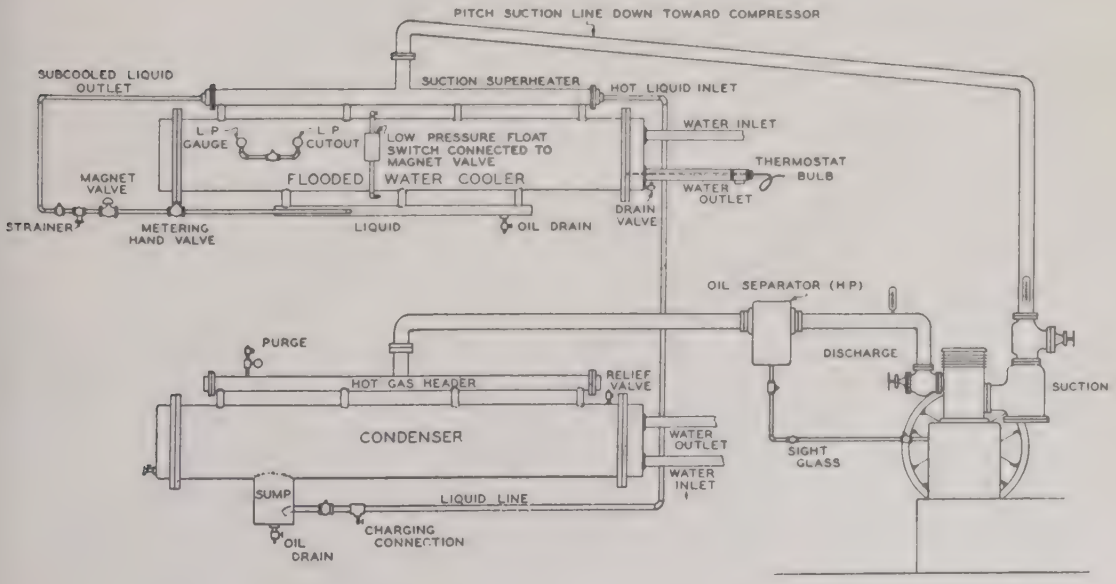

Fig. 7. Recommended Hook-up of Freon Water Cooler

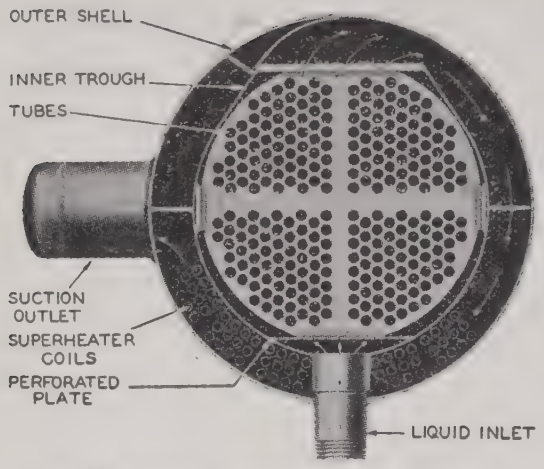

Fig. 8. Finned Tube Freon Water Cooler

it has shell side baffles. The U-tube type is shown.

The advantages of this cooler are that the water friction is low, the control is simple with the thermal expansion valve, Freon charge is small, and there is no difficulty with oil return since the feed is positive. This cooler does not readily freeze or burst tubes, and for this reason, lower Freon temperatures can be used. The overall heat transfer is lower than with flooded coolers, but generally the actual installed cost of the dry cooler is about the same as the flooded. The figures which follow indicate design factors used by one manufacturer, based on a constant water velocity:

Table 6. Specifications of Typical Freon Flooded Water Coolers with Bare Copper Tubes, $\frac{5}{8}$-in. O.D.

Shell diam, in.	Length, ft	No. tubes	Surface area, sq ft	Weight, lb	Normal capacity, tons 10° MTD and 5 fps vel	Freon charge, lb
$10\frac{3}{4}$	5	91	74	500	7.4	75
	10	91	148	950	14.8	150
$12\frac{3}{4}$	10	133	216	1,150	21.6	200
	16	133	345	1,840	34.5	320
14	14	166	380	1,600	38.0	345
16	14	220	504	2,200	50.4	445
18	16	306	800	3,360	80.0	620
20	16	364	951	4,240	95.1	800
24	16	566	1,480	6,000	148.0	1,080

Table 7. Specifications of Typical Freon Flooded Water Coolers with Finned Copper Tubes, $\frac{5}{8}$-in., 16 Fins per in.

Steel diam, in.	Length, ft	No. tubes	Surface area, sq ft	Weight, lb	Normal capacity, tons 10° MTD 8 fps vel	Freon charge, lb
16	12	52	320	1,530	32	236
20	12	108	665	2,225	66	520
24	12	200	1,235	3.210	124	919
30	12	338	2,080	4,660	208	1,590
36	12	564	3,475	7,215	348	2,620
42	12	764	4,700	9,940	470	3,600
50	12	1,250	7,700	14,315	770	5,900

D, deg F	*U Btu per sq ft hr deg F*
8	80
10	105
12	110
15	115

Other manufacturers have based their ratings on varying water velocities rather than mean temperature difference.

The straight tube design, while having the disadvantage of lack of access to the water side, has the advantage that it may be re-tubed if necessary. Re-tubing, however, is seldom necessary with Freon in the tubes, since there is little likelihood of rupture due to freezing. From the practical standpoint, the removability of the U-tube bundle is not too great an advantage.

Shell-and-Coil Coolers

The shell-and-coil cooler has its application chiefly in drinking water systems, for bakeries and photographic laboratories. It consists of a steel or alloy tank with a circular coil in which the refrigerant is circulated. The water or brine is in the tank outside the coil, as in **Fig. 11.** This is generally a dry expansion application, for the reason that the resistance to heat transfer is so much greater on the water side than on the Freon side, that there is little point to going to further complication than the simple thermal expansion valve. This type of cooler has the advantage of flywheel effect, making it popular for applications where peak loads are high and infrequent.

There are a number of specialty applications of small water coolers for the same purpose as the shell-and-coil, in two of

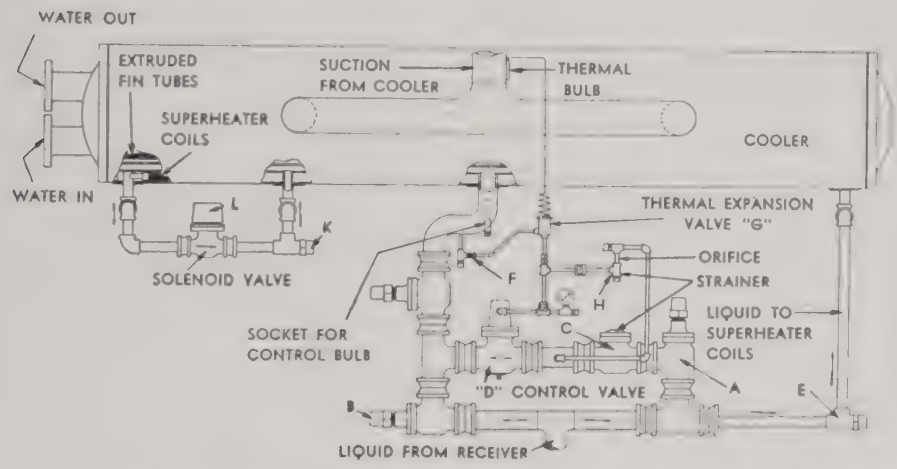

Fig. 9. Typical Thermal Valve Hook-up of Freon Water Cooler

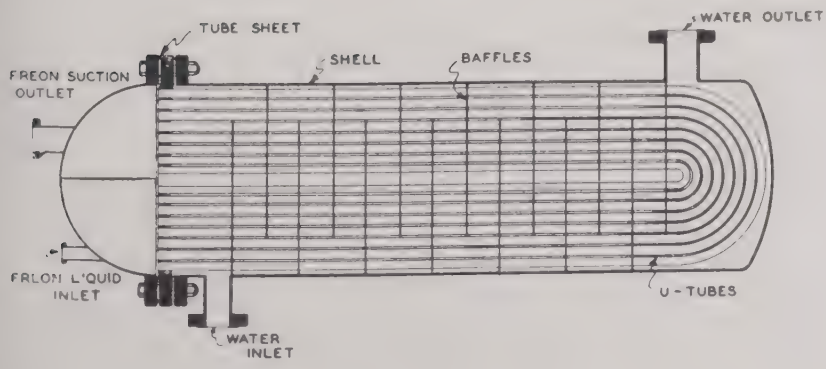

Fig 10. Dry Expansion Freon Water Cooler

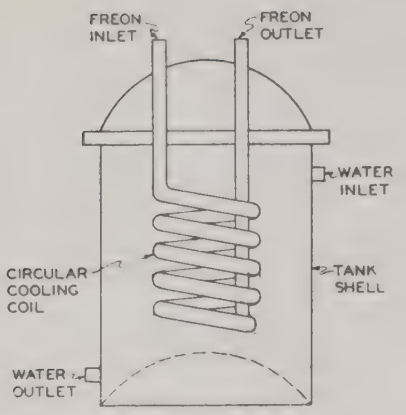

Fig. 11. Shell-and-Coil Cooler

which an attempt is made to obtain the efficiency of a high-velocity evaporator combined with the holdover effect of a tank cooler. This has the disadvantage that once the incoming water has passed the high-velocity evaporator, it must have been cooled, since it is not recirculated. The arrangement does not allow the continuous cooling at low average rate as in the simple pressure tank or shell-and-coil cooler. Also, the high-velocity cooler can freeze and cause damage to the coil, whereas this is very unlikely with the shell-and-coil type. Due to icing, these coolers should not be used where it is desired to cool water below about 38 F.

The Baudelot Cooler

There are many industrial applications requiring water chilled to 35 F, sometimes as low as 33 F. Shell-and-tube or shell-and-coil coolers are not suitable for this service as they will ice up. The Baudelot cooler of **Fig. 12** lends itself very well to producing near-freezing temperatures. Although it, too, can ice up, no damage can be done, and since it is open to visual inspection, ice build-up can be controlled. It consists of a series of pipes one under the other, through which refrigerant circulates and over which the water to be cooled trickles down in a thin film from an overhead pan in which small distributing holes are drilled. A more modern design comprises stamped corrugated stainless steel sheets in which the corrugations form channels for the refrigerant flow. The sheets are highly polished and form a sanitary, easily cleaned surface. Also, they provide a continuous surface offering better control of water distribution.

Ammonia type is usually flooded, using the conventional gravity feed system with surge drum and low pressure float. Freon type is usually either dry expansion or combination dry and flooded.

The double-pipe cooler is not used very widely in modern practice for other than a few specific applications. It is used in the

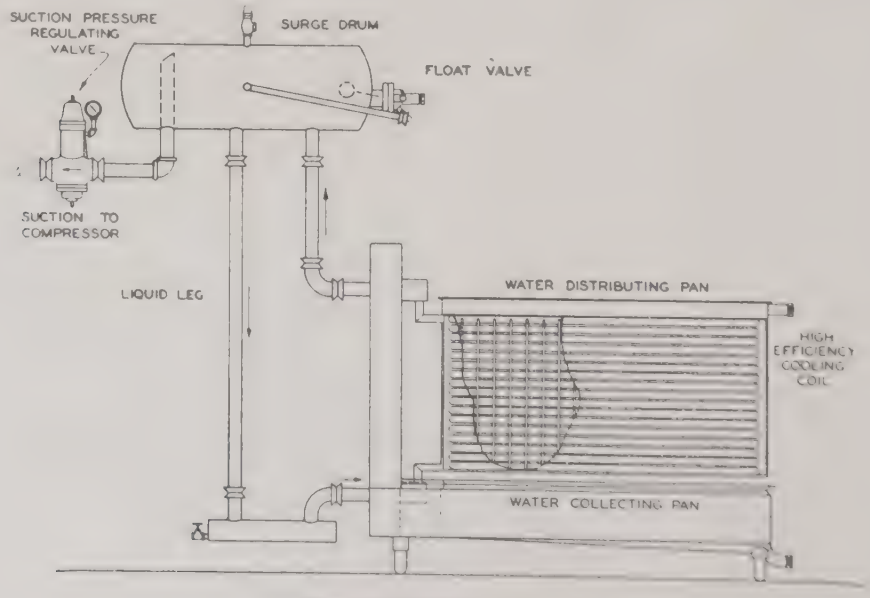

Fig. 12. Baudelot Cooler

cooling of wine and beer during manufacture and in the oil refining industry for oil chilling. The cooler is efficient, being adaptable to counterflow, but takes up more space than the shell-and-tube type, has more refrigerant side friction and is not readily cleanable. It is virtually obsolete for cooling of water or brine.

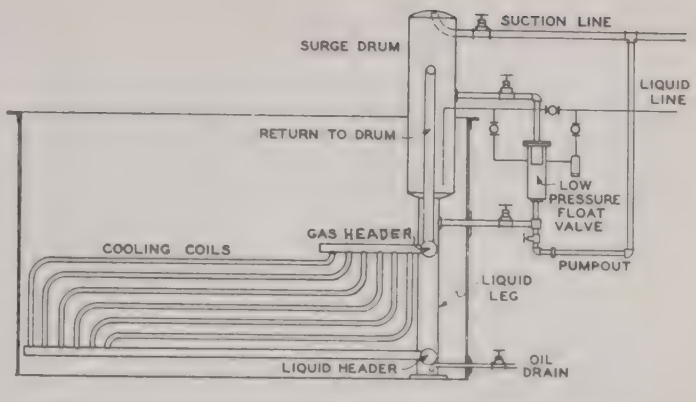

Fig. 13. Typical Boiler Type Coil with Flooded Control

Tank-Type Cooling Systems

The tank or boiler-type coil system (**Fig. 13**) for cooling brine or water has been in almost universal use for large systems for many years, where sanitation is secondary. A large steel tank contains the fluid to be cooled. At one side or the center of the tank is placed a closely nested set of pipe coils, baffled off from the tank proper and fitted with an agitator to circulate the brine or water in the tank over the coils at a relatively high velocity (100 to 150 fpm).

The agitator circulates large quantities of brine or water to maintain uniform temperatures throughout the tank with total temperature difference less than 1 deg F. Coils are usually arranged with short length of travel from liquid to suction header to minimize refrigerant pressure drop. In an alternate arrangement to the one shown, the coils are bent in V-form between the headers.

This same type of system is used for freezing ice in can-type tanks. The tank is enlarged to accommodate the ice cans with brine flowing past the cans at approximately 25 fpm.

Typical performance of these coils is given in **Table 8.** For flooded ammonia operation, correction must be made for static height of liquid from liquid level in surge drum to liquid header.

Some ice-making tanks still use flat coils between rows of cans, usually controlled by hand expansion. Heat transfer values for these coils are given in Table 1.

Brines

Calcium chloride brine is used for most industrial refrigeration jobs. Sodium chloride is used in open spray systems where contact of vapor with food is involved. Its heat transfer characteristics are very similar to calcium chloride. Calcium chloride is seldom used below −30 F because of poor heat transfer due to its increasing viscosity. For very low temperatures, methanol and chlorohydrocarbon liquids can be used. Two of these which have been used in recent years to temperatures as low as −100 F are methylene chloride and trichlorethylene. Some of their properties are shown in **Table 9.**

Oil and Heat Exchangers

Since the presence of oil in evaporators, although unavoidable, interferes with performance, means should be provided to reduce the quantity as much as possible. In ammonia coolers the oil, being heavier than the refrigerant, usually settles out at low spots from which it can be drained.

With Freon-12, oil raises the boiling point one degree for each 5% of oil present. Also, oil is miscible with liquid Freon-

Table 8. Typical Performance of Boiler Type Coils (High Velocity)

(Overall heat transfer, Btu per sq ft hr deg F)

Velocity, fpm	Water	Brine			
		30 F	15 F	0 F	−15 F
50	68	52	47	42	38
75	88	72	66	60	53
100	110	94	85	77	68
125	132	116	105	95	84
150	150	133	121	109	97

Table 9. Properties of Chlorohydrocarbon Liquids

	Methylene chloride	Trichlor-ethylene
Freezing pt, F	−142.1	−126.4
Sp ht at −22 F, Btu/lb/F	0.275	0.222
Sp gr at −22 F	1.42	1.55
Viscosity, centipoises at −4 F	.676	.90

12 and cannot be drained to atmosphere. Various recovery devices are available to "still" some of the oil and return it to the compressor, thus maintaining a tolerable oil concentration (not more than 10%) in the cooler. For float-controlled flooded coolers, the oil still is commonly used. It is a form of double-pipe heat exchanger using liquid to the cooler as a source of heat. A sample of liquid in the cooler is passed through the still, the liquid evaporated and the remaining oil returned to the compressor.

Suction-line heat exchangers are used with Freon-12 evaporators for a two-fold purpose: (a) to assist in oil return, and (b) to serve as liquid precoolers. As oil return devices, they apply particularly to thermal expansion valve fed evaporators, where the superheat required to operate the expansion valve is produced in the heat exchanger with warm liquid as the source of heat. This action forces some liquid-oil mixture spill-over from the evaporator. The exchanger evaporates the liquid and permits the oil to return to the compressor via the suction line. Used as liquid precoolers, heat exchangers can show some gain in overall system capacity. They have been thoroughly discussed by Crampton,[4] Newcum[2] and St. Onge.[3] The theoretical gain is about 0.5% per degree subcooling of warm liquid. Great care should be taken in their application to avoid suction-line pressure drops which can overbalance the possible gain.

The **gravity feed system** has long been employed to control refrigerant flow through evaporators, particularly those comprising some form of pipe coil, although it has also been applied to shell-and-tube type apparatus. Its main function is to insure thorough flooding of the evaporator, at the same time protecting against liquid carry-over to the compressor. It consists of a surge drum elevated above the evaporator, a liquid feed leg connecting the bottom of the surge to the bottom of the evaporator, and a return line from the top of evaporator to surge drum, as shown in Fig. 12. In principle, it is based on the fact that the return leg, including the evaporator, being a mixture of liquid and gas, is lighter in weight than the liquid feed leg containing solid liquid. The effect is not only to flood the evaporator, but actually to recirculate liquid refrigerant, the rate of recirculation being dependent on the difference in weight of the two columns, a function of the heat per sq ft of evaporator surface. The surge drum serves as a separating chamber to return liquid to the evaporator and insure dry gas to compressor. Liquid level in the surge drum is usually controlled by means of a low-pressure type float expansion valve, although drain-type floats and thermal valves are sometimes used.

Liquid and return lines and pipe coil circuits should be sized for very low friction, as the success of the system depends on circulation by means of gravity flow only.

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4. Crampton, David, *Refrig. Eng.*, Dec. 1940
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26. LOW TEMPERATURE SYSTEMS (GAS LIQUEFACTION AND DRY ICE)

I. Air Products

1. Low temperature refrigeration systems had their **laboratory development** in the liquefaction of the so-called "permanent gases" such as air, oxygen, nitrogen, and later hydrogen and helium. The first commercial development of low temperature refrigeration was effected by the attempt to extract pure oxygen from air by first liquefying air and then recovering the oxygen in a relatively pure state by fractional distillation. After the principle of separating a gas mixture by liquefaction and distillation was successfully applied to air, the technique was extended to the recovery of helium from natural gas, and hydrogen and ethylene from petroleum cracking still gases, as well as to the separation of the rare gases such as argon, xenon, and krypton from air.

In some of the simple types of gas liquefiers the gas itself acts as the refrigerant. In other types, greater efficiency is obtained by employing separate refrigerants. In all liquefying machines the gas must be under sufficient pressure that it can be allowed to expand to a lower pressure, causing a lowering in temperature by either of the two following methods:

a. Joule-Thomson expansion—adiabatic irreversible expansion of gas at high pressure to a lower pressure with a lowering in temperature. At the end of the expansion process the gas is at a lower temperature and lower pressure, with an entropy increase.
b. Expansion of the gas reversibly or irreversibly in a machine in which it can perform work; the gas undergoes a decrease in temperature, pressure, and enthalpy.

For a particular gas at a chosen initial temperature and pressure expanding to a chosen final pressure, the expansion (with work produced) will result in a greater lowering of temperature than the adiabatic Joule-Thomson expansion. However, only in recent years has the expansion principle been used extensively.

In all liquefying machines the gas cannot expand from a higher pressure to a lower pressure and have a sufficient reduction in temperature for liquefaction to occur, but the gas must first be cooled to a sufficiently low temperature that the final expansion is below the critical temperature. For the cooling of the gas before expansion another essential part of the liquefying machine, aside from the compressor and expansion valve, is a heat interchanger. Combination and modification of these three essential parts are used in various ways, as further described, in refrigeration systems.

Linde or Hampson System

2. This is the simplest type of liquefaction system, consisting of the three minimum parts, i.e., compressor, heat interchanger and expansion valve. **Fig. 1** is a schematic flow diagram of the liquefier.

The air leaving the compressor A is under a pressure of 100 to 200 atmospheres, cooled to about −120 C after leaving the heat interchanger B, and expanded adiabatically to atmospheric pressure with a further drop in temperature to the boiling point. If the gas is compressed to 200 atmospheres, about 10% of the gas leaving the expansion valve C liquefies; the remaining unliquefied

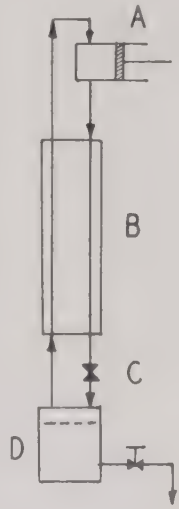

Fig. 1. Linde System

air passes through the cold side of the heat interchanger, cooling the high pressure gas from the compressor, after leaving the interchanger at a temperature about 2 or 3 degrees above room temperature. The quantity of gas that liquefies after an expansion is a function of pressure and the thermal properties of the gas.

This type of liquefier, because of its simple construction, is often used where small quantities of liquefied gas are needed. However, if a more efficient machine is needed, this system can be modified and is designated as the improved Linde system.

Improved Linde System

3. In order to improve the efficiency of the Linde liquefiers a two-stage expansion and compression system was devised. **Fig. 2** indicates the arrangement of this system. The air is compressed by the first-stage compressor A to 40 atmospheres; the second-stage compressor B further compresses the gas to 200 atmospheres. The compressed gas then passes through the inter-coolers C and expansion valve D, where the first adiabatic expansion reduces the pressure to 40 atmospheres into chamber E. Here 20% of the gas is further expanded to 1 atmosphere as it passes through expansion valve F into chamber G. For the final expansion about 8% of the original gas is liquefied.

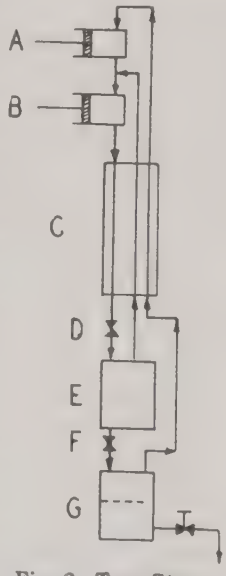

Fig. 2. Two-Stage Linde System

No liquid will result from the first expansion unless the temperature in chamber E is below the critical pressure. For air this would have to be below −222 F. The optimum pressure for minimum power consumption per unit quantity of liquefied air produced is the same as the pressure given above, namely 40 atmospheres.

From chamber E about 20% of the air under 40 atmospheres is expanded on further to chamber G, while the remaining 80% is returned to the compressor for recompression.

The expanded air from chamber F and part from chamber D pass counter-current to the incoming compressed air in intercooler C and then on to their respective compressor intakes.

Economy in power expenditure is achieved in this scheme because a major portion of the air (about 80%) is compressed from 40 to 200 atmospheres instead of from 1 to 200 atmospheres. Since the energy required to compress a gas varies as the logarithm of the ratio of the pressures, the work of compression from 40 to 200 atmospheres is about $\frac{1}{6}$ the work of compression from 1 to 200 atmospheres. Further power economy can be achieved by precooling the compressed air by an auxiliary refrigeration cycle.

Claude System

4. As previously mentioned, the expansion of a gas under reversible conditions produces a cooling effect which is greater than an adiabatic irreversible expansion from the same initial condition to the same final pressure. The application of the above stated principle alone in the expansion of a gas in a cylinder to liquid has been found to be impractical at low temperatures, because of the low efficiency of expansion and difficulty of lubricating the cylinder.

The Claude system is a combination of both principles of cooling, i.e., the Joule-Thomson effect and the approximate reversible expansion of the gas in an expansion engine.

The Claude liquefier is schematically given in **Fig. 3.** The air is compressed to about 40 atmospheres by compressor A; the compressed air passes through the interchanger B and separates into two parts. One stream, about 20%, passes through the interchangers C and D and is finally expanded in valve F to container E at atmospheric pressure, where the liquid and vapor portions separate. The unliquefied part from the expansion returns to the intake of the compressor after passing through the three intercoolers D, C, and B. The remaining 80% of the stream that separates from the main stream be-

tween interchangers B and C is expanded in cylinder G of the expansion engine, and joins the returning low-pressure air from the interchanger D on its way back to the intake of the compressor.

The theoretical power consumption for a given quantity of liquid produced is a function of the exhaust pressure of the compressor A, the fraction of the air that passes through the expansion valve and the inlet temperature of the air to the expansion engine. The conditions given above with an inlet temperature of about −80 C are optimum for power consumption.

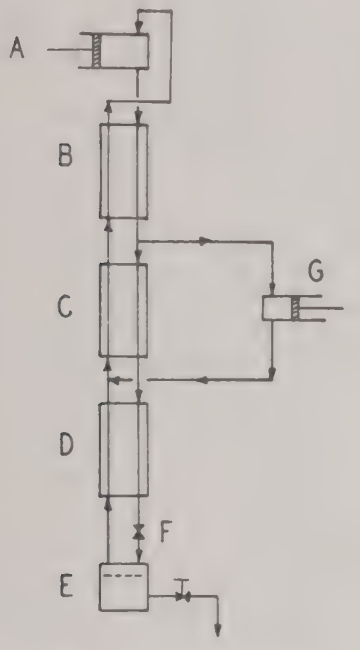

Fig. 3. Claude Liquefier

The difficulty of lubricating the cylinder of the expansion engine can be overcome to some extent by using leather or fiber piston rings, and according to Russian practice[6] the engines have been successfully replaced by turbines in which the lubricating problem at low temperatures does not arise as the turbine bearings are at relatively warm temperatures at which lubricating oil can be used.

The power generated by the expansion engine is sometimes used to "help" the compressor through a belt connection. In other cases, the expansion engine may run a dynamo.

Heyland System

5. If the compressor discharge on the Claude system is raised to 200 atmospheres the optimum inlet temperature for the expansion engine will be at approximately room temperature and the first intercooler, as shown in Fig. 3, is not needed. The conditions just described are requisite to the Heyland system. Such conditions are of a practical advantage in that the difficulties of lubricating the cylinder of the expansion engine are lessened. Light lubricants have been found satisfactory.

In the Heyland system minimum power consumption for the 200 atmosphere compressor discharge is achieved by passing 55% of the air through the expansion engine and the remaining 45% through the valve. Thus this liquefier differs from the Claude liquefier in which 80% of the air is expanded in the engine.

Cascade System

6. The cascade system of refrigeration employs a series of gases of progressive lower boiling points. Each gas is used as a cooling medium to condense under pressure the gas of the next lower boiling point. This system was first proposed and employed by Pictet[8] in 1877 to liquefy oxygen. The intermediate gases chosen were sulfur dioxide and carbon dioxide. Kammerlingh-Onnes[4] developed a cascade system to liquefy oxygen and nitrogen, using as intermediate gases methyl chloride and ethylene.

In the cascade system proposed by Keesom[5] for liquefying nitrogen the following gases were employed in this respective order: ammonia, ethylene, methane, and nitrogen. This system is indicated in **Fig. 4.** The ammonia is compressed to 10 atmospheres by compressor A_1, liquefied at room temperature and expanded in valve D_1 to 1 atmosphere. The ammonia that vaporizes from evaporator C_1 passes through the interchanger B_1, cooling the compressed ethylene from compressor A_2. The ethylene under a pressure of 19 atmospheres is liquefied in the submerged coil of evaporator C_1. The liquefied ethylene expands in

valve D_2 to atmospheric in evaporator C_2. The vaporized ethylene from C_2 passes through the interchanger B_2, cooling the methane from compressor A_3.

The methane is liquefied under a pressure of 25 atmospheres in the submerged coil of evaporator C_2, expanded in valve D_3 to 1 atmosphere in evaporator C_3. The vaporized methane from the evaporator C_3 is divided into two streams, passing through the interchangers B_2 and B_3, cooling the compressed methane and compressed nitrogen streams respectively. The nitrogen at a pressure of 19 atmospheres from compressor A_4 liquefies in the submerged coil of evaporator C_3. The liquefied nitrogen is expanded in valve D_4 to 1 atmosphere. The liquefied portion from the expansion may be withdrawn from accumulator E while the vaporized portion returns to the compressor after first passing through B_3.

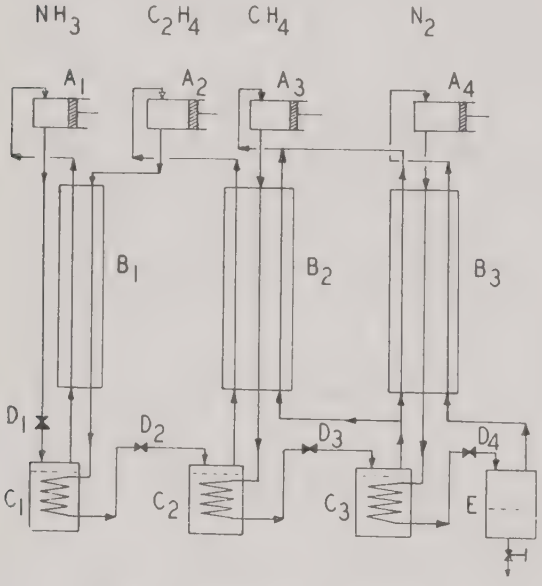

Fig. 4. Liquefaction of Nitrogen

The cascade system of obtaining low temperature is unique in that the power consumption per unit of gas liquefied is less than that of the other processes described.

7. In order to make a comparison of the efficiency of the various processes Keesom[5] made a detailed analysis of the power requirements for the production of 1 kilogram of liquefied nitrogen under 1 atmosphere, starting with gaseous nitrogen at 25 C and 1 atmosphere. **Table 1** is a tabulation of the result of this analysis on the power requirements for the cascade, Linde, and Claude processes. The theoretical power requirement for a hypothetical process operating under perfect reversible conditions is also listed in Table 1.

The reversible process requires the minimum amount of work and is independent of any mechanism. Actual processes have greater power requirements because there are energy losses inherent in any practical process as distinguished from a reversible process. The reversible process cannot actually be achieved, but it is set up as a standard for purposes of comparison.

In Keesom's analysis of the three processes the nature and amount of the energy losses are given. Aside from energy loss due to friction in the moving parts of the compressor and the fluid flowing through the system, there is the irreversible effect of heat transfer in the interchangers, evaporators, and heat leakage into the apparatus from the surroundings. The greatest losses occur in the expansion of the gas or liquid in the expansion valves. The nearer the compressed gas is to saturation conditions the less irreversible the expansion (indicated as less increase in entropy). Moreover, the expansion of the liquid in the valve has less irreversible effect than the gas.

Table 1. Theoretical Power Required, hp-hr per kg of Liquid Nitrogen

Hypothetical reversible process	0.29
Cascade process	0.73
Linde process	1.41
Claude process	1.19

The cascade system is more economical in power requirement because only liquids are expanded. If the compressed gas in the other systems is precooled with an external refrigeration system there will be an improvement in the overall power economy.

Commercial Liquefaction of Natural Gas

8. A recent commercial application of the cascade system was the liquefaction

of natural gas for storage purposes. A plant erected at Cleveland, Ohio, was built for the purpose of liquefying the natural gas at times when the consumption was low, storing and re-gassifying it at peak load conditions when the pipe line from the gas field would not be able to furnish the necessary quantity.

The liquefaction of natural gas reduces the volume of required storage space 600 times. This plant was designed to liquefy 4,000,000 cu ft of natural gas per day and had a storage capacity of three spherical tanks each 57 ft in diameter. The liquefied natural gas was stored at a temperature of −250 F and at a pressure of approximately 12 psig. The storage spheres were made of nickel steel rested in cork insulation inside of a steel sphere 63 ft in diameter. From experimental work the heat absorption was estimated to be relatively small, and if the storage spheres were full, at the end of a year's time only half of the liquid would be evaporated.

Before liquefaction, the natural gas is processed to remove CO_2 and water vapor in order to prevent plugging of the lines, and nitrogen is vented off at intervals to prevent too great an accumulation in the recycle system. This plant was unfortunately destroyed by an explosion and fire.

Detailed descriptions of this plant have appeared in various publications.[1,2,3,7] In general, the refrigeration cycles used are the same as proposed by Keesom, with the exception of the nitrogen cycle, namely, ammonia, ethylene, and methane.

II. Dry Ice

9. Carbon dioxide ice or dry ice is useful as a refrigerant agent in many applications, especially in transportation. Its high cost of manufacture limits its field of use, but it has extraordinary merit in being readily handled and cut dry—sublimating to gas, non-corrosive and non-toxic, high as to latent heat and low in temperature. In the gaseous form it is a good insulator.

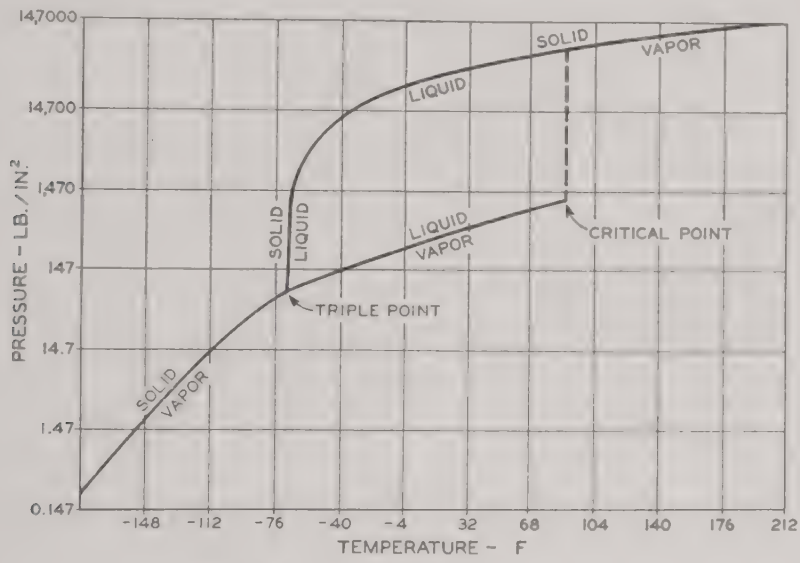

Fig. 5. Pressure Temperature of CO_2

Manufacture of dry ice divides into the process of making or recovering the gas, a chemical process, and its liquefaction and solidification by a compression cycle.

Carbon dioxide has been obtained commercially from natural springs and fissures, from wells, from heating or acid treating the carbonates of alkalies and alkali earths, from fermentation of organic substances, and from the burning of carbonaceous substances.

The refrigerating effect is the sum of the heat of sublimation (latent heat of the solid) or 248, plus the sensible heat derived by raising the temperature to say 32 F, or about 27, giving 275 Btu per lb.

See p. 129 for table of physical properties. Pressure-temperature relations are shown in **Fig. 5** and Chart 4, Chap. 7.

10. Limestone, magnesite and dolomite have all been used and are being used as sources of CO_2 gas, and appear to be the most rapidly growing sources. Two methods are used to separate the CO_2 from the carbonates: (a) by heat, (b) by acid treatment. In the first method, the stone is heated in kilns, preferably of the vertical type. The offcoming gases (containing the CO_2 from the carbonate and that from the combustion of the fuel used to heat the carbonate) are found to be from 25 to 45% CO_2. The yield of CO_2 is approximately 50% of the weight of stone used, e.g., the "ammonia-soda" process.

In the acid method the carbonate is treated with acid and the offcoming gas is practically pure CO_2. The yield is from 44 to 52% the weight of stone, dependent on the kind of carbonate used. This latter method appears to be applicable only where the salts produced from the acid treatment are marketable.

All **organic matter** in **fermenting** generates CO_2. Commercially the organic matter used is molasses and grain. The dry gas is about 99.5% CO_2 and can be recovered in proportions of about 5 lb of CO_2 per gal of alcohol produced. The odor of the impurities in this case makes satisfactory purification a difficult task, one of several methods being as in **Fig. 6.**

Coke ranks next to limestone as a commercial source of carbon dioxide. Usually it is burned under boilers in such a manner as to obtain the maximum concentration of CO_2 in the flue gas. The steam generated is used for generating power and for aiding in the purification process. The coke should be of high carbon content, low in sulfur and of good structural strength; 72-hr foundry coke has been the standard of quality for years. Lately high grade by-product coke has been used satisfactorily.

Theoretically, 1 lb of coke (90%) will generate 3.0 lb of CO_2; actually, the recoverable yields have in the past been nearer a ratio of 1 to 1. Good practice with present equipment should yield a ratio of 2 lb of CO_2 per lb of coke burned. The flue gas, under good firing conditions, will run 15 to 18% CO_2, about 80% nitrogen and small amounts of oxygen, carbon monoxide, sulfur dioxide and hydrocarbon gases.

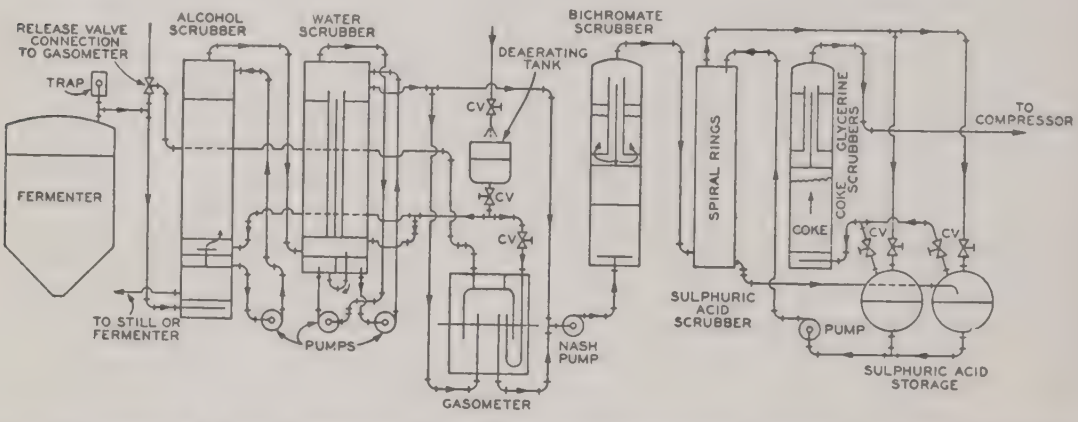

Fig. 6. Diagram of Reich Process of Carbon Dioxide Gas Production, Showing Purifier Apparatus

About four tons of CO_2 are generated in making one ton of steel. Most of it is generated in the **blast furnace** from the limestone and coke used. The gases as they leave the blast furnace itself run high in carbon monoxide, but on being burned in the stoves or in the blowing engines, the composition is about 20 to 30% CO_2, 70 to 80% nitrogen, with small amounts of carbon monoxide, oxygen and sulfur dioxide.

A vast potential source of CO_2 is found in the **petroleum industry** as a by-product in hydrogen production for hydrogenation, where CH_2 represents a waste hydrocarbon. The reaction may be written

$$CH_2 + H_2O \rightarrow CO + 2H_2O$$
$$CO + H_2O \rightarrow CO_2 + H_2$$

Fuel oil and natural gas have been suggested as sources of CO_2, but so far they have not proved commercially practical. They yield a flue gas of less than 15 and 10% CO_2 respectively, but with the other considerations such as low price of oil and gas and smoothness of operation, they may be attractive as sources of CO_2.

Underground sources of CO_2 have been found in drilling for oil or natural gas; several such sources have been profitably exploited in the U. S. in connection with dry ice. One source in Mexico is of commercial importance.

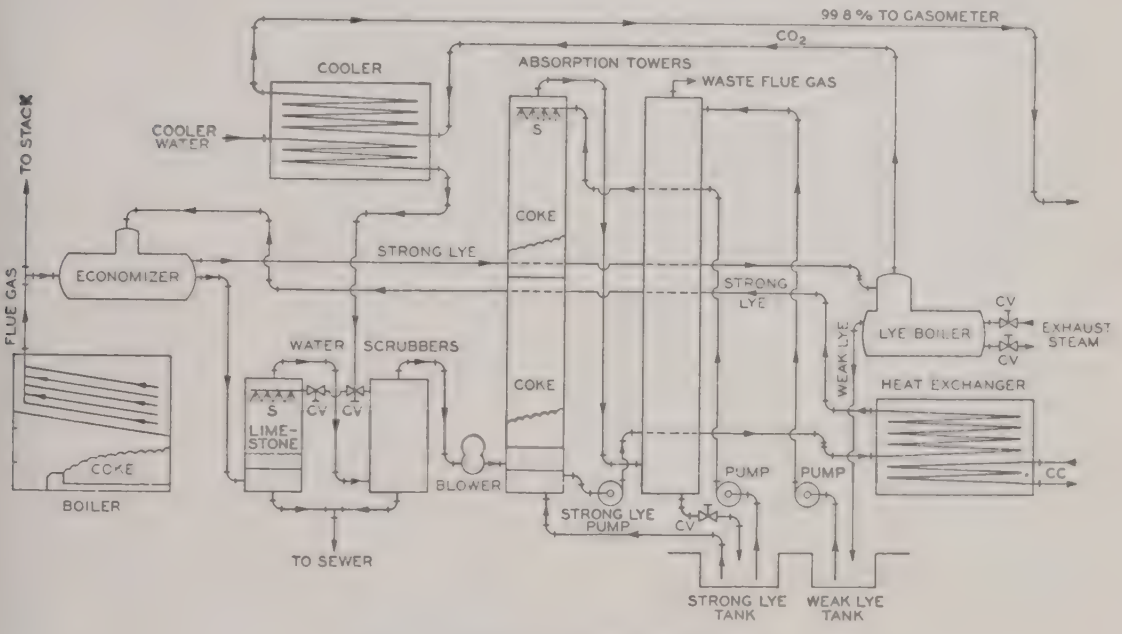

Fig. 7. Alkali Absorption Process

11. The **alkali absorption process** is applicable to those sources of CO_2 gas in which the other components of the gas mixture are relatively inert. It is used extensively in recovering CO_2 from the flue gases of coke and from limestone. It has been found commercially feasible for gases containing from 15 to 50% CO_2 and is the process most commonly adopted, **Fig. 7.**

The gases from the flue or from the kiln are first washed with water to free them from dust and sulfur dioxide and to cool them to about 110 F. This washing is done in steel or concrete towers, filled with limestone of about 6 in. diameter. These tanks are lined with acid-proof material to prevent corrosion. Water is circulated over these towers, and as it becomes charged with sulfur dioxide it is allowed to flow to waste.

The gases are next passed through the absorbers. In the type of plant most commonly built, these absorbers are steel towers, two in series, packed with coke, placed on grids at say 15-ft intervals in the height of the tower. Manhole plates are provided to permit replacing of coke when needed.

A solution containing from 7.5 to 8 lb of sodium carbonate or 12 to 12.5 lb of potassium carbonate per cu ft is circulated over the absorbers and, dissolving out some of the CO_2 in the flue gases, issues from the absorbers containing about 3 lb of sodium carbonate and 6 lb of sodium bicarbonate per cu ft.

The gas enters the absorbers with approximately 17% CO_2 content, and is vented to the atmosphere at about 8 to 9% CO_2.

From the absorbers the **strong lye** (mixed bicarbonate and carbonate solution) is pumped to a **lye boiler** where it is heated to about 240 F and thus freed of its CO_2 content. Chemically:

Weak lye:

$$Na_2CO_3 + H_2O + CO_2 = 2NaHCO_3$$

Strong lye:

$$3NaHCO_3 + Heat = Na_2CO_2 + H_2O + CO_2$$

The CO_2 gas mixed with steam is cooled in order to free it from most of the water vapor and is sent to a gasometer or directly to the compressors.

The following operating conditions for plants using sodium carbonate will contribute greatly to proper results. Summer conditions have been chosen, because that is generally the time of peak load, when control is most important.

Temperatures (solution)	*F*
Leaving last absorber—entering interchanger	120–140
Leaving interchanger—entering lye boiler	220–230
Leaving lye boiler—entering inter changer	240–250
Leaving interchanger—entering absorbing system	140–160
At pump circulating solution over first absorber	135–155
Between first and second absorbers	130–150
At pump circulating solution over second absorber	125–145
Temperatures (gas)	
Entering scrubbers	300
Leaving scrubbers	130–150
Leaving lye boilers	210–235
Leaving gas coolers	70–100
Entering compressors	25– 60
H.P. liquid CO_2 leaving water coolers	50– 70
H.P. liquid CO_2 leaving refrigerating coils	20– 50
Analysis:	
Flue gas of CO_2 leaving furnace, %	15– 20
Flue gas of CO_2 leaving absorbers, %	6– 9
Solution leaving absorbers, sodium bicarbonate, lb per cu ft	5– 6
Solution leaving absorbers, sodium carbonate, lb per cu ft	3– 4
Solution leaving lye boilers, sodium bicarbonate, lb per cu ft	2 3
Solution leaving lye boilers, sodium carbonate, lb per cu ft	5– 6

A variation of this absorption system is used by the ammonia-soda industry, consisting essentially in absorbing the CO_2 from kiln gas in strong lye under slightly raised pressure and freeing the CO_2 in lye boilers, in which heat is applied and the pressure reduced. With the use of pressure, the size of equipment is reduced, and a considerable improvement in percentage of CO_2 gas recovered is claimed.

Another variation is the use of triethanolamine as an absorbent in place of an alkaline carbonate. The reaction is

$$\text{Triethanolamine: } 2N(C_2H_4OH)_3 + H_2O + CO_2 = \text{Triethanolamine carbonate: } (N(C_2H_3OH)_3H)_2CO_2$$

The triethanolamine carbonate breaks down on heating, releasing the CO_2. Theoretically, 6.77 lb of this compound can absorb 1 lb of CO_2.

Still another absorption process is one involving the solution of CO_2 in flue gas by means of water under relatively high pressures. After the water is carbonated to its maximum under the conditions, it is passed through a low-pressure vessel where the CO_2 is freed from its solution and passes on in a pure state. This method, while old, has met with very limited favor in this country.

Many plans have been drawn and several plants have been built in which CO_2 is freed by means of compression and fractional distillation. It is quite possible to freeze out the CO_2 by such means, but the pressures are excessive when applied to gases of low CO_2 concentration because of the partial pressure of the extraneous gases (nitrogen) present. High power costs and numerous operating and maintenance difficulties have caused the abandonment of the few plants that were so designed for handling flue gas. The process has possibilities for higher concentration sources.

12. Removal of odors. Another process using activated carbon has been used for purifying CO_2 from oil wells. The gas containing hydrocarbons and sulfur comes from the wells at about 1,000 psi, and is first passed through towers containing iron oxide to remove the sulfur compounds, and then through a series of towers containing activated carbon which absorbs the hydrocarbons. Revivification is accomplished by use of superheated steam at periods of every few hours. The oxide towers are revivified by passing air through the oxide at controlled rates of flow. The CO_2 gas, free from all odorous impurities, flows from the towers through cooling coils where, at the pressures involved, it liquefies and is ready to be solidified.

One of the processes peculiarly well

suited for the removal of water-soluble odorous impurities is the **silica gel** process. In this process the gas is collected, usually from fermentation vats, and after a preliminary water scrubbing to remove the bulk of alcohol and other organic matter, it is passed through the first and second stages of compression—i.e., up to 380 psi, and then through the absorbers containing gel. When the absorber is saturated the gas is passed through a companion absorber while the first absorber is revivified with heated air.

A modification is the purification of low-pressure CO_2 with silica gel; the CO_2 from the fermenters is scrubbed and cooled to about 45 F and then at about 5 psi is forced through absorbers containing gel. The revivification is accomplished by hot air as above. This low-pressure system naturally requires larger equipment and is more costly. The manufacturers guarantee a product free from all odor.

Alumina gel is claimed to serve satisfactorily as a purifying medium for CO_2 gas.

Liquefying and Solidifying

13. The **compression** of CO_2 gas is usually accomplished in three stages, the first taking the gas from about atmospheric pressure up to 65 to 80 psi. The gas is then cooled and passes into the second stage of compression, where it is carried up to 300 to 400 psi, is again cooled, and often purified at this point. Then it is passed into the third stage where a pressure of 900 to 1,200 psi is reached. This CO_2 is usually liquefied in coils cooled with water. The liquefying temperatures for various pressures are found in the tables of thermal properties.

To convert this liquid into **solid** CO_2 the liquid may be (a) frozen by the removal of heat by external means, (b) evaporated so rapidly that a portion of it freezes, or (c) ejected through a small orifice or nozzle, under which condition a portion of the liquid freezes out as snow.

The first method was first proposed about 1895, and has been tried on a small scale but discarded. In the second method the liquid CO_2 is put in a pressure vessel surrounded by a source of cold. Inasmuch as the boiling point of CO_2 is a much lower temperature than its freezing point (-109 as against -69 F), such a source is not difficult to obtain. The liquid CO_2 occupies a much larger volume than the resultant ice, and for this reason it has been possible to supply the freezing chamber with liquid CO_2 under pressure during the freezing period. The ice produced is translucent and hard, and has a density of about 90 lb per cu ft.

When a cylinder of liquid CO_2 is opened with its valve uppermost and the gas is permitted to escape rapidly, a portion of the liquid is frozen into a porous disintegrated mass known by the descriptive name of "pop corn snow" by a constant enthalpy process as in the case of any expansion. The liquid may evaporate directly in a mold and the porous solid is pressed under hydraulic presses into dense ice. At the present time several commercial plants are utilizing modifications of this simple method.

The third general method of solidification is that involving the **expansion** of liquid CO_2 itself. When liquid CO_2 is forced through a nozzle the liquid flashes into gas and snow. The proportion of snow is dependent on several factors, particularly that of temperature of the liquid CO_2. Snow formed in this way is usually compacted by tamping and then pressed by hydraulic means. **Fig. 8** shows the power required per ton of dry ice for various cycles at various temperatures of the liquid.

Most of the various processes that are being used today are combinations of these three methods. Several of these make snow in a chamber maintained at about the triple point; in this way liquid and snow are injected at intervals into the chamber, forming a wet sludge. This is later compressed to form a dense ice. The Carba process injects the wet snow into the chamber with considerable force, and the machines are so designed that no further pressing is necessary to produce a dense ice.

Other variations in the simple conversion of liquid CO_2 into solid CO_2 arose in the methods advocated for the **precooling** of the liquid CO_2. Some prefer to cool the liquid by external means such as an ammonia refrigerating system, others by using the liquid CO_2 as a refrigerant, ex-

panding it and recompressing it through the third stage of the compressors. Others expand a portion of the CO_2 between each stage of compression much along the lines laid down by customary refrigeration practice to obtain maximum efficiencies.

14. Storage and transportation. Because of the extremely low temperature of the solid, CO_2 does not lend itself readily to long-time storage. Two theoretically possible methods of slowing down the meltage are (a) mechanical refrigeration of the storage room or (b) by means of pressure applied to the storage space; neither is feasible because of the costs involved. The ice must serve as its own refrigerant.

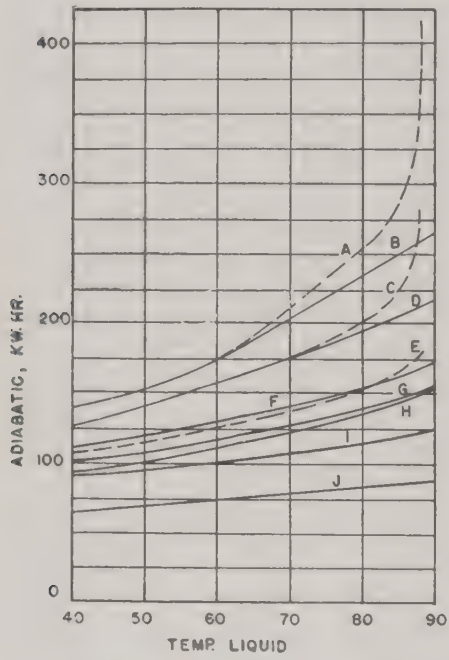

Fig. 8. Summary of Calculations of Theoretical Cycles of Snow Production

A, B, Simple cycle; C, D, same with precooler; E, F, bleeder cycle; G, H, bleeder with precooler; I, binary cycle; J-Carnot cycle. Dotted lines are based on saturation pressures, solid lines are based on optimum pressures.[1]

[1] Stickney, A. B., *Refrig. Eng.*, Dec. 1932.

Attempts have been made to store the CO_2 for several months in large insulated gas-tight silos. In these, ice made in winter and spring is to be stored for the summer peak, and the CO_2 gas liberated by the melting ice is returned to the compressors for reliquefaction and reconversion into ice. The impression is that these have not proved commercially advantageous because (a) it is difficult for men to work in the silos on account of the large proportion of CO_2 in the atmosphere, (b) the ice changes in crystal structure and becomes sugary, and (c) the capital expense and the labor costs are excessive. Storage for one month or six weeks has been successfully accomplished where the ice is of satisfactory crystal structure. These temporary storage boxes are so constructed that a man may load and unload and yet keep his head above the CO_2 gas level (see Table 2).

Table 2. Dry Ice Losses per 24 hr, %

Shipping carton for 50 to 100 lb of dry ice, insulated with dry wood pulp	15–20
Shipping box for 200 lb of dry ice (3-in. insulation)	8
Shipping box for 2,000 lb of dry ice (5-in. insulation)	3
Shipping box for 4,800 lb of dry ice (6-in. insulation)	1.5
Insulated truck, well-type body	1.2
Insulated railway car, well-type cargo space	0.8
Plant storage, well-type storage box	0.3

The **well-type storage container** is the only one commonly used today. The exterior walls and bottom are generally constructed of sheet metal and sealed against leakage of cold air. Several inches of efficient insulating materials are interposed between outer and inner walls. A light hinged lid or detachable cover forms the top of container. A mattress is often provided to protect the top surface of the solid CO_2 stored within the box. Storage containers vary in size from 8 to 54 cubes capacity, the standard cube being 10 cu in.

Shipping boxes generally follow the design of storage containers except that weight is kept to a minimum through the use of less insulation and lighter structural members. The capacities of shipping containers for solid CO_2 vary from 1 to 8 cubes.

There are now in use specially designed trucks and cars for transportation of solid CO_2. These are of the top opening or well type with a number of separate compartments, each with independent covers.

Shipment of dry ice involves many difficulties, losses up to 30% being typical where the product is submitted to several

handlings, regardless of the distance shipped.

Dry ice from a chemical analysis is always of a **purity** above 99% CO_2, but for commercial use its quality is judged mainly for freedom from odor and for structure. Dry ice from fermentation sources is examined primarily for the **odor** of beer or molasses; from combustion of fuels, from lime kilns, etc., for odors of hydrogen sulfide or the odor of "smoke"; from natural wells for odors of hydrogen sulfide and gasoline; and from all sources for an odor of oil brought about by incorrect lubrication of the compressors.

In **structure** dry ice should be neither soft nor too brittle. Ice that is well made and is of satisfactory density may be sawed into thin slices 10 in. $\times$10 in. $\times$ $\frac{1}{4}$ in. and these slices be sufficiently rugged that they may be handled. Densities of 85 to 95 lb per cu ft (equivalent to 50 to 55 lb per 10-in. cube) have been found most satisfactory.

Summary of Uses

15. As a refrigerant dry ice is convenient, economical in application to both stationary and mobile units, and susceptible to a readily controlled temperature range from the intense cold of frozen food to that of the moderate temperature of a butter box. It can profitably be used as a refrigerant in the transportation and dispensing of unfrozen perishable foodstuffs such as fish, meat, dairy products, fruits, and vegetables, as well as frozen foods such as fish, meat, dairy products, fruits, and vegetables. It is also used in the metal-working industry, electrical industry, chemical industry, in the laboratories, and in certain applications of food freezing and of air conditioning.

As a source of carbon dioxide gas, dry ice offers economies in cost of production, transportation and storage. It can be profitably used for the innumerable purposes to which carbon dioxide itself is put:

a. In carbonating beverages
b. For preserving perishables
c. In the canning and food packaging industry
d. In the chemical industry
e. In fire prevention
f. In the medical profession
g. As a source of power
h. In miscellaneous uses.

Low Pressure Air Liquefaction

16. During the latter part of World War II and subsequently, extensive developments have been made for producing oxygen and nitrogen from air in low-pressure systems (less than 100 psi). This involves extensive use of efficient heat transfer surface and use of an expander engine (see ref. 9).

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27. LOW-TEMPERATURE SYSTEM OPERATION AND PRACTICE

PRIOR to World War II applications of refrigeration equipment for the production of temperatures below approximately −40 F were unusual, except for cases of liquefaction of air and various other gases. Due, however, to aircraft developments and the higher altitudes at which planes were flying, it became necessary to develop a program for "altitude testing" of the various components of an aircraft system that were affected by low temperatures and high altitudes. These developments added to the need for test facilities at temperatures ranging from −40 F on down.

For refrigerant temperatures above −40 F single-stage operation is generally satisfactory. This, however, can only be achieved under favorable circumstances with normal condensing temperatures, due to the high compression ratio across the refrigerant compressor and the consequent inefficiency of the compressor. In general then, the production of temperatures below −40 F requires a somewhat special technique, different from that commonly used with the ordinary refrigeration compressor.

Freon Equipment Well Adapted to Low Temperatures

Ammonia, long used for refrigeration at all levels of temperature, has been successfully employed for the production of refrigeration temperatures as low as −90 F. This is approximately the low level for ordinary use, inasmuch as it approaches the triple point of −107.86 F. Very satisfactory results have been obtained using ammonia, but most manufacturers employ low-speed compressors for ammonia duty as compared with the commonly used high-speed Freon compressors. In addition, the compact, highly efficient and inexpensive low-sides that can be employed with the Freon refrigerants are not generally available for use with ammonia.

Freon-12 and **Freon-22** are also commonly used for the production of low refrigerant temperatures. Freon-12 is satisfactory for use down to approximately 0.5 psia, which corresponds to an evaporator temperature of −125 F. Freon-22 can be used to a somewhat lower temperature if the same limitation on absolute pressure at the compressor suction prevails, making the low limit for this refrigerant about −137 F. However, when using reciprocating compressors it is inadvisable to apply these limits, −115 F being a practical limit for Freon-12 and −125 F for Freon-22.

To illustrate the compression ratios involved in the compression of various refrigerants from low temperatures, **Fig. 1** shows the overall compression ratio for ammonia, Freon-12, and Freon-22 from suction temperature to a fixed condensing temperature of 100 F.

At the lower temperatures and extremely low absolute pressures, a very low density gas is available at the suction of the compressor. It is therefore essential that **suction valves** with little or no spring tension be employed to insure the filling of the cylinder in the suction stroke.

When handling rare gas, ample consid-

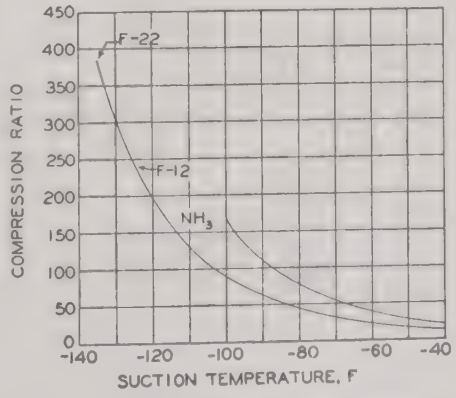

Fig. 1. Compression Ratio versus Suction Temperature at 100 F Condensing Temperature for Freon-22, Freon-12, and NH_3

eration must also be given to the effect of **suction line superheating** of the gas as well as **cylinder reheating** inside the compressor. Due allowance must therefore be made for the increase in gas volume which requires greater compressor displacement than would theoretically be considered. Centrifugal compressors and rotary compressors have also been used satisfactorily as boosters on low-temperature refrigeration, when the volume of gas handled warrants the selection of these somewhat larger compressors. Use of the centrifugal compressor and rotary compressor has been very satisfactory, due to lack of difficulty with the valves, as mentioned above with the reciprocating compressors, and also due to the minimizing of reheating of the gas when entering the compressor.

Compressor displacement may be obtained from **Fig. 2,** which shows cfm per ton for the various refrigerants. In determining these displacements, it is necessary to use the volumetric efficiency for the compressor involved. These displacements have been computed from the thermodynamic properties of the refrigerants, using a reheat factor based on figures obtained in practice. Furthermore, it has been assumed that a maximum compression ratio of 6 was used on the compressor, which in turn fixed the maximum amount of liquid cooling available.

Except for the fact that Freon-22 has higher absolute pressures and therefore can be used satisfactorily to somewhat lower temperatures, there is little to choose between Freon-12 and Freon-22. Freon-22 requires somewhat smaller compressor displacement, permitting a first cost saving in this item. The compressor, however, runs somewhat hotter since Freon-22 has a higher adiabatic exponent than Freon-12. Since the heat transfer equipment will be identical for Freon-12 and Freon-22 and the motor horsepower will be about the same, the overall saving in the system is small with Freon-22. Its use, however, is justified in many cases on the basis of the small compressor size and somewhat higher suction pressures involved. A further practical advantage of Freon-22 is that smaller refrigerant quantities are in circulation through the mains which, coupled with the higher density, leads to somewhat smaller line sizes, with a consequent saving in cost of the line and insulation.

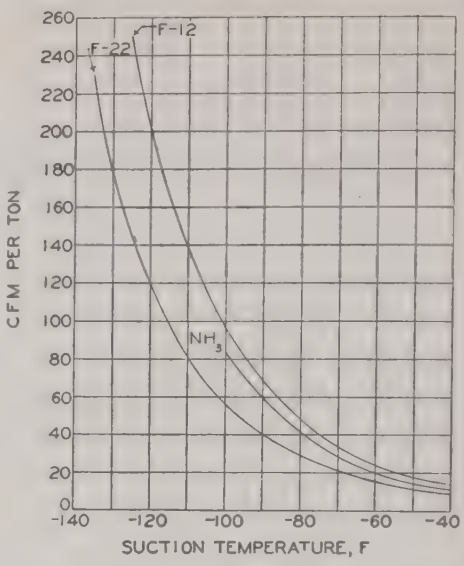

Fig. 2. CFM/Ton versus Suction Temperature at Liquid Temperature Based on 6 Ratios of Compression

As previously mentioned, Freon equipment is preferred to ammonia equipment due to the high speeds used with Freon equipment and the resultant saving in size involved. It should also be understood that Freon is preferable because of the inherent advantage of a refrigerant that is nonflammable and non-toxic. Its hazard in use is considerably less than that of ammonia.

Multi-Stage Compression Systems Required

Whether a two or three-stage compression system is to be used is dictated by the limitations of compression ratio on the compressors involved, as well as by operating cost and first cost. This is true when deciding whether to run one or two-stage compression, or whether to run two or three-stage compression. In general, practice has indicated that, for temperatures below −75 F, three stages of compression are usually required.

In order to give some idea of the horsepower per ton requirements, **Fig. 3** has been prepared, showing the approximate indicated horsepower per ton for the various refrigerants employing two or three-stage compression.

For the production of temperatures below −40 F multi-stage compression in

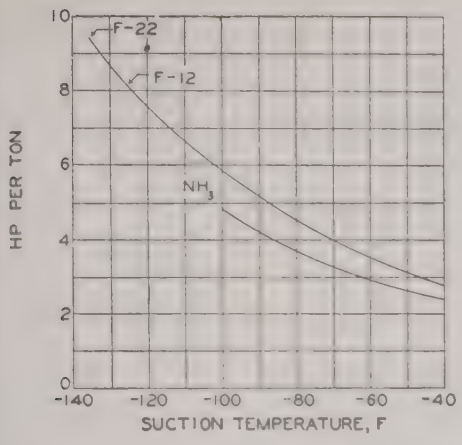

Fig. 3. Indicated Horsepower Per Ton versus Suction Temperature at 100 F Condensing Temperature

some form is generally required. The first and most simple method of multi-stage compression involves the use of **compressors in series,** the discharge of the low-stage compressor being the suction of the intermediate or high-stage compressor. Usual practice limits the selection of compressors to two or three stages.

The second method, called the **cascade system** of refrigeration, employs a series of gases of progressively lower boiling points. Each gas is used as a cooling medium to condense under pressure the gas of the next lower boiling point. This method is often used in the liquefaction of nitrogen.

For the common applications of low-temperature refrigeration encountered during the war, there is little justification for using a cascade system in the range of temperatures from −40 F down to approximately −135 F. Below the figure of about −135 F its use is justified with a refrigerant such as ethylene or several refrigerants of the Freon series which are not yet readily available. In these cases the cascade system is justified only because the pressures involved at normal condensing temperatures are so high as to require special high-stage compressors. Advances in the art of low-temperature refrigeration have indicated that there is little or no reason for cascading when using Freon-12 or Freon-22 as the low-temperature refrigerant.

One of the objections to the use of a cascade system lies in the fact that there must be an overlap of temperatures in the condenser for one refrigerant, which condenser is the evaporator for the next higher stage refrigerant. This overlap of temperatures often leads to inefficiencies in operation, with somewhat greater displacement and horsepower being required. Furthermore, it has been found to be extremely difficult to operate a cascade condenser with a pressure much below 5 psig. Attempts to operate in a vacuum or at approximately atmospheric pressure have not been too successful, due to the fact that a small amount of air in the condenser cuts the condenser performance considerably. A further difficulty with cascade condensers when operating at low condensing pressure is that the differential between the evaporator pressure and condenser pressure is so small as to make difficult the use of commonly employed refrigerant control devices, such as thermal valves or floats. The low pressures involved makes the use of extremely large control valves mandatory.

Use of Intercoolers

It therefore appears that compound compression or multi-stage compression is desirable whenever conditions permit. The operation of a multi-stage compression system requires cooling of the gas between stages in order to prevent overheating of the compressor when taking in a highly super-heated gas with a consequent rise in temperature through the compressor. At the same time it is desirable to employ **liquid cooling** between stages to increase the availability of the liquid delivered to the evaporator and thereby reduce the volume requirements of the low-stage compressor. **Intercooling** is more important with ammonia equipment than with Freon equipment because of the higher adiabatic exponent for ammonia. Liquid cooling is, however, more important with Freon than with ammonia, since the Freons have lower latent heats and any increase in availability is therefore of more significance.

With multi-stage compression, and with the discharge temperatures generally encountered, it is essential to provide for inter-stage gas cooling. With Freon refrigerants this is readily accomplished with small equipment and thermal valves. Un-

fortunately, thermal valve control of ammonia intercoolers is not feasible, making the use of float-controlled intercooling equipment necessary.

Intercooling equipment has two duties to perform: (1) to cool the gas to each machine; and (2) to cool the liquid ammonia on its way from the high-stage condenser to the low-stage evaporator. The most important function is that of gas cooling, to avoid burning up the machines due to high discharge temperatures. The least important function is the liquid cooling, since the ammonia liquid cooling is not so important as with Freon due to the difference in latent heat of the two refrigerants.

Two types of **ammonia-cooled intercoolers** are generally used for a combination of gas and liquid cooling:

1. Flash-type gas and liquid cooler
2. Shell-and-coil gas and liquid cooler.

Both types cool gas by bubbling it up through a body of liquid at some intermediate temperature. The **shell-and-coil cooler** provides cold liquid by means of a surface-type heat exchanger, in which a pipe coil is submerged in cold liquid of intermediate temperature, and through which coil the liquid to be cooled flows on its way to the low-stage evaporator. The **flash-type cooler** separates the liquid and flash gas resulting from the expansion of liquid at condenser pressure to the intermediate pressure; in the course of the expansion, the liquid is cooled to intermediate temperature.

Figs. 4 and **5** show diagrammatically these two forms of intercoolers for use with ammonia. They are generally sized as a gas and liquid separator, insuring a velocity sufficiently low that liquid may be separated from the gas which is withdrawn from the shell on its way to the compressor.

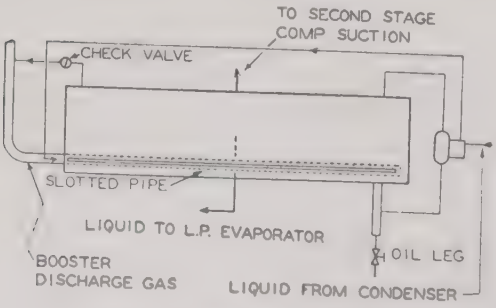

Fig. 4. Flash-Type Intercooler

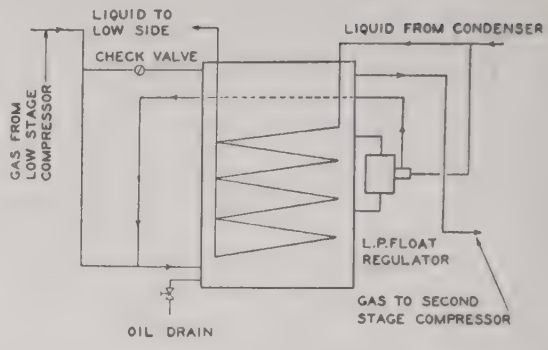

Fig. 5. Shell-and-Coil Gas and Liquid Cooler

Both types of ammonia intercoolers have merit. Utilizing the flash-type cooler makes it possible to cool the liquid proceeding to the evaporator down to the temperature corresponding to the intermediate pressure, at a sacrifice, however, of pressure differential which can be used for the operation of liquid control valves at the evaporator. The advantages are two-fold in that a slight thermodynamic gain is obtained with the lower liquid temperature, and a simple piece of equipment is possible, since a bare shell is used in place of a surface-type cooler.

The flash-type cooler, however, adds several disadvantages. The first is the loss of pressure differential, which means large liquid control valves and possible sluggish action. Another disadvantage is that any heat pick-up in a long liquid line between the flash-type cooler and the liquid control valve means that gas is generated in the liquid line. Such gas is extremely difficult to get through the liquid control valve and as a consequence the control valve becomes gas-bound. On three-stage ammonia systems the last stage of liquid cooling should always be made shell-and-coil type; if this is not done, on this third stage there would be very little pressure differential remaining to drive the liquid into the evaporator.

When only gas cooling is required between machines, the flash-type cooler may be utilized with the omission of the liquid connection to the L.P. evaporator.

A piece of intercooling equipment often used in connection with ammonia systems

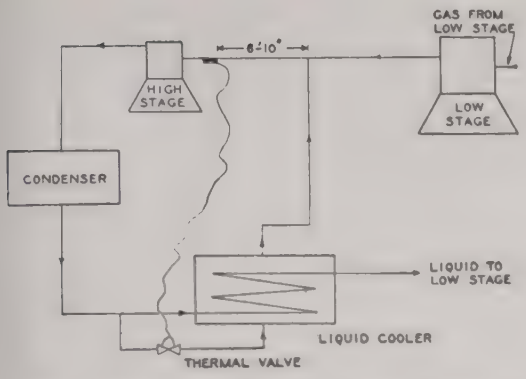

Fig. 6. Freon Liquid and Gas Cooler Hook-up

is a **water-cooled gas cooler.** Due to the fact that the discharge temperature from some of the compressors is sufficiently high to permit cooling by the use of water, these coolers are sometimes used. The amount of heat that they take out of the gas is, however, so small that no appreciable operating saving can be made. The slight savings that can be made in operating cost are in most cases overshadowed by the initial investment requirement for the piece of equipment as well as the additional cost of installation.

Both flash-type and shell-and-coil type liquid and gas coolers were used at one time on Freon systems, but they have been largely replaced by the shell-and-coil type liquid cooler, which is considerably simpler and is used only for cooling of the liquid. Gas cooling is obtained by proper mixing of the wet vapor with superheated discharge gas from the compressor. **Fig. 6** indicates a typical hook-up showing the position at which the thermal valve bulb is to be located to provide for the proper gas cooling.

Use of Oil Separators

The oil problem in connection with the compressors used on low-temperature refrigeration systems has been minimized by the use of oil separators. On a multi-stage compression system using shell-and-coil intercoolers, the **discharge line oil separator** on the high-stage compressor is all that is required to keep the oil in circulation in the system to a minimum. On systems using flash-type intercoolers it is essential to use a discharge line oil separator not only on the high-stage compressor but also on the compressor discharging into the flash-type intercooler. The use of discharge line oil separators keeps the oil where it belongs —in the compressors, allowing a minimum to go over to the low side.

In the case of ammonia systems, oil on the low side is detrimental to heat transfer effect, since the oil film has low thermal conductivity and tends to produce poor heat transfer. This same effect is not, however, encountered with Freon systems, because in the case of Freon-12 the oil and Freon are completely miscible, and it takes considerable oil concentration in the refrigerant to effect heat transfer. In the case of Freon-22, there is a separation of oil and Freon and the oil forms a separate phase lying on top of the Freon, not blocking the heat transfer surface nor causing too much difficulty in operation.

Oil can, however, cause some difficulty in ammonia and Freon-22 systems in connection with the operation of the liquid control devices. If too much oil circulates with the liquid refrigerant, it tends to accumulate in the liquid control valve, gumming up there and blocking off the flow of refrigerant through the valve. The use of the discharge line oil separator generally keeps down the flow of oil with the liquid to the point where no difficulty is encountered.

Due to the low mass flow of refrigerant through the compressor when operating at these low temperatures, the compressors tend to overheat. For small-size compressors with splash lubrication the surface area of the compressor is generally sufficiently large to dissipate this heat without difficulty. However, with larger compressors this is not the case, but fortunately many of these compressors utilize forced feed lubrication of the various rubbing surfaces, and, therefore, it is possible to add an external oil cooler to the oil circuit. This effectively removes the heat and maintains the oil at a temperature that does not cause difficulty in operation.

Sizing of Motors

As in all low-temperature applications, the compressor can run under heavy loads during periods of high suction pressure,

especially when starting when the system is warm. To take care of this situation it is therefore necessary to make adjustments to the motor horsepower and thereby prevent serious overloading of the motor during the starting period. Careful consideration must be given to sizing the motor, or special attention must be paid to the operation of the system when starting initially. If the compression system is to operate only at a fixed low temperature, it is generally possible to avoid the oversizing of motors to any great extent, providing for careful operation when the system is first put on the line. This method will permit putting in motors for the actual operating brake horsepower without any additional motor sizing.

For those cases where a definite pull-down time is required for the reduction in temperature from ambient conditions to some extremely low temperatures, the compression system will be operated at high suction pressures for considerable periods of time. For this reason it is necessary to oversize motors. General practice is to drive the high-stage compressor with a motor that will carry the compressor at the highest expected evaporator temperature. This is generally the "air conditioning" rating of the unit, in the case of Freon units. For intermediate or low-stage compressors, it is generally sufficient to size the motor to take care of double the balance load indicated horsepower plus the friction horsepower.

In line with recommending larger motors in cases requiring a pull-down, it is also necessary to give some consideration to the condenser that is used. It is generally advisable to oversize the condenser to some extent, certainly making it larger than the size required for the balanced load condition. Water-cooled condensers are not too critical in this regard, but extreme caution must be taken with economizer or evaporative condenser applications, since this type of condenser is more easily overloaded than the water-cooled condenser.

Use of Low Sides

Some mention should be made of the low sides that are generally used in low-temperature refrigeration. With ammonia systems, **gravity-feed air coils** are frequently used with a large surge drum, refrigerant flow being controlled by a high-side float. Flooded **shell-and-tube coolers** are also used for cooling liquids such as trichlorethylene or methylene chloride.

For Freon-12 or Freon-22, various types of coils are used for cooling air, depending to a great extent upon the temperature level and the type of application. For rooms having considerable mass in which a slight variation of temperature of the air off the coil is not critical, thermal valves and direct expansion coils can be used satisfactorily for air temperatures down as low as approximately −70 F. Over the same temperature level, for wind tunnel applications, the direct expansion coil does not give as satisfactory results because of fluctuation in air temperature during a thermal valve cycle. For this reason gravity-feed coils have been utilized to avoid the cycle of the thermal valve and maintain reasonably uniform air temperatures leaving the coils.

As temperatures get well below −70 F, it is not possible to allow the necessary refrigerant pressure drop through the direct expansion coils, nor is it always possible to take the static head penalty that results with the gravity-feed coil. It is therefore essential on these applications to resort to pump recirculation of the refrigerant through the coils, using a down feed of the evaporator coil and thereby preventing static head penalties. In this type of pump recirculation, evaporation of the refrigerant takes place inside the tubes of the evaporator and a mixture of gas and liquid returns to a pump receiver in which the liquid and gas are separated, the liquid recirculating by means of the pump and the gas returning to the compressor.

Another type of liquid recirculation is sometimes used on temperatures below −70 F, depending to a great extent upon the size of the job and the space requirements for the evaporator itself. This method involves the **circulation** of the **refrigerant** in the **liquid phase,** refrigeration effect being taken care of by a warming up of the liquid refrigerant as it passes through the evaporator. The liquid refrigerant which is maintained under pressure is then allowed to flash into a large vessel, this flashing serving to reduce the liquid re-

frigerant to the pressure and temperature being maintained by the compressor. The liquid and flash gas are separated in the vessel and the liquid is then recirculated by means of the pump through the coil. The gas is taken to the compressor to complete its cycle.

For cooling liquids such as trichlorethylene or methylene chloride, shell-and-tube equipment is used with flooded operation for the small sizes where static head penalty is not too severe, and for larger sizes the liquid refrigerant is sprayed on the outside of the tubes.

Selection and Operation of Controls

Numerous types of liquid refrigerant control are available for use on low-temperature systems. For both ammonia and Freon, **high-side floats** are preferred to low-side floats if there is any choice. High-side floats operate at a much higher temperature level as far as the float body and orifice are concerned, and therefore are less susceptible to difficulties from temperature variations or oil in circulation.

Where liquid coolers are used for precooling of the liquid on the way to the evaporator, a pilot-type high-side float is necessary. This is true inasmuch as it is desirable to keep the condenser drained, but at the same time insure a seal on the liquid line to evaporator.

Fig. 7 illustrates a form of pilot-type float which uses a float switch in conjunction with a solenoid valve close to the evaporator. Low-side floats can also be used where there are multiple evaporators. Care must, however, be taken to avoid congealing oil in the float body which can cause considerable difficulty. Liquid legs from the evaporator to the float must be carefully trapped and the line sizes should be kept large.

It should be noted that neither type float automatically provides for oil return to the compressor on Freon systems. It is sometimes possible to utilize an **oil still** on a float control job. The oil still is best used in conjunction with a liquid refrigerant pump, which can be used to provide sufficient lift for a small sample of the refrigerant that can be put through the oil still. The oil still merely takes heat from the liquid refrigerant at a higher temperature, boiling off the refrigerant and sending a mixture of refrigerant gas and liquid oil back to the compressor.

Automatic oil return can be obtained on Freon systems using a properly installed **thermal valve feed.** Thermal valves can be used for direct-expansion coils or can be used to feed refrigerant to a surge drum on gravity-feed jobs. In both cases it is advisable to utilize a superheater in conjunction with the evaporator, to provide dry refrigerant gas returning to the compressor and also give sufficient power for the operation of the thermal valve.

The operation of the system also calls for the use of a suitable low-temperature lubricant. This is not necessary from the standpoint of the compressor, but once it is recognized that some of the oil from the compressor inevitably passes over to the low side, its need can be seen. The oil that is used should be water-free for use in all refrigerant systems and should also be as wax-free as possible. It has generally been noted that wax comes out of solution at lower temperatures and can easily accumulate at control valve orifices, thereby causing a restriction. The oil should also have a sufficiently low pour point that it does not congeal in the lines. This is more important with ammonia than it is with Freon-12, since the solution of Freon-12 in the oil effectively depresses the pour point. This is true also to some extent in the case of Freon-22, although separation of oil and Freon into two separate phases takes place at temperatures as high as 20 F.

The control of the refrigeration of low-temperature systems depends to a considerable degree upon the application itself.

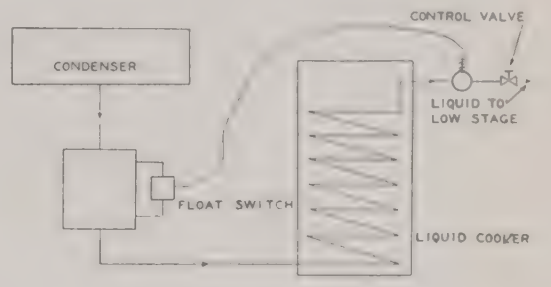

Fig. 7. Pilot High-Pressure Float

Starting and stopping of the compressors has been utilized to maintain temperatures in large spaces where there is considerable flywheel effect. It is necessary, however, to have some temperature fluctuation in order to actuate the temperature controller, which causes the temperatures to vary inside the conditioned space. Where these variations are not too serious, and sufficient temperature change can be utilized to prevent short cycling of the equipment, reasonably satisfactory results can be obtained. For multi-stage compression systems this means that some provision must be made for time delay between motor starters in order to prevent all machines from coming on the line at once.

If too close a temperature control is not required, it is often possible to permit the machines to rebalance at lower temperatures, since as suction temperature is reduced machine capacity falls off very rapidly. This slip in temperature is generally not too great as long as excess capacity has not been provided for a pulldown or for large operating loads that may fail to exist.

It is also possible to throttle the suction to the low-stage compressor, thereby providing for temperature control. This is generally dangerous from the standpoint of the machine since it causes the machine to run considerably hotter than normal, and, in the case of forced feed lubrication of the compressor, may cause the oil pump to fail because of too low an absolute pressure suction to the oil pump. When the dangers involved in the use of a throttled suction are recognized and suitable provision is made against the difficulties that may be encountered, this method does give reasonably good control.

One of the best methods for temperature control that can be made automatic is "hot gas bypass," which provides for artificially maintaining the load on the compression system by supplying gas from the high side through a control valve, supplementing the gas that is obtained from the low side. As the load falls off and less gas is provided by the low side, more gas is provided through the hot gas bypass control valve. This method takes more operating horsepower but gives very close control of the system.

28. AUXILIARIES AND THEIR OPERATION

BEFORE any refrigeration system is put into operation, it is necessary that certain precautions be taken in order to obtain good efficiency from the standpoint of both economical performance and maximum life of equipment.

In line with these precautions the most important considerations are:—

I. Preparation of new systems
   1. Cleaning
   2. Tightness
   3. Removal of moisture and air
   4. Charging.

II. Operation of systems
   1. General operation
   2. Instrumentation
   3. Moisture
   4. Oil
   5. Purging.

I. Preparation of New Systems

1. Small or complete refrigerating systems of the package type are **cleaned** by the manufacturer, by the circulation through coils, shells and tubing of an agent for dirt elimination. Compressor parts are thoroughly cleaned during assembly.

In large systems, shells, coils and compressors come from the manufacturer free from dirt, and piping that is erected during installation should be thoroughly cleaned as erected. After the test for tightness is made, the entire system should be blown down to remove dirt and foreign matter.

2. Package equipment and factory-assembled systems are usually tested for **tightness** in conjunction with air and moisture removal (see Par. 3).

Large systems erected on the job should have the entire piping work tested with compressed air for leaks under pressures up to those specified for equipment using various refrigerants. These air pressures are obtained generally by use of the refrigerant compressor. When pumping a pressure on the system it is well to do so in stages rather than all at one time, in order to avoid excessively high air temperatures such as may cause explosion. If the pressure is raised to 50 psi and tested for major leaks, then brought up to the final test pressure in two or three more pumping operations, no trouble will be experienced. When the machine is used as an air compressor the usual amount of cooling water must be circulated.

After the system has been made tight under air pressure, all coils, coolers and pipe work generally should be **blown out** to remove dirt. The system should be tested under vacuum as a final check for leaks; when subjected to external atmospheric pressure, small pin holes closed by dirt while under internal pressure will often show up.

Before starting the compressor to pump up an air pressure, it is important that the cylinders be well **lubricated.** If the machine is fitted with a mechanical lubricator for supplying oil directly to the cylinders, oil may be pumped to the cylinders by the hand attachment. If the machine is not provided with such a lubricator, the cylinder heads should be removed and the cylinder walls well covered with oil.

Before starting the air test, open the stop valve on the compressor discharge pipe, and all other valves between the compressor and inlet valves to the various cooling coils or evaporators, and close the suction stop valve at the compressor inlet. Then open the valve on the suction inlet to the compressor between the suction stop valve and compressor to permit air to be drawn into the cylinders when the compressor is started. If no valve has been provided at this point it will be necessary to open a flange to provide the necessary air intake.

With the compressor in operation, pressure will be built up on the entire high side of the system, and finally brought up to 300 psi in the case of ammonia. After the high-side test is finished, the low side

should be tested and made tight at about 150 psi. All joints, valve stems and the like are then tested again for leaks.

If the compressor is driven by either a steam engine or an internal combustion engine, it may be operated at slow speed during the air test operations. If driven by a constant speed motor, it will operate at full speed, building up the air pressure more rapidly; for this reason the pressure gage should be closely observed by a man stationed at the starting switch to stop the machine promptly when the pressure has been raised to the predetermined point.

After the air test is completed, the cylinders are opened for examination. The scale trap or strainer in the suction line at the compressor is also cleaned out.

3. Package equipment and factory-assembled systems are **evacuated** and **dehydrated** by the use of high vacuum pumping and heating by oven or other means so that the systems can be brought down in some cases as low as 50 microns (.001 millimeter absolute pressure). This procedure also serves as a tightness test.

Large plants erected on the job should have had most of any moisture removed during the blow-down following the test for leaks. Air removal is done by the use of a small portable pump-down compressor which usually evacuates to 25 in. of mercury and leaves the system commercially free of air.

4. Small factory-built systems and package equipment are **charged** with definite quantities of refrigerant, predetermined during the process of development for maximum efficient operation.

Large systems are charged with refrigerant usually introduced into the system through a charging valve and connections located on the outlet of the liquid receiver between the receiver outlet valve and low-pressure side of the system. A drum of refrigerant is connected to this charging valve and the compressor started. The drum-charging valve is then opened partly, as are also one or more expansion valves.

Liquid refrigerant will then flow from the drum through the valve, which is opened wide, charging connection and liquid line to the cooling coils, where it will be evaporated by the absorption of heat; then in the form of a gas it will pass through the suction line to the compressor from which it will be discharged to the condenser. As condensation occurs, the liquid will collect in the receiver, where its presence will be indicated by the receiver gage glass.

Since the outlet of the receiver is closed, none of this ammonia can pass out to the coils; therefore as the charging operations proceed it will continue to collect in the receiver until the system is fully charged. It is, however, a good plan to put only one or two drums in the system at first, then stop the compressor and make a careful check for possible leaks.

A good method of detecting ammonia **leaks** is to go over the entire system with lighted sulfur sticks, as any mixture of sulfur and ammonia gas will produce a white fog. In the case of brine tanks, the coils should be similarly tested before making brine. Leaks in ammonia condensers may be detected by the use of litmus paper held in the stream of the outlet water. The presence of ammonia in the water will be manifested by a discoloration of the litmus paper, white litmus paper turning red in water containing ammonia.

Testing for Freon leaks is done by means of the Halide torch.

During charging operations it is of course important that water be circulated over the condensers.

II. Operation of Systems

1. General operation.

a. Starting and stopping. Modern compressors are automatically started and stopped with predetermined temperature changes in the refrigerated rooms, usually of the vertical enclosed type, driven from an electric motor.

Where automatic control is employed, no opening or closing of the stop valves or bypass valve is necessary, since the motor, which is usually of the double squirrel-cage type having a high starting torque, will start the compressor without difficulty and without manual attention.

In the case of the synchronous motor-driven compressor, of either the belted or direct-connected type, special equipment provided for starting and stopping operations includes a push-button start and stop arrangement which makes it necessary

only to push the starting button to start the compressor after the operator has closed the suction and discharge valves on the machines and opened the full size bypass. After the motor has been brought up to full speed, the discharge valve is opened while the bypass is closed, after which the suction stop valve is opened slowly and the load gradually put on the machine.

The same procedure is followed when starting other non-automatic compressors of either the vertical or horizontal type.

After the compressor is started and the suction stop valve is fully opened, the liquid outlet at the receiver is opened in order that the evaporator may be supplied with liquid.

In the case of manually controlled machines, the first step in shutting down is to close the main liquid outlet valve at the receiver; after the evaporating pressure has been pumped down to a few pounds above zero the compressor is stopped and the suction stop valve closed. This simple operation is all that is necessary for a small plant.

In larger plants, where two or more machines may be in use, one or more of them may be taken off the line by simply stopping the compressor and closing the suction stop valve.

b. General operating pressures. The operation of any given plant will depend of course upon the conditions to be met, but the fundamental principles of operation are the same for all types of installations.

The evaporating pressure, commonly called the **suction pressure,** is determined by the room and brine temperatures required. For some types of coils it is necessary to maintain a temperature difference of from 10 to 12 F between the room or brine temperature and the evaporating ammonia. Other types of evaporators, such, for example, as those used in the more modern freezing tank design, require a temperature difference of only from 2 to 3 F.

Similar variation is found also in rooms using evaporators over which the air is moved at high velocity by means of a fan. The reason for this difference in performance is that air, brine, or other media give up heat more rapidly when moved over the cooling surfaces rapidly. As previously stated, the operator must determine from experience what evaporating pressure is best suited to the particular conditions, but in any event this pressure should be carried as high as possible in the interest of power costs.

The amount of surface in any evaporator will have a direct bearing on capacity; in cases where the surfaces are insufficient, a much lower evaporating pressure will be needed.

The **condensing pressure** should be held as low as possible. It depends upon the amount and temperature of the condensing water available, and the total amount of cooling surface, as well as, to some extent, on the type of condensers used. Where liberal condensing surfaces have been provided, it should be possible to hold the condensing pressure to that corresponding to a temperature of about 8 to 10 F above the temperature of the water leaving the condenser.

2. For maintaining high operating performance, **pressure gages** and **thermometers** should be installed at several points throughout the plant and periodic checks of certain relations made. It is well to have pressure gages for both suction and discharge of the compressor and also thermometers at both places. Refrigerant gages can be obtained which have a temperature scale corresponding to the pressure scale. By comparing the temperature of the suction gas with the saturation temperature on the suction gage the superheat in the suction gas is obtained. When such superheat is from 5 to 15 F, a good condition prevails.

Where operation is at normal conditions and proper suction gas superheats prevail, an average normal discharge temperature results. Should the discharge thermometer show considerably higher temperatures than normal, there is trouble in the compressor due to leaky or broken valves or piston rings and an examination of the compressor is necessary.

With compressors equipped with forced lubrication, there should be a gage on the oil pump which when compared with the suction gage will show oil pressure. This pressure should be that specified by the manufacturer for satisfactory lubrication.

A pressure gage at the condenser gas

inlet and a thermometer in the condenser liquid outlet are used to check condenser performance by comparing the liquid temperature with the saturation temperature for the gage pressure at condenser inlet.

If the liquid temperature shows more than 20 F below the saturation temperature when the condenser is a free draining one, there are sufficient non-condensible gases present so that purging of the system is indicated.

Thermometers on the water inlet and outlet of the condensers will show any change in both initial and outlet water temperatures, which in turn will serve to indicate when the condensers are dirty and need cleaning. For example, suppose the inlet water temperature is 75 F and the outlet temperature 85. If, after a day's operation it is found that while the inlet temperature remains constant the outlet temperature continues to decrease and approach that of the inlet temperature, this is conclusive proof that the water surfaces of the condensers are becoming dirty. As a consequence, heat transmission has been so reduced that the ammonia is prevented from giving up its heat to the condensing water.

3. Moisture in a system may cause troubles such as copper plating or freezing in an expansion valve, particularly where the water is not soluble in quantity in the refrigerant. In the case of ammonia such moisture will form aqua ammonia in the low side and affect the boiling point.

Moisture may be admitted to a system when opening the low side for repairs, thus admitting humid air. The moisture content of the system may also be increased by an air leak on the low side or a very slight water leak in commercial equipment using a water-cooled condenser. In case of failure through ice formation in the expansion valve, a drier may be installed in the liquid line for a sufficient period of time to remove the moisture, if it has occured from air being admitted to the system. However, an air leak or water leak will require the necessary repairs, dehydration of the system, and recharging with dry refrigerant and dry oil.

4. Oil contamination due to oil pumping of the compressors may cause inefficient performance of parts of the system.

A great portion of the oil leaving a compressor discharge may be returned to the crankcase from suitable oil separators installed in the discharge lines. However, oil separators are not 100% efficient in performance, and hence some oil passes on to the condenser and is carried with the condenser liquid into the low side or evaporator. When the refrigerant is such that oil is not miscible to any great extent, as is the case with ammonia, oil will collect in the low side and will have to be removed by draining.

If oil is miscible with the refrigerant it may return with the suction gas to the compressor, which is usually provided with means of returning such oil to the crankcase. Or if the evaporator will not return oil with suction gas, either draining or distilling may be used for its elimination.

In all cases oil accumulations in evaporators will affect their performance. Such performance in the case of refrigerants in which oil is not miscible will have effective surface reduced; and in the case where oil is miscible in the refrigerant, will raise the boiling temperature of the mixture.

5. When condenser performance indicates the presence of non-condensible gases, the system should be subjected to **purging.** In general, such non-condensible gases are air and can be removed by properly designed purging devices with which many plants are equipped.

In general, purging should be done at the proper point for economical and rapid elimination of the non-condensible gases; this point is not the same for all refrigerants.

When the air has a greater density than the refrigerant gas in the high side of a system, as is the case with ammonia, there is a definite tendency for the heavier air to be forced to the lowest point above the liquid level in the receiver when the plant is in operation; hence purging should be done just above the liquid level.

When the air has a lesser density than the refrigerant gas in the high side, as is the case with Freons, such air tends to concentrate in the top of condenser and purging should be done at this point.

Purging devices, generally designed to bleed small quantities of gas from the proper place in a system, are equipped with cooling means to condense out as much refrigerant as the suction pressure of the system will accomplish; the remaining gas, which is a high percentage non-condensible, is discharged to the atmosphere.

If no devices are installed, hand purging can be accomplished by attaching an empty refrigerant drum to the purge point with a quarter-inch line, bleeding into this drum until flow ceases, and then condensing as much refrigerant as condensate can be charged back into the system through a sight glass and then discharging the gases to the atmosphere. This process should be repeated until the liquid temperature of the system indicates the removal of non-condensible gases.

PART V

DOMESTIC AND COMMERCIAL SYSTEMS

Associate Editor, John W. Craig, General Works Manager,
The Crosley Corporation, Richmond, Ind.

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29. HOUSEHOLD REFRIGERATORS AND FREEZERS

THIS chapter covers all types of home refrigerators and freezers—ice-cooled electrically driven mechanical systems, and heat-operated equipment.

REFRIGERATORS

Sizes and Components

1. Table 1 shows the estimated **annual sales** and the average **retail prices** of all refrigerators produced in the United States. Refrigerator sales are highly seasonal, with as little as 3% of the annual sales falling in October or November and as much as 16% in April.[3] Thus a factory must have large storage facilities or operate under widely varying load conditions. The storage problem is further increased by a number of sizes, and quality variations to appeal to various consumer tastes. Table 2 breaks down the 1947 sales of 11 National Electrical Manufacturers Association companies.

2. A **household refrigerator** consists of an insulated cabinet finished in vitreous enamel or synthetic baked enamel outside, and vitreous enamel inside. It is usually refrigerated by an evaporator located in the upper part of the food storage space, either in the middle or toward one side, though the main food compartment may be cooled entirely by coils on the walls. Liquid refrigerant is supplied to an expansion device from a condenser. If cooled by forced air it is ordinarily a finned-tube condenser, **Fig. 1.** If cooled by thermal convection it may be either a finned-tube type, **Fig. 2,** or a plate type, **Fig. 3.**

The **compressor** may be reciprocating, **Fig. 4,** or rotary, **Fig. 5.** Many compres-

Table 1. World Household Refrigerator Sales of U. S. Manufacturers

Year	Unit sales	Av retail price	Retail value
Up to 1920	10,000	$600	$ 6,000,000
1921	5,000	550	2,750,000
1922	12,000	525	6,300,000
1923	18,000	475	8,550,000
1924	30,000	450	13,500,000
1925	75,000	425	31,875,000
1926	210,000	390	81,900,000
1927	390,000	350	136,500,000
1928	560,000	334	187,040,000
1929	840,000	292	245,280,000
1930	850,000	275	233,750,000
1931	965,000	258	248,970,000
1932	840,000	195	163,800,000
1933	1,080,000	170	183,600,000
1934	1,390,000	172	239,080,000
1935	1,688,000	166	280,208,000
1936	2,180,000	164	357,520,000
1937	2,500,000	171	427,500,000
1938	1,410,000	172	242,520,000
1939	2,085,000	169	352,365,000
1940	2,720,000	154	418,880,000
1941	3,700,000	160	592,000,000
1942	520,000	—	—
1945	263,860	—	—
1946	2,100,000	207	434,700,000
1947	3,400,000	240	816,000,000

sors are equipped with suction and discharge mufflers, in the form of a series of chambers connected by small orifices as in **Fig. 14,** or other types. For regulating cabinet temperature usually a single **temperature control,** as in **Fig. 6,** starts and stops the motor in response to changes in box temperatures or evaporator temperature or both, though some refrigerators are equipped with two controls to give independent control of frozen-food storage and above-freezing storage temperatures. The motor, similarly, requires starting and protective equipment.

Trends in Design

3. The significant trends in household refrigeration over the last several years have been increased food storage space, reduced energy consumption, greater cooling capacity, lower cost, lower noise level, improved appearance, and larger and colder frozen-food storage compartments.

In the **cabinet,** there has been an abandonment of wood framing. The cabinet has been changed from a structural skeleton on which are hung exterior and interior coverings, to a design in which the exterior is a single fabricated steel structure, supporting the inner compartment and the mechanism. This construction is cheaper, better adapted to mass production, and thermally more efficient.

The handling of **door** and **breaker strips** has been changed in many cabinets from the design shown at the top of **Fig. 7** to the one at the bottom, where the door covers the whole front of the cabinet and

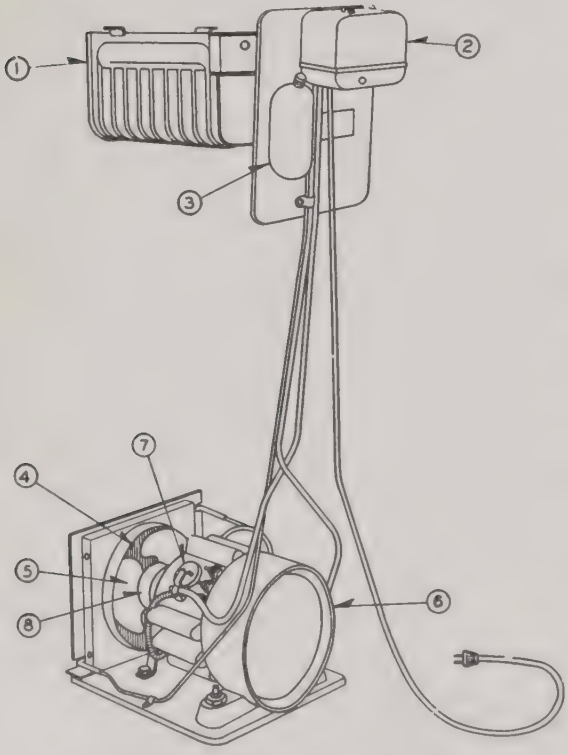


Fig. 1. Refrigerator Unit Assembly with Finned Condenser and Fan

1. Embossed and welded sheet metal evaporator
2. Control
3. Float valve
4. Finned-tube condenser
5. Fan
6. Hermetic motor-compressor unit
7. Temperature and current responsive motor protective device
8. Fan motor

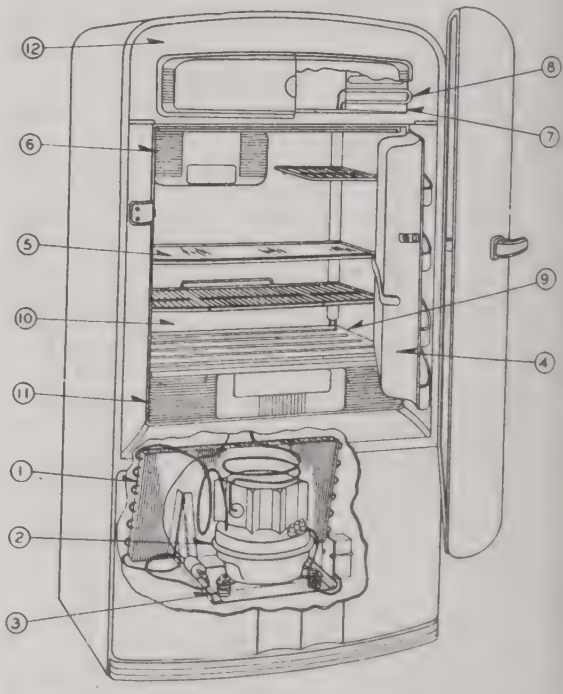


Fig. 2. Cabinet Assembly with High Humidity Compartment and Gravity-Cooled Finned-Tube Condenser

1. Large finned-tube condenser for gravity cooling
2. Hermetic motor compressor unit
3. Compressor mounting springs
4. Hinged inner door assembly carrying narrow shelves
5. Glass shelf separating upper low-humidity and lower high-humidity compartments
6. Meat storage drawer
7. Evaporator
8. Secondary condenser fastened to side of primary evaporator
9. Secondary evaporator for cooling high-humidity compartment
10. High-humidity compartment
11. Vegetable storage drawer
12. Breaker strips

Table 2. Household Refrigerator Sales Statistics for National Electrical Manufacturers Association Companies, for Year 1947[2]

Sizes	Domestic (48 States and D.C.)	Canadian	Other Foreign	Total
1. Less than 4 cu ft	2,729	—	12	2,741
2. 4 cu ft	18,534	1	4,923	23,458
3. 5 cu ft	—	—	—	—
4. 6 cu ft	331,393	854	21,356	353,603
5. 7 cu ft	1,608,202	11,861	112,817	1,732,880
6. 8 cu ft	453,052	958	18,172	472,182
7. 9 and 10 cu ft	565,695	2,470	26,028	594,193
8. 11 and 12 cu ft	10,758	—	—	10,758
9. 13 cu ft and up	3,934	—	—	3,934
10. TOTAL	2,994,297	16,144	183,308	3,193,749

the plane of the breaker strips is parallel to the back wall. A single plastic sheet is used for the interior surface of the door, as well as for the door breaker strip. This design gives greater accessibility and a more pleasing appearance. Some cabinets provide shelves or containers fastened to the inside of the door, or on a separate door, as in Fig. 2.

4. The trend toward specialized **interior appointments** has been marked. **Meat storage** compartments maintained at temperatures slightly above freezing with humidities of 90% are common. **High-humidity** compartments provide a space into which uncovered dishes can be placed without excessive drying. The increased use of **frozen foods** has necessitated special storage compartments, as large as 2 cu ft, maintained well below freezing. Temperatures of zero or lower are claimed for many models. A heated compartment for maintaining butter at a temperature convenient for spreading is in use.

Bacteria and mold increase as food is maintained in a more moist condition. Thus, while high humidity retards drying, it tends to hasten mold and bacterial deterioration, necessitating temperatures close to freezing for maximum storage time. To supplement the effect of low temperature, many commercial and domestic refrigerators are being equipped with **ultra-violet lamps** that give off a major part of their radiation in the bactericidal region of the ultra-violet spectrum, and in addition produce small quantities of ozone. These lamps are also effective in reducing cabinet odor level and odor transmission from strong to sensitive foods. Activated charcoal odor-adsorbers are also used in domestic cabinets. These adsorbers can be reactivated at a temperature of from 375 to 475 F every few months and replaced once a year.

5. The early mechanical refrigerators were ice-boxes with an evaporator substituted for the ice and with the condensing unit in a remote location. With better design, the units became quieter and smaller in size and were usually placed in the bottom of the cabinet under the food storage space, Figs. 1 and 2.

6. Almost all present-day domestic units are of **hermetic** design, Figs. 1 and 4, in which the motor and compressor are sealed into a gas-tight housing eliminating the shaft seal, belt, pulleys, and open motor

Table 3. Refrigerator Sales, by Size of Box

(1934–1947)

Year	6 Cu Ft & Under	7 Cu Ft & Over
	%	%
1934	78.1	21.9
1935	83.5	16.5
1936	85.2	14.8
1937	83.9	16.1
1938	84.4	15.6
1939	90.3	9.7
1940	90.2	9.8
1941	80.1	19.9
1946	9.3	90.7
1947 (9 Mos.)	11.4	88.6

bearings, all sources of noise and trouble. A motor compressor unit designed as a single integral assembly can usually be built more cheaply than one made up of two elements, providing there is a sufficient quantity for mass production. On 50 to 60-cycle, 4-pole motors are used, and on 25-cycle, 2-pole motors are used, driving the compressor at motor speed. The **pump** is either **reciprocating** or of positive displacement **rotary** design. If reciprocating, the compressors are either of the automotive type, with a wrist pin in the piston, or a Scotch yoke type, **Fig. 9.** The advantage of the latter type is the flexibility between the axis of the piston and the motor axis. With the automotive type it is necessary that the wrist pin be exactly at right angles to the piston axis, otherwise noise will be introduced due to misalignment of parts. When parts are not built with sufficient rigidity, misalignment under load may occur, even though no-load accuracy exists. This condition does not exist in the Scotch yoke design because it has flexibility in two directions.

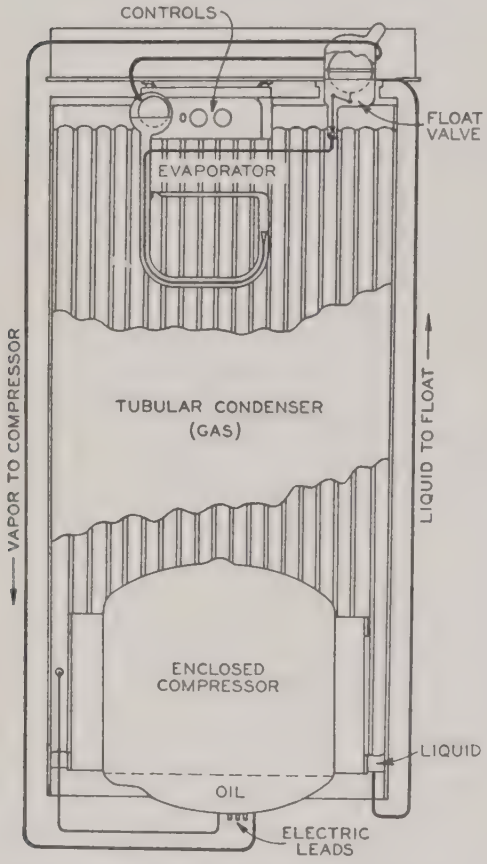


Fig. 3. System with Plate-Type Condenser

Rotary type pumps, Fig. 5, are generally of two types. In one type the integral rotor and shaft assembly is concentric with the circumference of the rotor and eccentric with the circumference of the cylinder. Rectangular blades are set into slots in the rotor which bear out against the circumference of the cylinder by means of gas pressure, springs, or centrifugal force, and as they move around the circumference the gas trapped between them is compressed.

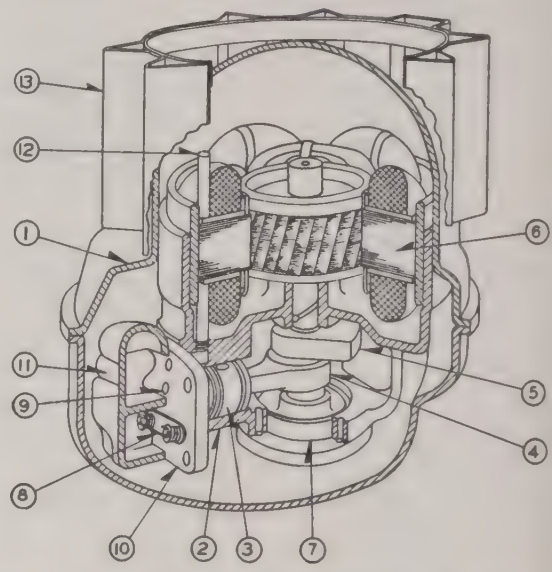


Fig. 4. Hermetic Motor Condenser Unit with Reciprocating Pump

1. Hermetic housing
2. Cylinder
3. Piston
4. Connecting rod
5. Counter balance
6. Motor
7. Outer bearing
8. Discharge valve assembly
9. Suction valve port
10. Valve plate
11. Cylinder head
12. Suction gas intake pipe
13. Cooling fins

The other type employs a shaft and an enlarged eccentric assembly that is concentric with the cylinder wall. On the eccentric is a cylindrical rotor and of such size that it operates at one point close to the cylinder wall. A blade operates in a slot in the cylinder wall and is held in contact with the rotor by means of springs or gas pressure to provide a seal between the high and low-pressure sides. As the eccentric rotor revolves, gas is drawn in just after it passes the blade and is then compressed and discharged near the other side

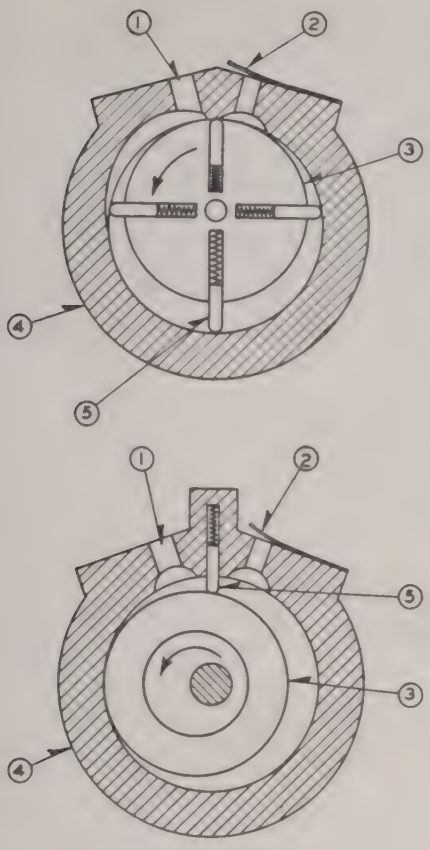


Fig. 5. Rotary Pumps

1. Intake port
2. Discharge port and valve
3. Rotor
4. Cylinder
5. Blade

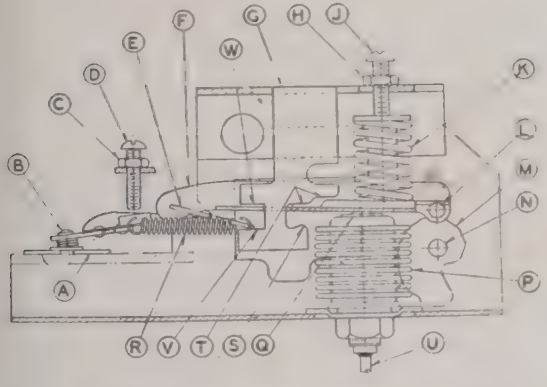


Fig. 6. Cabinet Temperature Control

A. Stationary contacts
B. Contact arm
C. Adjusting screw lock nut
D. Contact adjusting screw
E. Toggle wire
F. Rocker arm
G. Frame
H. Temperature control lock nut
J. Temperature control screw
K. Temperature control spring
L. Rocker arm shaft
M. Control cam
N. Cam shaft
P. Bellows
Q. Pivot point
R. Toggle spring
S. Lower rocker arm stop
T. Upper rocker arm stop
U. Bellows tube
V. Lower toggle wire stop
W. Upper toggle wire stop

of the blade. A modification of this latter design employs a blade fastened to this rotor that operates in a slot in the cylinder wall between two rockers.

The **valves** are ordinarily metal discs or reeds bearing on a valve seat **(Fig. 10)**. As valves are a prolific source of noise and inefficiency, much ingenuity has gone into their design.

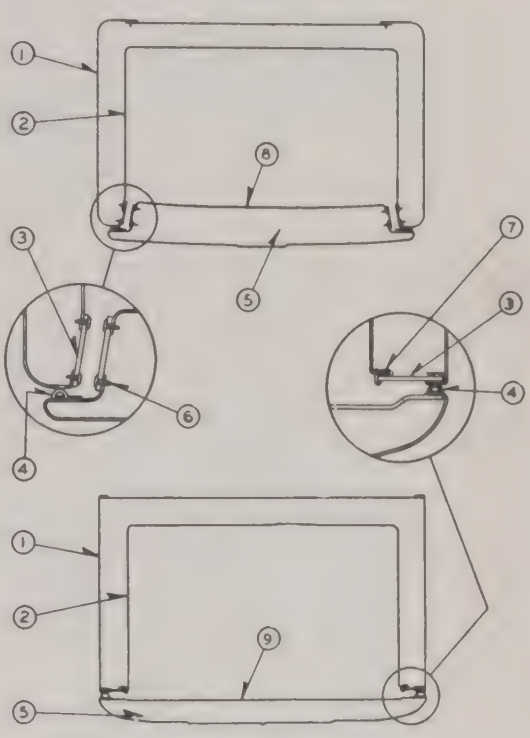


Fig. 7. Typical Cabinet Sections

1. Outer shell
2. Inner liner
3. Breaker strip
4. Door gasket
5. Door
6. Sheet metal screws
7. Stainless steel breaker strip retainer
8. Steel inner door pan
9. Plastic inner door pan

The **motors** in the hermetic machines are of the squirrel-cage induction type, Figs. 4 and 9. They may be either a straight split-phase design, **Fig. 11,** or they may employ a capacitor in the auxiliary winding for starting, running or both. For connecting the auxiliary winding into its starting position an external magnetic or thermal switch is used **(Fig. 12),** commonly actuated by the variation in current through the main winding.

7. With the elimination of external moving parts in the compressor, there is no longer a convenient motor shaft for driving the fan. This necessitates the use of a sepa-

rate **fan motor** (Fig. 1), if it is desired to use forced air circulation. This motor may be single-phase or it may be a two-phase motor receiving its power from the compressor motor operating as a phase converter. Many machines employ **plate-type condensers** of large area mounted on the back of the cabinet. Two common methods of building such a condenser are employed. In one, two steel sheets are spot and roller-welded together to provide refrigerant passages between the sheets (Fig. 3). The other employs a serpentine steel tube, copper-brazed or resistance welded to a single steel sheet.

A plate-type condenser performs differently than a finned coil with forced draft, in two particulars: (1) a portion of the heat is dissipated by radiation rather than by convection; and (2), due to the larger mass of the condenser, there is an effective storage of heat during the running cycle and disssipation of heat during the off cycle. Modifications of this system are in use, such as a large, finned-tube condenser, Fig. 2, which relies for air movement upon thermal convection through a flue on the back of the cabinet.

The trend toward simplicity has led to wide use of **capillary tubes** or similar restrictors **(Fig. 13).** While some capacity is sacrificed, under certain conditions, by the use of the capillary tubes, the trouble-free operation resulting therefrom is believed to more than offset this handicap.

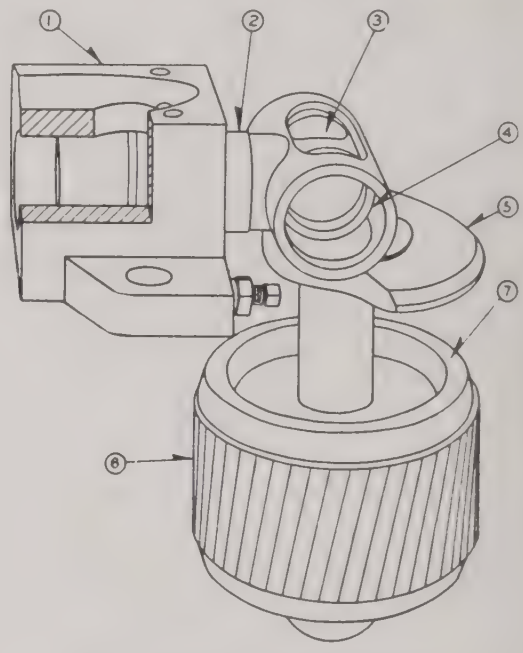


Fig. 9. Scotch Yoke Compressor

1. Cylinder
2. Piston
3. Crank pin
4. Cross slide
5. Counter balance
6. Rotor punchings
7. Die cast conductors

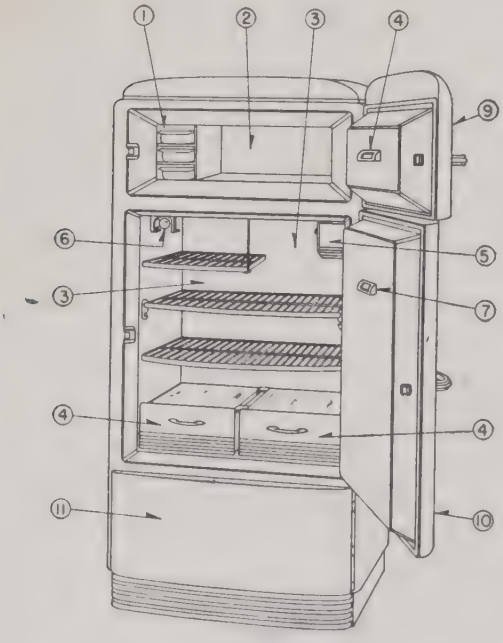


Fig. 8. Cabinet with Top Frozen-Food and Ice-Tray Compartment, Lower High-Humidity Compartment and a Separate Door Over Each

1. Ice-tray compartment
2. Frozen-food compartment
3. High-humidity compartment (wall cooled)
4. Vegetable storage drawers
5. Butter storage compartment (heated)
6. Ultra violet lamp
7. Thermometers
8. Bottle storage space
9. Door—frozen-food compartment
10. Door—high-humidity compartment
11. Compressor compartment

An additional advantage is that during the off period, the pressure can equalize between the high and low side, thereby permitting the use of a relatively low starting torque motor without an unloading device.

A marked trend in the domestic refrigerating field has been the swing to chlorinated hydrocarbon types of refrigerant of the **Freon** series. In fact, they are now the only refrigerant being used in domestic refrigerator production.

8. Over the last few years there has been a general increase in size of **evaporators** caused by the desire to increase the ice storage and the frozen-food storage capacity. In some cases, this has resulted

in the necessity for baffling about the evaporator.

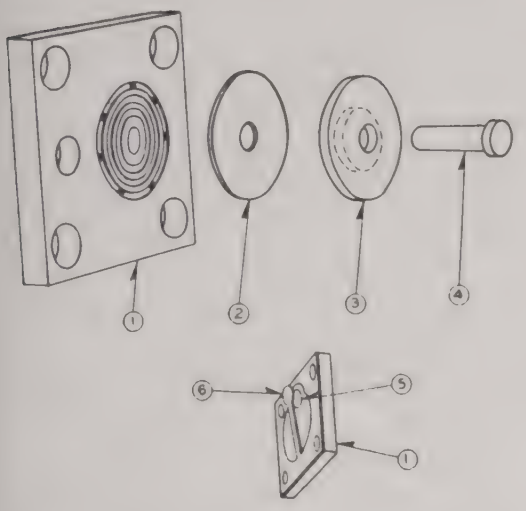


Fig. 10. Suction and Discharge Valve Assembly

1. Valve plate
2. Suction valve disc
3. Suction valve retainer and stop
4. Rivet
5. Discharge port
6. Discharge valve

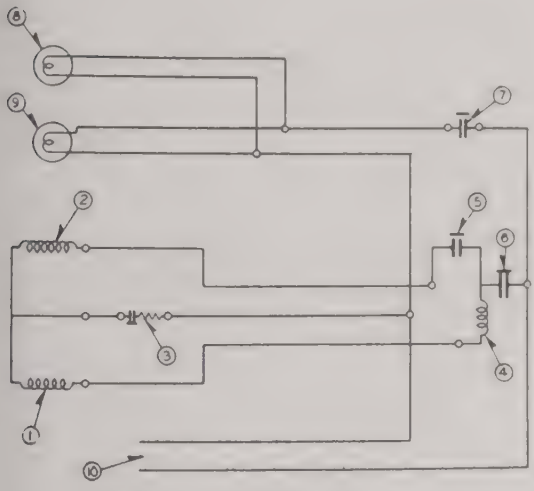


Fig. 11. Unit Wiring Diagram

1. Main winding of split-phase motor
2. Starting winding of split-phase motor
3. Motor protective device
4. Starting relay coil
5. Starting relay contacts
6. Thermostat contacts
7. Door-operated light switch
8. Cabinet light
9. Dial light
10. Power source

A widely used evaporator is formed of two sheets of metal (Figs. 1 and 13), one or both of which may be embossed with a design, resistance-welded or furnace-brazed together to form refrigerant passages. Stainless steel, brass or bronze with an electro-tin plate finish, ordinary carbon steel with vitreous enamel or a multilayer metallic plated finish is employed.

Each year **capacity** and **efficiency** have steadily increased and noise has decreased. These trends have been caused largely by competitive considerations, stimulated by public demand.

Design of Cabinet

The fundamental factors in the design of a cabinet are its **capacity** in cubic feet and its **dimensions**, both of which are influenced by competitive consideration. The method of figuring useful cubical contents and useful shelf area is contained in a set of standards put out by the ASA publication 1338.1–1944.

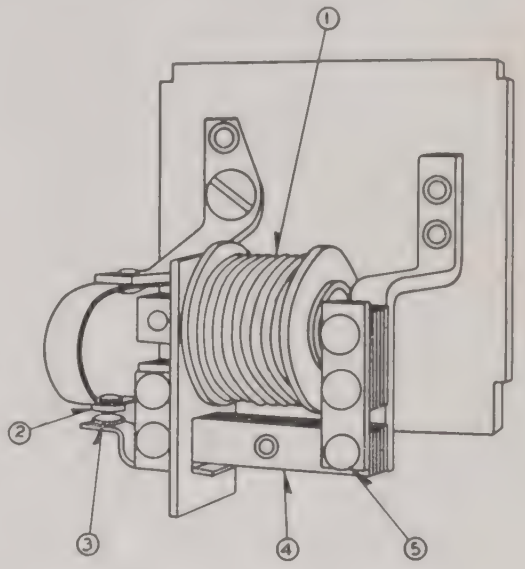


Fig. 12. Starting Relay

1. Coil
2. Stationary contact
3. Movable contact
4. Armature
5. Pivot

9. Insulation in a domestic cabinet varies from 2 to 3½ in. in thickness, the greater thickness ordinarily being used with the large size cabinets. Insulating materials in common use are rock wool, glass wool, balsam wool, corrugated paper, etc. Although it is desirable to obtain as nearly a vapor-tight exterior cabinet shell as possible, consideration should be given to the fact that when water vapor is migrating from the ambient air into the insul-

ation it is also passing from the insulation into the area around the cold evaporator. Consequently, the design of door gaskets, breaker strips and other parts should provide maximum barriers for vapor flow from the room air to the insulation, but as free passage as is practicable between the insulation and the inside of the cabinet.

The effect of insulation thickness on heat leakage in itself is not all-important. A more important consideration is main-

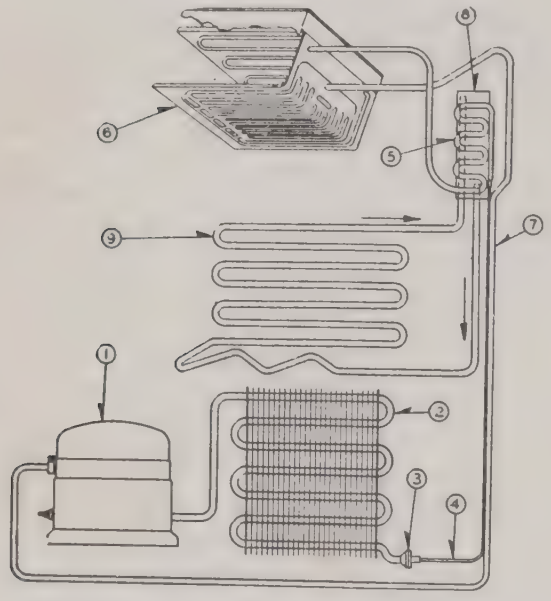


Fig. 13. Refrigerant Circuit of Machine with Secondary Cooling System

1. Compressor
2. Condenser
3. Filter
4. Capillary tube restrictor
5. Cooling coil clamped to secondary condenser
6. Evaporator
7. Suction line in heat exchange relation with capillary tube
8. Tank-type secondary condenser
9. Secondary evaporator

taining the surface of the cabinet at a sufficiently high temperature to prevent condensation. Areas where condensation usually occurs will be around the hardware or along the breaker strip. Good practice suggests that with a 40 F cabinet in a 80 F room, no external surface temperature should be more than 7 deg below the room temperature.

10. Mechanically, the overall **structure of the cabinet** should be strong enough to stand shipping. A cabinet that will ship is almost certain to be strong enough to stand daily usage. The inner food liner should be designed so that it does not chip or craze at points of support. Shelves must be designed with adequate strength so that they will not sag under the heaviest usage load. The hardware should be satisfactory for at least 50 door-openings per day for a period of perhaps 15 years.

11. In the early days, **cabinet finishes** presented one of the most serious problems because they deteriorated rapidly. However, modern synthetic baked enamel finishes give an excellent account of themselves. Customarily, they are applied over a chemically treated surface, which gives better adhesion to the paint and prevents the spreading of rust if the finish is scratched through. In addition to the synthetic enamel finishes, there are improved vitreous enamel finishes that usually consist of one ground coat and one or two finish coats, each of which is fused separately. The cabinet liners are usually finished in white vitreous enamel.

The industry has gotten away from selling refrigerators on the basis of mechanical details. The user assumes that the mechanism will operate satisfactorily, and emphasis is now placed on appearance and conveniences, such as meat storage drawers, vegetable compartments, removable or sliding shelves, ornamental evaporator and drawer fronts, butter storage spaces, mechanical means for releasing ice trays and removing cubes, glass shelves, high-humidity compartments, ultra-violet radiation, etc.

12. High-humidity compartments are used to retard evaporation of moisture from uncovered foods. In a conventional refrigerator the humidity in the food storage compartment is low, due to the presence in the cabinet of the cold evaporator surface, and rapid loss of moisture from uncovered food results. The drying effect can be greatly reduced by cooling the food storage compartment by means of a large area of cooling surface kept at a temperature just a few degrees below the cabinet temperature.

These high-humidity compartments may be as small as a single drawer for the storage of meat or vegetables or they may include the whole above-freezing space of the cabinet. The large compartment may be

designed for only moderately high humidities for the storage of left-overs, and drawers may be provided in addition for the very high humidities needed for the storage of leafy vegetables (**Fig. 8**). If the small, drawer-size compartment is intended for the storage of meat it will be placed, if possible, just below the evaporator in order to get temperatures at or near 32 F. If the drawer is intended for vegetable storage, it may be placed anywhere in the cabinet, usually at the bottom.

The humidity in a closed drawer will be high because the cooling surface, which is the whole external surface of the drawer, operates at practically the same temperature as the air in the drawer. This same principle of large, air-cooled, external surfaces is extended to larger size compartments by providing an enclosed compartment two or three shelves high, with space between the food liner and the outside of the compartment for the circulation of cold air from the evaporator.

Most commonly, however, a large, high-humidity compartment is cooled by providing cooling coils on the outside of the liner. The coils are usually the evaporating portion of a closed, secondary refrigerant circuit, the condensing portion of which is either tubes in contact with the primary evaporator, or tubes or a tank cooled by a separate primary loop buried in the insulation. This latter arrangement is shown in Fig. 13. The secondary circuit is charged with a volatile fluid, usually the same material as in the primary circuit. As the liquid refrigerant in the tubes on the food liner evaporates, it passes upward to the secondary condenser where it condenses and then returns to the lower tubes to repeat the cycle. The secondary condenser section is removable, coupled to the main cooling surface so that when the main unit is removed the secondary circuit can remain in the cabinet.

This system has two main advantages over one in which part of the primary circuit is used for cooling the high-humidity compartment. These are: (1) in case of unit failure there is no need to remove the high-humidity cooling circuit; (2) the same unit can be used in refrigerators that do or do not employ high-humidity sections. High-humidity compartments have also been cooled by large areas of finned coils, although this design is not common now.

High-humidity compartments may have their own inner doors or they may be sealed by the single main cabinet door. In the latter case, the partition separating the low from the high-humidity areas, usually seals against the door with a rubber gasket. A third arrangement employs two outside doors, one closing the low-temperature compartment and a larger one covering the high-humidity main-storage area.

In the winter time, the low ambient humidity makes it difficult to maintain a high humidity in the compartment unless the compartment is well sealed and large, comparatively warm, cooling areas are provided. However, if these precautions are taken, excessive interior wetness may result in the summer time when the ambient dew point is high. Each time the cabinet door is opened, moisture is deposited on all of the walls, shelves and dishes in the refrigerator. As the design of the box is such as to reduce evaporation from food, evaporation from all the other surfaces is also very slow and moisture collects due to repeated door openings until drippage may result.

13. While the bactericidal effects of **ultraviolet** and **ozone** have been known for a

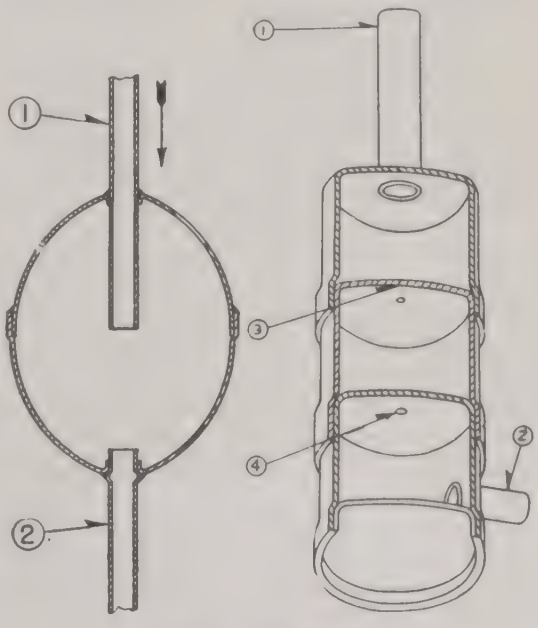


Fig. 14. Mufflers

1. Intake
2. Outlet
3. Baffles
4. Ports

long time, it is only recently that suitable lamps for use in domestic refrigerators have been available. A lamp must have sufficient ultra-violet power and deliver a controlled amount of ozone, yet give out very little heat. Lamps are available that radiate the major part of their energy at the most effective bactericidal wave length, 2537 Angstrom units. These lamps are most needed in a humidity above 60%, since at lower humidity food will dry out so rapidly that no mold or slime will occur, anyway. As the germicidal effect which occurs in the shade is produced largely by ozone from the lamp, the lamp should be placed so that any circulation will tend to carry the ozone throughout the food storage space. Fig. 8 shows an ultra-violet lamp mounted just above the door opening in a high-humidity cabinet.

Design of Mechanical Unit

14. As mentioned previously, practically all household refrigerators utilize hermetic units. Therefore the following discussion is based on this type of unit which uses either **rotary** or **reciprocating compressors.** Two refrigerants are used in present production of hermetic machines, **Freon-12** (dichlorodifluoromethane), and **Freon-114** (dichlorotetrafluoroethane). Generally speaking, Freon-12 is used in both rotary and reciprocating compressors, and Freon-114 in rotary compressors.

15. Compressor **motors** used are all squirrel-cage induction motors with either built-up copper or die-cast aluminum rotor conductors (Fig. 9). The wires are insulated with two or three layers of impregnated cotton or enamel. If enamel is used, it is imperative that it be not soluble in the lubricating oil or refrigerant used. All of the other insulating materials, such as slot insulation, must be of such a nature that they are unharmed by lubricating oil or refrigerant. It is also important that these fluids do not dissolve any substance out of the insulation that will cause trouble in the system.

When applying a motor to a domestic refrigerator compressor, design changes can be made in the associated equipment to give desirable motor loading. In this respect the situation is different from run-of-mine motor applications. There are three points that must be checked in the application of a motor to a pump. First, the motor should be capable of carrying the maximum possible load that may be imposed steadily on the machine without harmful overheating. This load might be obtained by placing the machine in a 110 F room with the cabinet door open. Second, the motor should have sufficient maximum torque to carry the heaviest momentary load at the lowest probable voltage. Such a load might be achieved by placing the machine in a 110 F room with the cabinet door open to allow all parts to stabilize at 110 F, and then with the door closed, operating the unit to pull the cabinet temperature down. While the maximum load under this condition may be heavier than the one previously referred to, the temperature rise of the motor need not be considered because the low thermal capacity of a modern evaporator permits the machine to pull through this maximum load point quickly. With the voltage throughout most of the country ranging between 115 and 120, a machine which will pull through this maximum point at a voltage between 100 and 105 volts is probably safe.

The third requirement for the motor is that it have sufficient starting torque to start the compressor at 100 volts satisfactorily under any condition likely to be encountered.

16. In designs using a capillary type of restrictor and not employing an unloader, the **starting torque** is tied in with the degree to which the pressure between the high side and low side has an opportunity to equalize during shutdown. Necessary starting torque is thus a function of control differential as well as capillary restrictor design. If the motor protection is an automatically reclosing device, it is not serious if the motor occasionally fails to start due to insufficient equalization. If the protective device is not automatically resetting, it is imperative that the motor be able to start the load under all operating conditions. If the machine is not equipped with the type of restrictor that equalizes the pressure during a shutdown, then the motor must have enough torque to start the pump with maximum pressure differential across the piston, or an unloader, the design of which is based on the motor char-

acteristics, must be used to reduce the starting load.

17. The two classes of **condensers** are those of large mass and dimensions, cooled by radiation and thermal-convection air circulation, and the smaller type with forced-air circulation. Due to the limited space available for a plate condenser or a thermal-convection, cooled, finned condenser, there is a limit to the steady heat-transfer rate that can be achieved. This means that when heavy loads are imposed steadily upon the condenser, the discharge pressure will run high, and the pump design and motor torque will have to be adequate. However, under normal conditions when the percentage of running time is low, the high thermal storage of this type of condenser may produce lower average discharge pressures than a small forced-air circulating type. Further advantages to the large condenser are that it is not necessary to employ a separate motor to drive the fan and it gives practically no trouble from the standpoint of cleaning.

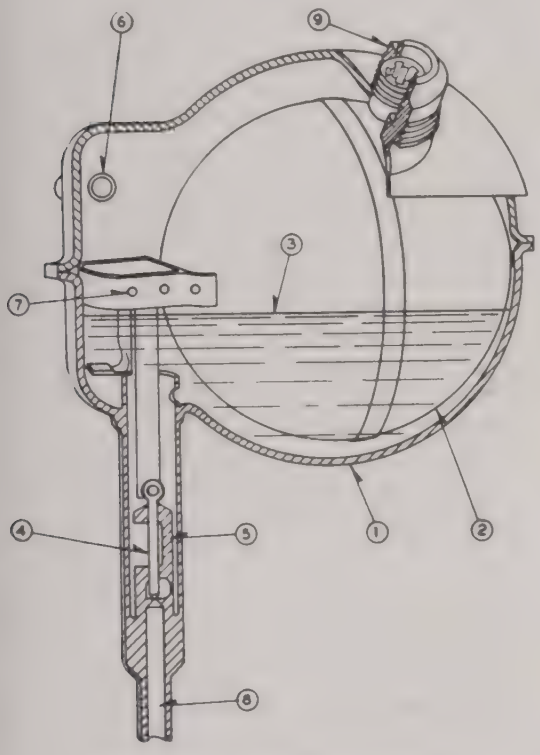


Fig. 15. High-Side Float Valve

1. Housing
2. Float ball
3. Liquid level
4. Valve needle
5. Valve needle guide
6. Inlet
7. Pivots
8. Outlet
9. Purging port

The forced-air design, on the other hand, has the decided advantage of light weight, compact design, and more satisfactory carrying of prolonged heavy loads.

18. Two types of **expansion devices (Fig. 15)** are now used in domestic refrigeration, these are the **fixed-orifice** type consisting of a long tube or an equivalent restriction, and a **high-side float valve** that allows all liquid refrigerant discharging out of the condenser to go to the evaporator except for the small quantity necessary to fill the float chamber.

The fixed restrictor has the advantage of simplicity and no moving parts. One disadvantage is that it has a smaller orifice than that used with the float valve and consequently more likelihood of plugging. However, a domestic refrigerating machine must be kept scrupulously clean inside. If this is done and no water, wax or other materials that harden at low temperatures are present, a fixed restrictor tube will not be troubled with plugging. Another disadvantage is loss of capacity under certain operating conditions. When the machine stops, some compressed gas passes into the evaporator, causing a loss in capacity. Also, even with the machine running, there is only a certain set of suction pressures and discharge pressures under which any particular capillary tube will not pass any gas or will not permit liquid to back up into the condenser. A capillary tube expansion device lends itself well to being placed in a heat-exchange relation with suction line, and a considerable gain in capacity is realized when using a Freon refrigerant.

A high-side float is more difficult to equip with a heat exchanger, because the heat exchanger must be in contact with only condensed refrigerant to be effective.

In spite of its lower efficiency, the fixed restrictor has almost entirely supplanted the float.

19. Modern **evaporators** are designed so that the amount of refrigerant contained in them is generally less than 1 lb. This is an advantage in lower cost. Evaporators are usually shaped in the form of a plate, U, or box, with refrigerated shelves within the U or box. The refrigerant path is usually from the restrictor, through the

shelves, if any, through the evaporator body and into a header of some type.

The desire of users to have more ice and frozen-food storage capacity causes the evaporators to be as large as practicable. However, for a fixed cabinet temperature, the larger the evaporator becomes the higher the temperature at which it operates and thus the greater the tendency for defrosting at low room temperatures. This will result in a large collection of ice on the bottom of the evaporator and will prevent satisfactory removal of the ice trays. This elevated temperature is undesirable from the standpoint of frozen-food keeping, and if a large evaporator and low temperatures are both necessary, baffles, an insulated shelf, or other means must be provided to reduce the heat transfer from the evaporator to the main storage compartment. If the evaporator is to be used with some type of secondary system for cooling a high-humidity cabinet, consideration must be given to suitable surfaces for contacting the secondary condenser.

Ingenious ways of putting the evaporator into the cabinet without breaking refrigerant lines have been developed. Evaporators are commonly placed through openings in the back or top, or the refrigerant lines may be run under the breaker strip and across the top. Whatever design is employed, care must be exercised to prevent external cold spots which cause sweating and to seal the outside shell of the cabinet against air infiltration.

20. Controls are usually operated by a change in evaporator temperature, or in response to both evaporator and cabinet temperatures. In the former method, the box temperature tends to fluctuate with changes in ambient temperature. This fluctuation is reduced in the latter method. The latter system has advantages from the standpoint of food preservation, while the former is better adapted to ice-freezing and keeping of frozen food temperatures constant under various ambient temperatures for any given control setting. Controls, ordinarily operated by a volatile liquid charged bellows system, are usually in the cabinet above the evaporator or in the machine compartment. Systems employing auxiliary valves in the secondary systems or automatically controlled dampers in the cold-air stream allow independent control of frozen food and box temperatures.

As it is not practical to use a centrifugal switch for connecting the auxiliary motor winding to its starting position in a hermetically sealed machine, it is necessary to provide some type of external starting relay. This is normally a magnetic or thermal contact-making device (Figs. 11 and 12), which the high starting current through the main motor winding moves into starting position. Then, as the motor speeds up and the current drops off, the relay moves into the running position. While, at constant voltage, a motor starting under no load has a current range of about 5 to 1 between the locked rotor current and the running current, it is necessary for these relays to operate between narrow current limits, in order to make contact under reduced voltage starting and to break contact after starting under heavy loads and high voltage.

Some type of **motor protection** is ordinarily provided. The protective device usually possesses thermal lag that allows the motor to carry brief heavy overloads without tripping out. There are two general types of overload devices, one that is mounted on the compressor and is responsive to motor-compressor unit temperature and motor current, and the other that functions only on motor current. The former arrangement has the advantage of being more responsive to the actual temperature of the machine. The protective device may be manual or automatic resetting.

21. Thorough **drying** of the system is essential. An improperly dried system will be troubled with ice collection in the restrictors, and corrosion troubles. Drying is usually done by heating the unit to a temperature above 212 F for several hours and removing the water vapor by evacuating the unit or passing dry air through it. Care must be taken to remove all air before charging.

Oil and refrigerant are measured by either volume or weight methods. Modern machines with their extremely low charges must be charged with considerable accuracy. **Charging methods** that involve measuring of a certain volume by means

of the lowering of the level in a sight glass or the use of a flow meter are accurate, but the charge cannot be checked in case of doubt.

A system that provides for complete removal of the air remaining from previous processing and permits accurate charging is described by McCloy and Haley.[16] In this system, the refrigerant is placed in a small container or bomb. This bomb can be weighed, during the charging operation, on scales that are sensitive to 1/64 oz. The bomb, equipped with a valve, is attached on the high side of the system with the valve closed. If the machine is full of air due to previous processing, it is charged with oil, then allowed to run and evacuate itself through a vent on the high side. This system results in some air being left in the machine, particularly with a capillary-tube type of expansion device. However, after the vent is closed and the valve on the bomb is opened, the flow of condensed refrigerant out of the bomb permits the collection of non-condensible gas up into the bomb where it is removed with the bomb. Oil and refrigerant of high purity are essential.

A description of heat-operated refrigerators appears in Chap. 3 and on page 504.

22. All points previously discussed deal primarily with the producing of a sufficient quantity of refrigeration to give satisfactory results in the field. The machine must be able to take care of the **heat leakage load** into the cabinet under high room temperature, 110 F being the usual value. It must also be able to take care of a reasonable number of **door-openings,** possibly 50 per day with an average length of ten seconds. It must be able to absorb the **latent heat of condensation** and fusion of the moisture that deposits on the evaporator. It must be able to **cool the product** and it must **freeze ice.**

Many attempts have been made to work out an artificial load cycle, but the overall capacity of the unit is largely dictated by competitive standards, and, as time changes, standards change. The specific capacity relationship between the various parts of the assembly usually depends on whether the manufacturer provides one size compressor for all size cabinets, whether different units are used in the South and North, and upon the effect of accessory equipment such as high-humidity compartments.

23. While the design of a refrigerator will have as its ultimate test the satisfaction of the user and the meeting of competitive standards, it must be remembered that refrigerators for sale to certain groups, such as government agencies, will have to meet specific **codes** that are set up by these agencies. In addition, there are certain **standard methods** for calculating gross and net food storage areas and usable shelf areas that are universally adhered to in advertising. The most important follow:

1. Minimum construction requirements of ice refrigerators, National Association of Ice Refrigerator Manufacturers, 1936.

These provisions represent minimum standards for substantial and durable construction to secure satisfactory and economical operation, according to the National Association of Ice Refrigerator Manufacturers.

1. In case of **exteriors over framework** tops should preferably be die stamped in one piece or well formed and brazed or welded. Tops of 6 sq ft area or less, 24 gage iron or steel; those over 6 sq ft 22 gage, or of equally rigid construction.

Ends of 6 sq ft area or less, 26 gage; over 6 sq ft and less than 12 sq ft, 24 gage; and all over 12 sq ft, 22 gage (U. S. gage used here).

Backs and bottoms, 30 gage up to 7 sq ft area; 7 sq ft to 12 sq ft, 26 gage; and all over 12 sq ft, 24 gage.

Doors should preferably be die stamped of one piece or well formed and brazed or welded: Doors of less than 4 sq ft area, 26 gage; 4 to 7 sq ft, 24 gage; 7 sq ft to 12 sq ft, 22 gage; and those over 12 sq ft, 20 gage.

Legs and leg bases are to be of gage and design as to make a firm base for the cabinet. Legs shall be secured to base frame with 12 bolts or screws not less than $\frac{1}{4}$ in. in diameter. (Superficial surface area of the insulated portion of the refrigerator shall be determined by using the outside width, depth and height of the insulated portion of the cabinet alone.)

2. The main members of the **frame** shall be accurately machined to size and shape of suitable grade pine, spruce, or equivalent. Joints all to be securely made, either mortised, halved together and screwed or nailed with not less than two screws or nails per joint, or they may be dowelled and glued.

Door frames to be made of the same materials as the main frames and jointed in the same manner.

3. In case of **exteriors without framework** the provisions of clause 1, above, substantially apply, as well as the following:

The exterior **shell,** except top and front rails,

shall be securely welded or screwed into a complete unit with suitable top braces and gusset plates. If spot welding is used the spot welds shall be spaced at intervals not greater than 5 in. and shall show a securely welded spot of sufficient strength that the weld itself will break loose as one piece when two sheets are rotated in opposite directions in the same plane with the surface of the sheets so welded.

Bolts for securing legs or bases to the bottom of the shell, not less than 12 in number and not less than $\frac{1}{4}$ in. in diameter, shall be securely welded through the bottom before assembly.

4. Ice chambers: **Interior** lining of the walls and top of all ice chambers less than 6 cu ft, shall be 26 gage galvanized steel, securely fastened in place; ice chambers 6 cu ft and over in size, 24 gage.

All ice chamber bottoms, racks, and grids shall be securely riveted or welded so as to stand the maximum load to which they will be subjected without permanent deformation.

Ice racks are to be formed of a single sheet of corrugated galvanized steel of an area not exceeding 1.5 sq ft, 24 gage; over 1.5 sq ft but less than 5 sq ft, 22 gage; 5 sq ft and over, 20 gage.

Provision chambers under 7 cu ft shall be made with walls lined with 28 gage steel; and where porcelain bottoms are used the steel shall be securely hemmed over the upturned flanges of the porcelain bottoms. Porcelain to be two coats best grade commercial porcelain on 20 gage enameling stock; where provision chambers are lined with porcelain they shall be made of 20 gage enameling stock, punched, formed, and welded into one piece, finished with two coats of best grade commercial porcelain, one ground coat and one white finish coat.

Ice chamber door linings: Ice door linings shall be die stamped or welded into one piece pans of 26 gage on all doors having an area of less than 5 sq ft; pans over 5 sq ft, 24 gage.

Provision chamber doors shall be die stamped in one piece or well formed and brazed or welded of 26 gage steel on all doors having an area of less than 6 sq ft; 24 gage for all doors 6 sq ft and over.

5. All refrigerators having a superficial surface area of the **insulated** portion less than 47 sq ft shall be insulated with approved insulation having a heat leakage value of 0.17 Btu per sq ft hr deg F. Refrigerators having a superficial surface area of insulated portion 47 sq ft and less than 60 sq ft shall have a value of less than 0.14. Those having a superficial surface area over 60 sq ft shall have a value less than 0.11.

Insulation shall be cut accurately to fit the space in the wall for which intended, with a tolerance not exceeding $\frac{1}{8}$ in. in each dimension, and shall be so installed and sealed in place as to prevent infiltration of moisture.

Bureau of Standards thermal conductivity tests shall be considered adequate in determining efficiency of the insulation.

6. All **doors** shall be equipped with compressible gaskets contacting all around the doors so that when the door is closed and locked a new dollar bill cannot be shoved underneath the gasket from the outside at any point.

7. All exposed parts of **locks** and **hinges** and pivots shall be made of brass of a thickness not less than .050 in., plated with the best grade chromium. Hardware not relying on exterior shell for strength may have cover plates of lighter gage brass chrome plated.

8. Shelves of less than $2\frac{1}{2}$ sq ft area shall be made with rim wires and cross braces, where used, of a number 3 gage wire to which bars of No. 11 wire or ribbons of $\frac{1}{16}$ to 3.16 in. in section, or suitable expanded metal fabric, shall be welded. All shelves are to be heavily hot tinned after fabrication in a mixture of 70% tin, 30% lead.

9. Drainage systems shall be made durable and sanitary with a water seal trap according to best current practice.

10. Interior **finish** may be first-grade commercial porcelain enamel, one ground coat and two white coats. Exterior finish may be a ground coat of high baked synthetic applied evenly over the surface and adequately baked; and a finish coat of the same character of material adequately baked; or a coat of good commercial grade lacquer properly applied.[12]

The maximum allowable **ice meltage** per cubic foot of food space per degree temperature difference between the box and room temperature, specified by the National Association of Ice Industries, is as follows:

Food space, cu ft	*Ice meltage per day per cu ft of food space, per deg temp diff*
3.5	0.14 lb
4.0	0.137
5.0	0.129
6.0	0.121
7.0	0.109

American Standards Assn. publications B38.1—1944 and B38.2—1944 give the method of computing the gross volume, the net food storage volume and the shelf area of a refrigerator, as well as a test code for mechanically operated household refrigerators.

Method of Computing Food-Storage Volume and Shelf Area of Automatic Household Refrigerators from B38.1—1944

1. Scope and Purpose

1.1. The purpose of this standard is to provide a uniform method for determining the food-storage volume and shelf area of automatic household refrigerators.

2. Inside Dimensions

2.1. Inside Depth shall be the mean distance between the inside door pan and the rear liner.

2.2. Inside Width shall be the distance between the inner surfaces of the side wall of the liner.

2.3. Inside Height shall be the distance between the inner surface of the floor and the ceiling of the cabinet.

2.4. Depth of Cooling-Unit Space shall be the depth of the cabinet as defined in 2.1, unless there is a clear space for food storage of 4 inches or greater in front or back of the cooling unit.

In the latter cases, if the food-storage space is located in front of the cooling unit, the depth of the cooling-unit space shall be taken as the distance from the rear wall of the liner to the foremost part of the cooling unit; if the food-storage space is located behind the cooling unit, the depth of the cooling-unit space shall be the distance from the inner surface of the door to the rearmost portion of the cooling unit.

2.5. Width of Cooling-Unit Space shall be the outside width of the cooling unit itself (neglecting suction headers near the top of the cooling unit) or, if side baffles are used, the over-all width including the baffles. If there is less than a 4-inch space between the unit or baffle and the side wall of the liner, the distance from the nearest liner side wall to the outer surfaces of the cooling unit or baffle (whichever is the greater) shall be the width.

2.6. Height of Cooling-Unit Space shall be the distance from the ceiling of the cabinet to the bottom of the drip tray or lower baffle, whichever is the greater.

In refrigerators equipped with non-automatic-defrosting-type cooling units and having removable-type drip trays designed for food storage, the height of the cooling unit shall be the distance from the ceiling of the cabinet to the bottom of the cooling unit.

3. Food-Storage Volume Determination

3.1. Gross Volume shall be the product of the inside depth (2.1), inside width (2.2), and inside height (2.3).

3.2. Volume of the Cooling-Unit Space shall be the product of its depth (2.4), width (2.5), and height (2.6).

3.3. Frozen-Storage Volume shall be the sum of the volumes of the ice-freezing compartments and of the frozen-food storage compartments. The volumes of these compartments shall be the products of their inside dimensions.

3.4. General-Storage Volume shall be the gross volume (3.1) minus the volume of the cooling-unit space (3.2).

3.5. Total-Storage Volume shall be the sum of the general-storage volume (3.4) and the frozen-storage volume (3.3).

4. Shelf-Area Determination

4.1. Net Shelf Area shall be the sum of the areas of the shelves, bottoms of suspended containers, the bottom of the liner, and the shelf areas in the frozen-storage section, as defined in 4.2 to 4.6, inclusive.

4.2. Full Shelves. The area of full shelves and of the bottom of the liner shall be the product of the inside depth (2.1) and the inside width (2.2). The area of any part of the shelves having less than 4 inches clearance above shall not be included in the net shelf area, except as provided in 4.4 and 4.5.

4.3. Recessed Shelves. When any shelf is recessed, the area of the recess shall be deducted. The width and depth of the recess shall be the distance from the edges of the recess to the adjacent interior surfaces, as used in determining the inside depth (2.1) and inside width (2.2).

4.4. Fractional Shelves. In computing the area of fractional shelves, the width and depth of the shelves shall be the distance from the adjacent interior surfaces of the cabinet to the outer edges of the shelves.

Whenever the space occupied by the drip tray is included in the general-storage volume, the part of the shelf supporting the drip tray may be considered as food-shelf area; otherwise it shall not.

4.5. Low-Temperature Shelves. The areas of the shelves and compartment bottoms in the ice-freezing and frozen-food-storage compartments shall be included in the net shelf area, provided there is a clear space of not less than $1\frac{1}{2}$ inches above such shelves and compartment bottoms.

4.6. Suspended Containers. The area of the bottom of a suspended container and the area of the shelf beneath shall not both be counted unless the clearance between this shelf and the bottom of the container is at least 4 inches. If the mean depth of a suspended container, as measured between its bottom and the cover, or shelf immediately above, is less than 2 inches, the bottom area of said container shall not be added.

5. Data to Be Reported

5.1 Food-storage capacity reports of refrigerators shall include the following:

(*a*)	Total-storage volume (in cubic feet)	(See 3.5)
(*b*)	General-storage volume [(*a*) minus (*c*); in cubic feet]	(See 3.4)
(*c*)	Frozen-storage volume (in cubic inches)	(See 3.3)
(*d*)	Net shelf area (in square feet)	(See 4.1)

The volumes as specified in 5.1 (*a*) and (*b*) shall be reported to the nearest 0.1 cubic foot.

The volume as specified in 5.1 (*c*) shall be reported to the nearest cubic inch.

The net shelf area as specified in 5.1 (*d*) shall be reported to the nearest 0.1 square foot.

3. Method of Calculating Usable Food, and Ice Storage Space, Rated Ice Capacity and Shelf Area of Domestic Ice Refrigerators. A Standard of the National Association of Ice Refrigerator Manufacturers.

4. Safety Code for Mechanical Refrigeration, sponsored by The American Society of Refrigerating Engineers and approved by the American Standards Association, April 20, 1939 (ASRE Standard No. 15ASA-B-9-1939).

5. Standard Method of Rating and Testing Mechanical Condensing Units,

sponsored by The American Society of Refrigerating Engineers, developed by Joint Committee (Standard 14).

6. Underwriters' Laboratories Standard for Unit Refrigerating Systems, dated June 1937. This latter code is a standard that must be met in order to get Underwriters' approval, and in many municipalities Underwriters' approval is necessary before the refrigerator may be sold. The following sections from this code apply to the mechanical design of the equipment:

Strength: All part of the high-pressure side of the refrigerating system which are subject to pressure shall be designed and constructed with a factor of safety of at least 5, based on the maximum pressures obtained during the operation at 40 C (104 F).

All parts of the low-pressure side of the refrigerating system which are subjected to pressure shall be designed and constructed with a factor of safety of at least 5, based on the vapor pressure of the refrigerant used at 70 F. In no case shall the hydrostatic pressure be less than 100 psi.

All parts shall withstand pressures higher than the relief pressure generated under fire conditions.

Manufacturer's tests: The manufacturer shall test all pressure-containing parts of the system at pressures shown in the table which follows.

Minimum Pressures

(psig)

Refrigerant	*High side*	*Low side*
Ammonia	340	230
Dichlorodifluoromethane (Freon-12)	195	140
Dichlorotetrafluoroethane (Freon-114)	65	50
Ethyl chloride	50	50
Isobutane	110	70
Methyl chloride	175	120
Methyl formate	50	50
Methylene chloride (Carrene)	50	50
Monofluorotrichloromethane (Freon-11)	50	50
Sulfur dioxide	135	85

For refrigerants not covered in the table the minimum test pressure for the high pressure side shall be not less than the vapor pressure of the refrigerant at 135 F. For the low side the test pressure shall be not less than the vapor pressure of the refrigerant at 110 F. In no case shall the high or low side test pressure be less than 50 psi.

24. The **performance** of domestic refrigerators refers to various aspects of the refrigerator behavior under different conditions of room temperature and humidity, internal temperature of the cabinet and various conditions of loading. The data obtained are usually:

a. Rate of consumption of ice, fuel or electrical energy.
b. Characteristics of auxiliary functions, such as production of ice cubes.
c. Reaction of the box to high humidity, such as condensation on the surface of the cabinet and absorption of water into the insulation.
d. Resistance of the finish to light, scratching, grease, etc.
e. Mechanical durability of the mechanism and hardware.

Problems dealing with durability of the equipment have been covered under the design heading. However, the **rate of energy consumption** will bear further discussion. When operating with the cabinet door closed, a refrigerator is practically independent of ambient humidity except as moisture may be absorbed into the insulation and thereby affect the insulating value. Ambient temperature does have an important effect on performance. In an ice refrigerator the cabinet temperature ordinarily varies with the ambient room temperature. J. E. Starr showed that the relationship between temperature within and without an ice refrigerator is theoretically linear. How closely this is approximated in laboratory measurements is shown in **Figs. 16** and **17.**

In a mechanical refrigerator a change in ambient temperature has a marked effect on energy consumption, assuming the box temperature remains constant. Three factors are involved as the ambient temperature increases: (1) the differential through the cabinet walls increases and thus the load on the cooling unit increases; (2) more energy is required because of the greater temperature difference between which heat is absorbed in the evaporator and rejected to the air; and (3) the efficiency of the system decreases because of more leakage past

the piston, more re-expansion and less perfect valve action.

In an electrically driven refrigerator, voltage increases may cause the power consumption to go up or down depending upon the design of the motor and the loading.

Figs. 18 and **19** show performance of typical 4-cu ft, 7-cu ft and 9-cu ft refrigerators in one manufacturer's line. **Figs. 20** and **21** show performance range of approximately 7-cu ft refrigerators produced by nine different manufacturers during 1939, 1940, and 1941.

25. Interior and external refrigerator temperatures may be measured by liquid in glass thermometers, electric thermocouples, or electric resistance thermometers. **Glass thermometers** are relatively inexpensive, but the refrigerator door must be opened in order to read them and this results in a change of box temperature. While this may not affect the accuracy of the reading, it slows up the testing of the refrigerator, as the box temperatures must be stabilized again before initial kilowatt hours or running time readings can be taken.

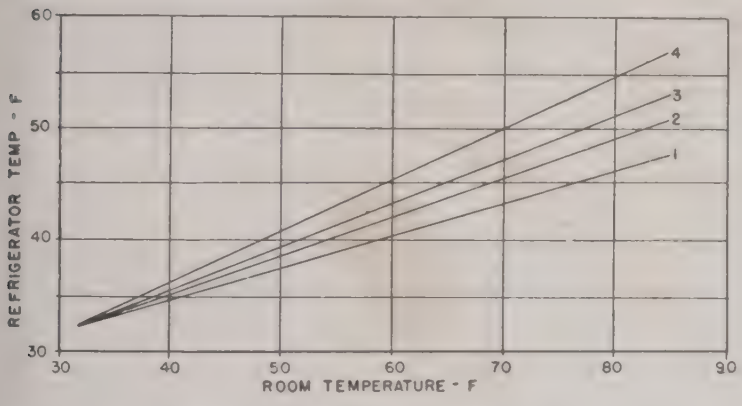


Fig. 16. Room Temperature Affecting Refrigerator Temperature, 70 lb of ice; Measurements at Four Points

Resistance thermometers are accurate and reliable for measuring cabinet and room air temperatures but they are usually too bulky to measure surface temperatures on evaporators, tubes, compressors, etc. **Thermocouples** with potentiometer indicators are the only satisfactory means of measuring temperatures for development work. This is because thermocouples are cheap, extremely small, and can follow any desired rate of temperature change. They can also be lagged by putting them in contact with a mass of metal or fluid to read average rather than momentary cycling temperatures. Recorders are available for use with thermocouples, or resistance thermometers that will plot as many as 16 records on one chart.

26. During testing, it is important that **temperature gradients** throughout the room be kept to a minimum because of the effect of room temperature on refrigerator performance. The temperature at the floor should not be essentially different than in the rest of the room, or the cold floor air being drawn through the condenser will give fictitious results. When testing a thermal convection-cooled condenser located on the back of the refrigerator, it is necessary to keep the room air velocity low enough so the heat dissipated from the condenser will not be affected. The NEMA household electric refrigerator standards require that the room temperature be maintained within plus or minus 1 F of the specific values, and that the vertical temperature gradient from the floor to a height of 7 ft shall not exceed a half a degree for each foot of vertical distance.

27. The best method for making an **ice-freezing** test is to measure ice-freezing time by means of thermocouples inserted in the ice-tray cells that will freeze last. As the water cools, the temperature in the

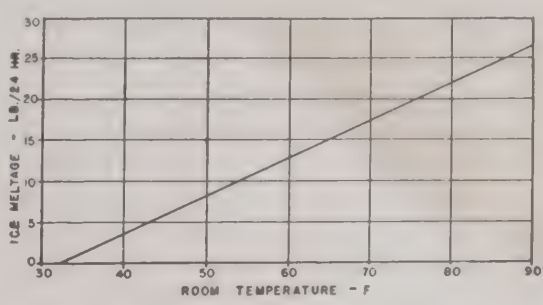


Fig. 17. Effect of Room Temperature on Ice Meltage with 70 lb of Ice

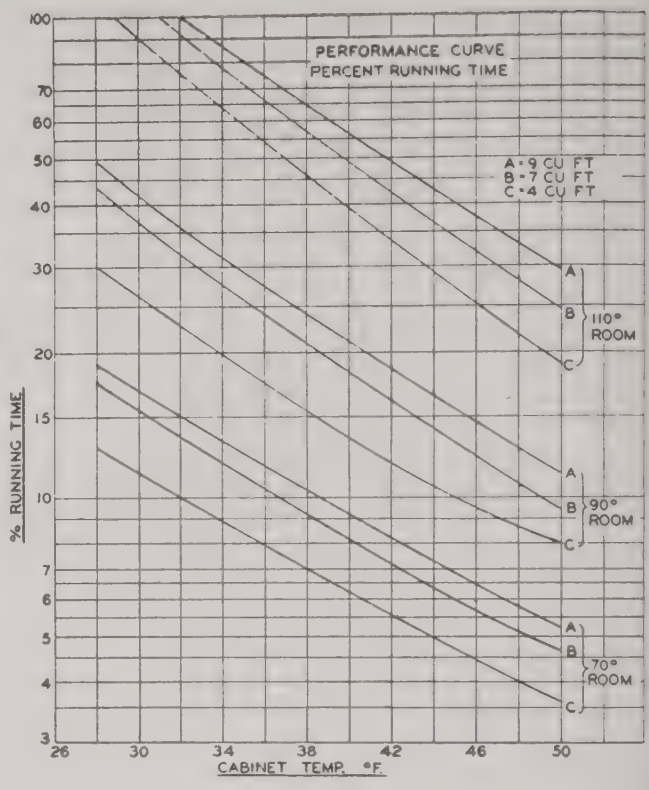


Fig. 18. Percentage Running Time vs. Room Temperature and Box Temperature for Typical 4-cu ft, 7-cu ft and 9-cu ft Refrigerators in One Manufacturer's Line

ice trays will go down rapidly to 32 F. During the actual freezing period the temperature will stay at 32 F and at the conclusion of this period will go down rapidly from this point. It is, therefore, easy to determine the ice-freezing time by means of thermocouples and a temperature recorder. If an ice tray is placed on the shelf of a dry evaporator, there will be an entirely different rate of ice freezing than if the contact between the tray and the evaporator were thoroughly wetted before the test. Differences as great as two to one in ice-freezing time can result from this contact variation. Therefore, it is usual to wet the evaporator surface before putting the tray in place.

Kilowatt hours can be readily measured by an integrating kilowatt hour meter having a scale multiplier of .01 or less, so that the data can be read to one-hundredth and estimated to one-thousandth. For measuring the running time self-starting electric clocks are most satisfactory.

28. Much can be learned from the standpoint of development by operating the compressor unit with a **calorimeter** of some kind. One type of calorimeter consists of a relatively short copper tube surrounding a tubular electric heater. Refrigerant is admitted in one end of the tube through a constant pressure expansion valve, and the heat input adjusted until the desired degree of superheat is obtained at the output. Modifications of this particular idea have been used in connection with flooded evaporators.

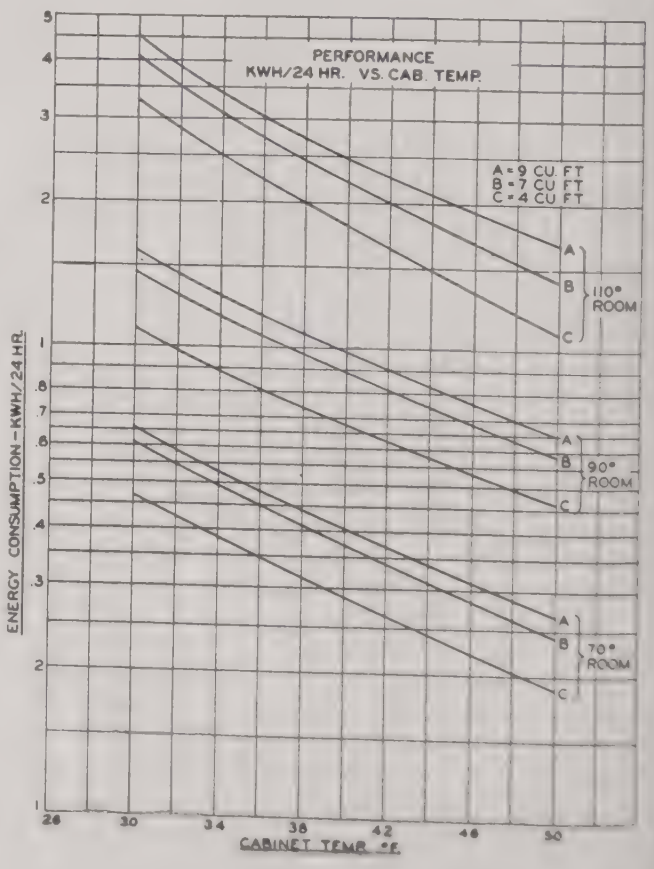
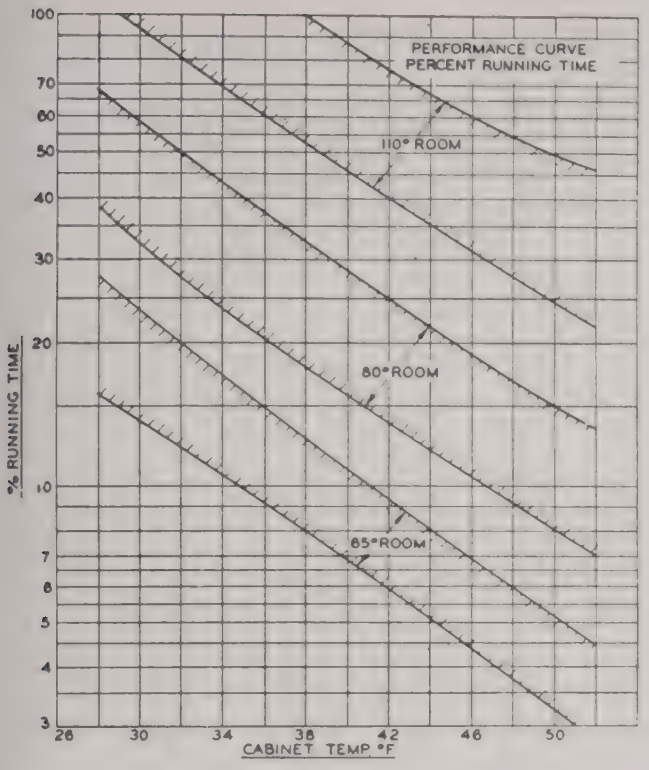


Fig. 19. Kilowatt Hours Consumption for 24 Hours for Typical 4-cu ft, 7-cu ft and 9-cu ft Refrigerators in One Manufacturer's Line



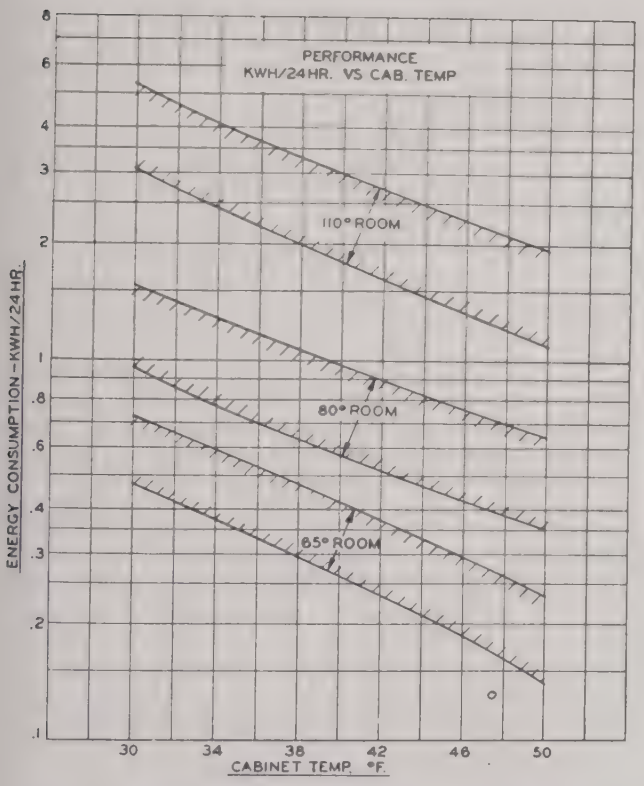


Fig. 21. Kilowatt Hours per 24 Hours for Refrigerators Produced by Nine Different Manufacturers during 1939, 1940 and 1941

Fig. 20. Percentage Running Time vs. Room Temperature and Box Temperature for Refrigerators Produced by Nine Different Manufacturers during 1939, 1940 and 1941

One feature of a good calorimeter is small thermal mass so that its response to a change is rapid, thus reducing testing time. A calorimeter achieving these characteristics particularly well was designed by D. D. Wile.[15] It employs a heater immersed in the liquid of a secondary refrigerant circuit, while the evaporator of the compressor under test functions as a condenser for this secondary circuit. A control on the calorimeter turns the heater on and off in response to the pressure of the secondary system, while the total heater input over a period of time is measured by an integrating kilowatt meter. Accurate tests can be obtained from readings extending over a period of only three minutes.

Cabinet heat leakage tests can be run in several ways:

a. The cabinet is placed in a hot room and refrigerant is passed to the evaporator at a rate necessary to maintain a constant cabinet temperature. The heat leakage per degree may then be determined from the quantity of the refrigerant circulated, the conditions of the refrigerant entering and leaving, and the interior and exterior cabinet temperatures.

b. The cabinet is placed in a cool room (about 50 F) and the cabinet interior heated with electric heaters to a point where the mean temperature of the insulation is about 75 F. The overall heat loss may then be determined from the heater input and the interior and exterior cabinet temperatures.

c. The cabinet can be operated in a hot room with a calibrated refrigerating unit of the type normally installed in the cabinet. This method takes into account

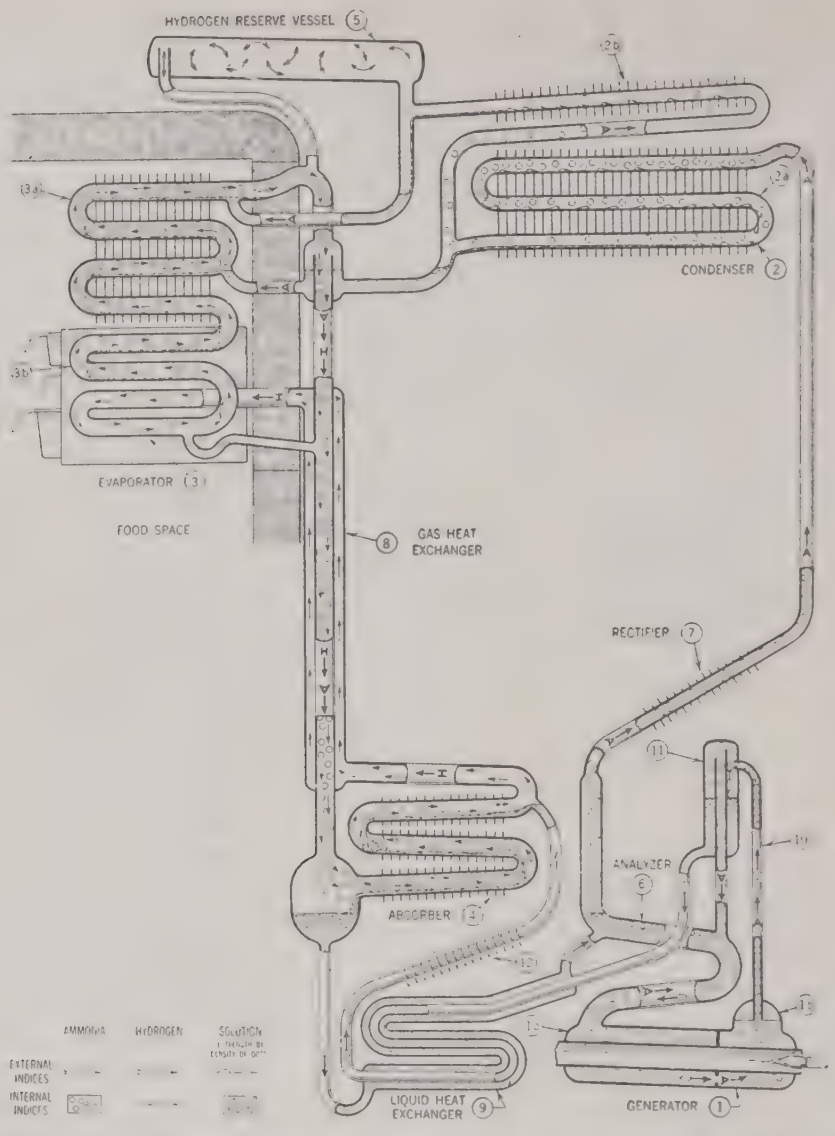


Fig. 22. Schematic Arrangement of Servel Absorption Machine (Platen-Munters System)

the localized heating of cabinet surfaces from the condenser and compressor. This method gives the most accurate overall result as the operating conditions are all normal.

d. Ice can be placed in the cabinet and its rate of melting measured.

The Operating Cycle of the Heat-Operated Servel Refrigerator

29. The Servel refrigerator is heat-operated, the **freezing unit** being an absorption machine of the Platen-Munters three-fluid type. The unit is made up of a number of steel vessels and pipes welded together to form a hermetically sealed system. All spaces in the system are in open communication with each other and hence are at the same total pressure, aside from minor variations which arise from the presence of fluid columns and which are used to circulate the fluids.

The **charge** includes an aqua-ammonia solution of a strength of about 30% concentration (ammonia by weight) and hydrogen. For a unit of sufficient capacity for a 5-cu ft cabinet, the approximate charge is 1.1 lb ammonia, 2.6 lb water, 0.03 lb hydrogen. The liquid is charged into the unit as solution and then hydrogen is added.

The **elements** of the system in **Fig. 22** in-

clude: (1) a generator, sometimes called boiler or still; (2) a condenser; (3) an evaporator; (4) an absorber; and (5) a hydrogen reserve vessel. There are three distinct fluid circuits in the system: an ammonia circuit including the generator, condenser, evaporator and absorber; a hydrogen circuit including the evaporator and absorber; and a solution circuit including the generator and absorber.

30. Starting with the **generator,** heat is applied by a gas burner or other source of heat to expel ammonia from solution. The ammonia vapor thus generated flows to the condenser. In the path of flow of ammonia from the generator to the condenser are interposed an analyzer (6) and a rectifier (7). Some water vapor will be carried along with the ammonia vapor from the generator. The analyzer and rectifier serve to remove this water vapor from the ammonia vapor. In the analyzer, the ammonia passes through strong solution which is on its way from the absorber to the generator. This reduces the temperature of the generated vapor somewhat to condense water vapor and the resulting heating of the strong solution expels some ammonia vapor without additional heat input. The ammonia vapor then passes through the rectifier (7) where the residual small amount of water vapor is condensed by atmospheric cooling and drains to the generator (1) by way of the analyzer (6).

The ammonia vapor, which is still warm, passes on to the section (2a) of the condenser (2) where it is liquefied by air cooling. The condenser is provided with fins for this purpose. The ammonia thus liquefied flows into the evaporator (3) at an intermediate point. A liquid trap is interposed between the condenser section (2a) and the evaporator to prevent hydrogen from entering the condenser. Ammonia vapor which does not condense in the condenser section (2a) passes to the section (2b) of the condenser and is liquefied, and flows through another trap into the top of the evaporator.

The **evaporator** is made up of two sections (3a and 3b). The upper section (3a) is provided with fins and directly cools the food space. The lower section (3b) is in direct contact with the ice-freezing compartment.

Hydrogen gas enters the lower evaporator section (3b) and, after passing through a precooling pipe part, flows upward, in counterflow to the downward flowing liquid ammonia. The effect of the placing of a hydrogen atmosphere above the liquid ammonia in the evaporator is to reduce the partial pressure of the ammonia vapor in accordance with Dalton's law of partial pressures. While the total or gage pressure in the evaporator and the pressure in the condenser are the same, there is substantially pure ammonia in the space where condensation is taking place, and consequently the vapor pressure of the ammonia substantially equals the total pressure.

Under Dalton's law, the total pressure of a gas mixture is equal to the sum of the partial pressures of the individual gases. Consequently, in the evaporator the partial ammonia vapor pressure is less than the total pressure by the value of partial pressure of the hydrogen. The lesser ammonia vapor pressure results in evaporation of the ammonia with consequent absorption of heat from the surroundings of the evaporator and the cooling of the surroundings which are in a well-insulated enclosure.

The cool heavy gas mixture of hydrogen and ammonia vapor formed in the evaporator leaves the top of the evaporator and passes downward through the center of the gas heat exchanger (8) to the absorber (4). In the **absorber,** ammonia is absorbed by water, and the hydrogen, which is practically insoluble, passes upward from the top of the absorber through the external chamber of the gas heat exchanger (8) into the evaporator. Perfect separation of gases is of course not possible and some ammonia vapor passes with the hydrogen from the absorber to the evaporator. It is probably more accurate to call the gas flowing from the evaporator to the absorber strong gas (hydrogen strong in ammonia), and the gas flowing from the absorber to the evaporator weak gas (hydrogen weak in ammonia).

Since the weight of a gas is proportional to its molecular weight, and the molecular weight of ammonia is 17 and the molecular weight of hydrogen 2, it follows that the specific weight of the strong gas is greater than that of weak gas. This difference in

specific weights is alone sufficient to initiate and maintain circulation between the evaporator and the absorber. Since the absorber is below the evaporator, it is possible to have upward gas flow in the evaporator. The long vertical column of strong gas in the central chamber of the gas heat exchanger is heavier than the vertical column in the absorber, external heat exchanger space and evaporator, despite the fact that the gas in the evaporator is heavy. Consequently the gas will flow as above stated, due to the difference in specific weights of the gases in the different vertical branches of the circuit. The gas heat exchanger transfers heat from the weak gas to the strong gas. This saves some cooling in the evaporator by precooling the entering gas. A liquid drain at the bottom of the evaporator is connected to the down-flow space of the gas heat exchanger.

Counter-current flow in the evaporator permits the location of the box cooling section of the evaporator in the most effective position, at the very top of the food space. Also, the gas leaving the lower temperature evaporator section (3b) can pick up more ammonia at the higher temperature prevailing in the box cooling evaporator section (3a), thereby increasing capacity and efficiency. There is still another advantage in that liquid ammonia flowing to the lower temperature evaporator section is precooled in the upper evaporator section.

The dual liquid connection between the condenser and the evaporator is advantageous in applying the unit to the cabinet. It permits extending the condenser below the top of the evaporator to provide more surface while having gravity flow of liquid ammonia to the evaporator.

The **two-temperature evaporator** partially segregates the ice-freezing function from the box-cooling function. This provides a better humidity condition in the food space because, due to the higher temperature of the box cooling section (though adequately low for proper preservation), and due to the reduced surface of the low temperature section permitted by the partial segregation of function, less moisture is extracted to form frost.

In the absorber, a flow of weak solution (water weak in ammonia) comes in direct contact with the strong gas. The liquid and gas flow is countercurrent. The weak solution is thus enriched or strengthened while the strong gas is weakened.

From the absorber the strong solution flows through the liquid heat exchanger (9) to the analyzer (6) and thence to the strong liquid chamber (1a) of the generator (1). Heat applied to this chamber causes vapor to pass upward through the analyzer (6) and to the condenser. The solution passes through an aperture in the generator partition into the weak liquid chamber (1b). Heat applied to this chamber causes vapor and liquid to pass upward through the small diameter pipe (10), as in an "air lift," to the separation vessel (11). Liberated ammonia vapor passes through the analyzer (6) and thence to the condenser. The weak solution flows through the liquid heat exchanger (9) and to the absorber. The liquid heat exchanger precools the liquid entering the absorber and preheats the liquid entering the generator. Further precooling of the weak solution is obtained in the finned air cooled loop (12) between the liquid heat exchanger and the absorber.

The heat which is liberated by absorption of ammonia in the absorber is carried away by air flowing in contact with the absorber fins.

The **hydrogen reserve vessel** (5) which is connected between the condenser outlet and the hydrogen circuit may be described as a reservoir for hydrogen gas while the refrigerator is operating under normal room temperature conditions. Under these conditions an appreciable part of the hydrogen in the system is stored in the reserve vessel. The remainder is located in the evaporator-absorber circuit and serves to balance the condenser pressure. The pressure must be adequate to liquefy the ammonia gas in the condenser. If the pressure is increased, the efficiency under normal conditions will be impaired, and yet it is necessary to have a higher pressure in the system to insure condensation of the ammonia under high room temperature conditions. The reserve vessel, through its connection in the system, is an automatic pressure variant to take care of the variable room temperature and loads. It permits lower operating pressure at lower room

temperature, thereby resulting in better efficiency, but provides the higher operating pressure necessary under extreme conditions to insure condensation of ammonia.

HOUSEHOLD FREEZERS

31. The household freezer in many ways is similar to two long established products —the household refrigerator and the ice-cream cabinet. In fact, the **upright** and **chest-type** freezers have evolved from household refrigerator and ice-cream cabinet, respectively. Both designs must meet the same requirements. Like the household refrigerator, the home freezer should be attractive in appearance, occupy a minimum of floor space, and be convenient to use[17] and operate with a minimum of noise and attention. Like the ice-cream cabinet, the home freezer must operate at a temperature well below freezing. The home freezer must be able to store commercially frozen foods and it should also be able to freeze and store home-processed foods.

Cabinets

32. In general, **cabinet construction** follows the pattern established in similar equipment, of all-steel construction with synthetic enameled steel exteriors. Liners may be steel, coated with synthetic enamel, vitreous enamel, or metallic coatings. Aluminum and stainless steel are also used. Insulation is similar to that in other types of cabinets, except that the lower operating temperatures than for household cabinets require greater insulating effect, which may be achieved with better insulation, greater thickness or both.[23] As many of these cabinets are placed in basements, or other locations where the relative humidity will be close to 100%, some external sweating will be inevitable. Thus the design and finish of the exterior parts, particularly about the bottom of the cabinet and around the lid of chest-type cabinets, should be such as to minimize the harmful effects of drippage and water collection. For installations in kitchens, drippage cannot be tolerated and the surface temperature should be kept as high as on domestic refrigerators.

Due to the low temperatures existing in these cabinets, extreme care must be exercised in the sealing of the exterior shell in order to prevent the entrance of atmospheric moisture. All joints in the external shell should be either soldered, brazed or sealed with wax, asphalt or some similar material.

33. Because of the great number of small packages stored in a home freezer, considerable thought should be given to the problem of **accessibility.** From this standpoint, the upright freezer appears to have a considerable advantage over the chest-type design.[18] It has been thought that a considerable advantage accrued to the chest-type design because of less air spillage when the door is open.[19] However, Tanner has shown that with two door-openings a day, the percentage increase in operating time for a 9-ft upright freezer is 1.86, and for a chest-type freezer 3.4. In both cases, the loss is negligible; the greater loss in the chest-type model is probably due to the greater open time and the increased warming up of the product since more packages must be removed each time in order to get at the desired package.[20] If, however, the freezers were to be used in a commercial establishment, where instead of the normal openings of two a day the door or lid was open almost continuously, the chest-type freezer would no doubt show an advantage over the upright type.

In order to improve accessibility, sliding shelves, doors, baskets[34] and drawers have all been used. In the upright models, inner horizontally hinged doors that can be let down to provide rearranging shelves have been used. These doors also improve the appearance of the freezer and reduce the power consumption with the main door open.

Evaporators and Refrigerant Circuits

35. Because the field is new, there is much variety in the **evaporators** and the cooling circuits used. In upright freezers, there may be placed in the top of the storage compartment an evaporator that is similar, though ordinarily larger, than that in the standard domestic evaporator. For cooling the product that may be tightly packed into the space below, thus making air circulation difficult, a secondary system

on the liner wall may be used that cools the compartment as described under domestic refrigerators, or primary coils may be fastened to the outside of the liner. Other upright freezers refrigerate the product load by means of primary coils fastened to or built into the shelves. A duct system may be used in the storage compartment, through which cold air is circulated by means of a small fan. In chest-type freezers, the primary refrigerant coils may be wrapped around the sides and sometimes the bottom of the food compartment, or evaporators may be put in the storage space.

36. Normal **storage temperature** is 0 F for this type of equipment. In some designs, small areas run at −5 to −20 F temperatures to produce more rapid freezing than would be achieved at zero. In the upright freezer shown in **Fig. 23,** this low-temperature area is in the evaporator, and in the chest-type freezer, **Fig. 24,** the freezing section is in the small compartment over the compressor.

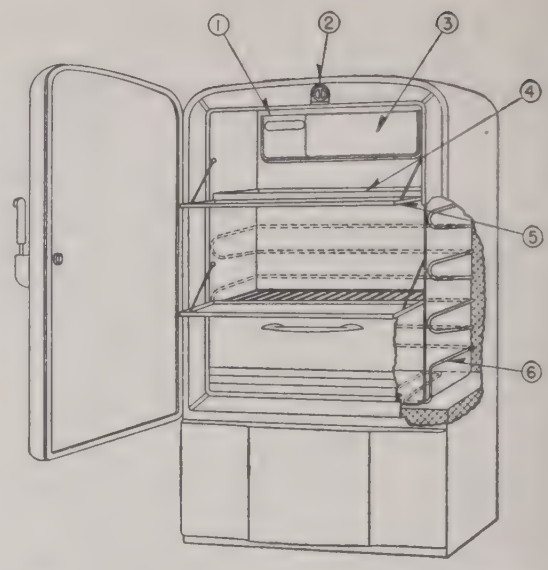


Fig. 23. 6-cu ft Upright Home Freezer Showing Evaporator, Compartment Doors, and Secondary System

1. Evaporator with secondary condenser attached to back.
2. Center knob
3. Freezing compartment
4. Defrost tray and shelf
5. Compartment shelves
6. Secondary evaporator coils on sides, back and bottom of compartment.

Defrosting a Home Freezer

37. Because of the low temperatures involved and the large product load that must be kept cold, **defrosting** is generally more difficult than in a domestic refrigerator. Fortunately, due to the limited number of door-openings frost collection is slow, and defrosting will ordinarily not have to be done more than one or two times a year. Upright or chest-type freezers that have evaporators separated from the walls and are so arranged that pans of hot water can be placed in them, are easily defrosted without scraping or removing the product load. If the cabinet is further so arranged that a removable pan can be used to catch the drip, the defrosting problem becomes very easy.

If the primary cooling coils are on the walls or on plates that cannot be defrosted by pans of hot water, then the frost is removed by scraping it from the cold surfaces. An alternate method is to remove the product load and allow the cabinet to warm up above freezing. Even if the main body of food storage space is cooled by a secondary system on the walls, the frost collection is on the primary evaporator. It happens that if the primary evaporator is thoroughly separated from the rest of the food storage space, the water vapor will not migrate to it but would collect on the walls. If this situation should exist, of course defrosting would have to be done by scraping. If, however, there is a primary evaporator in open communication with the main food storage space and if this evaporator runs at a temperature considerably below any other cooling surface, frost will collect upon it.

Compressors

38. Reliability is of paramount consideration in connection with home freezers.[21,22] Even a very small freezer can readily contain $100 worth of food.[20] The compressor, and in fact all of the elements that go to make up the system, should be designed with this objective in view. **Ease of replacement** is also desirable, so that in case of failure quick restoration of service can be achieved.

These considerations suggest that the direction of development of the home freezer refrigerating system will ultimately be the same as that of the home refrigera-

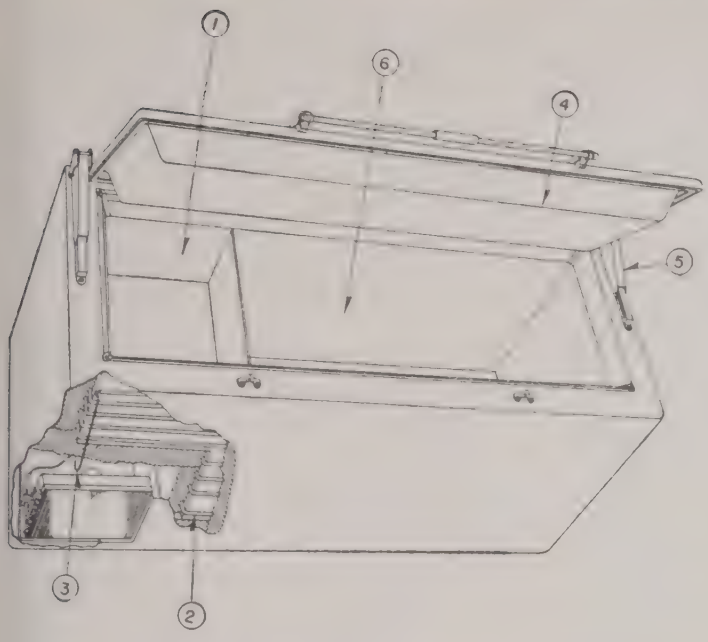


Fig. 24. 15-cu ft Chest-Type Freezer

1. Freezing compartment
2. Primary wall cooling coils
3. Capillary tube expansion device
4. One piece plastic inner lid pan
5. Spring loaded hinges to balance lid
6. Main storage compartment

tor, where the demand for reliability and easy service has led to almost exclusive use of unit-type construction and hermetically sealed compressors.[20,23] However, this type of construction does not now universally prevail in home freezer design. Many freezers are built with open compressors and only very few for unit-type replaceable refrigerating systems.

Due to the necessity of achieving a temperature of 0 F in the food storage space, compressors must be capable of carrying their load at a suction temperature of from −5 to −30 F, which necessitates low head clearances and good construction. For the same displacement, the motor load is less than in a domestic refrigerator; thus care must be taken to see that the motor capacity is adequate to provide the initial cooling of a warm cabinet. As in the household refrigerator, Freon-12 is almost universally used. Because of its ability to develop more capacity at low temperatures with the same displacement, Freon-22[24,25,26] has been considered. In applying Freon-22, particularly in a compressor designed for Freon-12, care should be exercised to be sure that the other elements of the system are adequate to carry the additional load.

39. Because of the lower temperatures encountered, **moisture** is more of a problem than in household refrigerators. The refrigerating systems must either be dried sufficiently during the initial processing or driers included as part of the system. Drying is discussed in Chapter 18. With hermetic systems particularly, the amount of water to be removed is so large that it would not be practical to attempt to substitute a drier for a good job of factory drying.

The **lubricating problem** also demands special care in a freezer because of the low temperatures involved; only oils that are known to be satisfactory from the standpoint of wax formation at low temperatures should be used.[21,28,29] (See also Chap. 17.)

Expansion Devices

40. Capillary tubes are commonly used as expansion devices. Thermostatic expansion valves are also used but care must be exercised here, because at lower temperatures proper control, due to the low pressure change per degree of the temperature change, introduces problems.[26,30] When such troubles are experienced there are special types of expansion valves for low-temperature operation that reduce this source of trouble.[31]

Condensers

41. Condensers generally follow the pattern established in domestic refrigerators. Home freezers are equipped with **finned condensers** and **motor-driven fans** in the larger sizes, and may have thermal convection condensers in the smaller sizes. No particular problem exists in the application of a condenser to the home freezer, except that it is part of the system that must be critically examined in connection with pulling down the warm cabinet. Because of the greater heat of compression per Btu of refrigerating effect, the condenser must

be larger than for applications producing the same number of Btu's at higher suction temperatures. During freezing and pull-down the load is higher, which must be considered in the choice of condensers.[21]

Controls

42. Controls for home freezers do not introduce any special problems, provided they are designed to operate at the temperatures encountered. The control should provide as **narrow** a **differential** as practical, in order to avoid fluctuations in temperature in the stored foods, inasmuch as such changes in temperature will cause effects to be discussed later.[32,43]

Refrigeration Failure

43. As mentioned, failure that causes spoilage of the entire contents of a home freezer will be serious. However, refrigeration failures up to 72 hr may not necessarily bring spoilage of the product load. This question is discussed in detail by Tressler, McCracken and others.[34,35] Some freezers are equipped with **alarms** that ring a bell or light a light in cases of power failure or abnormal rise in temperature.[23,35] These devices are very useful, especially in view of the fact that the freezer may be placed in a spot not visited too frequently.

Testing and Rating

44. The **testing** of household freezers, very similar to the testing of household refrigerators, is covered by **NEMA** Publication 45-106, quoted below.

HR9-5. Rating the Storage Capacity for Home Freezers

A. Food Storage Volume Rating
   1. The storage volume of home freezers shall be the volume in cubic feet of the space specifically arranged for the storage of frozen foods, except as modified in paragraphs 2 and 3.
   2. The volume of any space arranged for freezing shall be included in the storage volume since this space may also be used for storage.
   3. In determining the storage volume, the volume of permanently fixed shelves and partitions shall be deducted from the gross volume.

B. Food Storage Weight Rating
   The storage volume rating may be supplemented by a food storage rating which shall be 35 pounds per cubic foot of storage volume. (See Note 1.)
   Example. Storage volume, 6 cu ft
   Food storage capacity, 210 lb

Note 1.—Approximately 40 lb per cu ft loading is obtained with uniform size packages of chopped meat or fruits in syrup. Approximately 30 to 35 lb per cu ft loading is obtained with miscellaneous meats in family-size cuts. Approximately 25 to 30 lb per cu ft loading is obtained with an assortment of packages, including fruits, vegetables and miscellaneous cuts of meat.

Note 2.—It is recommended that the manufacturer include in his user instructions the maximum number of pounds of fresh foods which should be frozen in any 24-hr period.

Food Packaging, Freezing and Storage

45. It is not the purpose of this section on home freezers to treat exhaustively the matters of choice of products, blanching, freezing, packaging, etc. However, there are certain factors that should be mentioned.

Frozen-food **storage temperature** depends entirely upon the desired keeping time. There is general agreement at the present that 0 F will give satisfactory keeping for a period ranging from six months to a year, and this is the usual temperature for which home freezers are designed. This temperature should be held as constant as practical. Birdseye states that fluctuations of temperature will cause a certain percentage of the ice to melt during the warming-up period and refreezing during the cooling-down period, and that this fluctuation will cause crystal-growth giving the same result as slow freezing.[32] Woodroof has shown that fluctuations in temperature will also cause the migration of water from the product to the walls of the container.[33] Two degrees is referred to by Tressler as a satisfactory temperature variation.[34]

46. As ice has a considerable vapor pressure even at very low temperatures, the frozen food must be protected from **drying.** Frozen-food storage compartments that are cooled by coils on the walls maintain a higher relative humidity[39] than those cooled by a low-temperature evaporator; however, even in the case of the wall-cooled compartment, evaporation is far too rapid for good preservation of the frozen food and it must be very carefully packaged with vapor-proof material.[36]

47. Crystal size in frozen food has a bearing on quality. The higher the rate of freezing, the smaller the crystal size and the smaller the consequent rupture of cell walls.[32] Rate of cooling depends not only

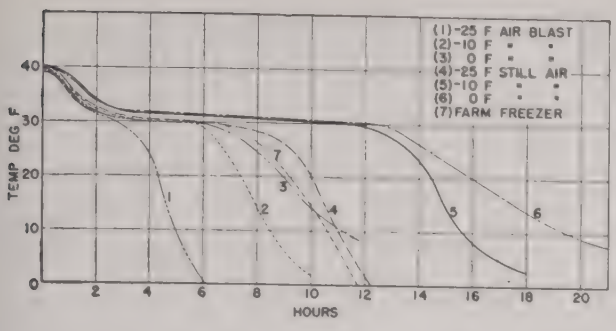


Fig. 25. Curves Showing the Rate of Drop in Temperature in Center of 3¾-lb Beef Roasts Frozen in Still and Rapidly Moving Air

on the temperature of the surrounding medium, but also on the ease of heat transfer between the surrounding medium and the package. **Fig. 25** shows the variation in freezing rates in still and moving air at different temperatures as determined by DuBois and Tressler.

Freezing is also accelerated by bringing the packages to be frozen into intimate contact with cold freezing surfaces. In addition to improving, somewhat, the crystal size, rapid cooling has another very important function in the household freezer, and that is the rapid lowering of the center of the individual packages or groups of packages from room temperature to a temperature at which spoilage is slow. These two factors indicate the desirability of some provision for rapid cooling of products to be frozen in the home freezer. The manufacturers of a freezer should be prepared to make specific recommendations in connection with his product, so that the user will know the maximum quantity of food that can be frozen in 24 hr at a satisfactory cooling rate. This problem is tied up not only with evaporator designs, but also with compressor capacity, as an evaporator designed for intimate contact between its surfaces and the product load might not cool rapidly enough if the loading were beyond the compressor heat-removing capacity. The size of the quick-freezing compartment and the design of the system should be such that when freezing a maximum load of warm food, the storage compartment temperature will not rise unduly, not over 2 degrees, according to Tressler.[34]

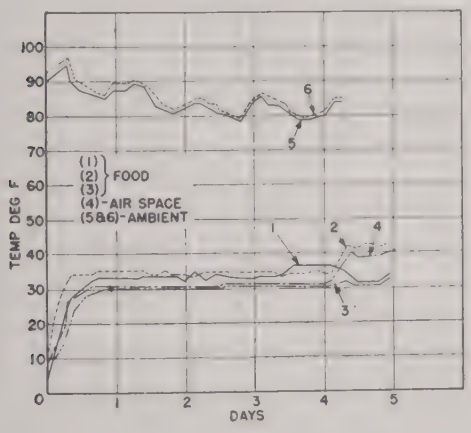


Fig. 26. Defrosting Test. Curves Indicating Temperature Rise in Package of Frozen Broccoli Held in 2-cu ft Top-opening Home Freezer during Power Outage

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30. INSULATION OF CABINETS

DESIGNING of a modern refrigerator cabinet involves the selection of insulation, thermal design, keeping insulation dry, handling insulation in production, and testing.

Selection of Insulation

1. Selecting the best insulating materials available for any cabinet design is itself a problem of primary importance and involves such factors as (a) conductivity, (b) cost and supply, (c) material odors, and (d) long life.

a. The **thermal conductivity** of an insulating material is a measure of its ability to conduct heat, or an inverse measure of its ability to resist the flow of heat, as obtained by test. For tables of conductivities of materials, see Chap. 12.

b. The **cost and supply** of insulation are obtainable by shopping, or, in other words, by calling in the various manufacturers or their representatives and requesting such information as is desired. Frequently the insulation best suited for the design in question may require fabrication to specific drawing dimensions, in which case it will be necessary to submit drawings when requesting quotations. The cost of handling and installing insulation varies widely with various materials. The weight, or density, is also important, as it affects the outgoing freight cost on the finished refrigerator.

c. Most established insulating materials are free of objectionable **odors,** but there are always chances for the product to become contaminated by defective materials or processes. It may also absorb odors in transit or storage, and a periodic check on insulating materials used on the production line is advisable. It is possible for a material to contain a characteristic odor and yet not impart it to foods. For this reason, it is not advisable to depend on the human sense of smell alone. The ability of the material to impart objectionable odors to foods should also be checked. This may be done by enclosing a sample in a glass jar, together with a sample of sweet unsalted butter, sealing, and refrigerating at 40–45 F for 24 hr. When the butter is tested, several opinions should be obtained.

d. The **life** of most insulating materials depends on the chemical stability of the raw materials and the combination of these materials in the finished product. For example, a fiber with a binder should be nonsoluble in water, and not inclined to deteriorate when exposed to hot, cold, humid, or dry atmospheres. Moisture and water are natural enemies of insulating materials. Usually, materials that have low absorption and capillary coefficients are better, but highly absorbent materials are often entirely satisfactory for cabinets where the insulation will normally remain dry.

Thermal Design of Insulation

2. The first consideration in the thermal design of cabinet insulation is determination of the proper **thickness** of wall insulation. The two factors which determine this are cost and elimination of exterior sweating. If thicknesses greater than 3½ in. for 40 F or 4½ in. for 0 F refrigerators are used, the cabinet insulation space may be greater than the available food space, and as a result the design is bulky, heavy, and costly. Thicknesses appreciably less than 1½ in. for 40 F conditions and 2½ in. for 0 F conditions are impractical, due to the exterior surface temperatures running below the dew point, with attendant sweating under high temperature and high humidity conditions. It is sometimes desirable for reasons of style to design the cabinet with walls which differ in thickness, or are of non-uniform thickness.

Heat leakage through **framing** and **breaker strips** may be kept to a minimum by reducing their cross section. Frequently, thin plastic sections of great strength have been substituted for heavy wood members, effecting a reduction in heat leakage despite the higher conductivity of the plastic material.

Air spaces extending through a wall, or from the food space to the outside wall, transmit appreciable amounts of heat. They should thus be kept at a minimum in number and cross section. If the insulation is to be of a rigid or semi-rigid board or batt type, step joints at corners, as in **Fig. 1,** will help to reduce the joint losses.

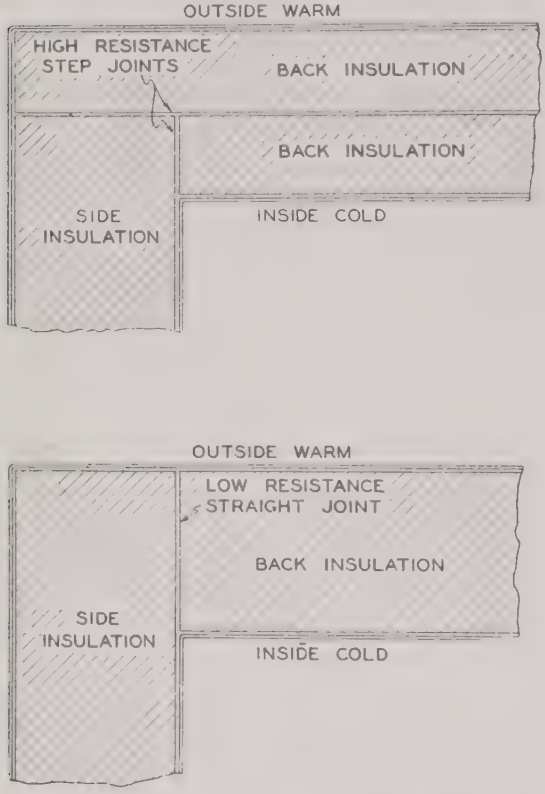


Fig. 1. Above—Good Insulation Design Using Step Joints; Below—Poor Insulation Design

The rate of **heat transmission** into a cabinet is obtained by calculating each wall separately and adding the sum to the rate of heat transmission through the breaker strips, air gaps, and all other channels not included in the wall calculations. The rate of **heat transmission** through a wall is calculated thus:

a. Wall insulated with one material only:

$$Q = \frac{kA(t_1 - t_2)}{L}$$

where

Q = Rate of heat transmission, Btu per hr
k = Thermal conductivity, Btu in. per sq ft hr deg F
A = Mean wall area, sq ft
t_1 = Temperature of outside hot surface, deg F
t_2 = Temperature of inside cold surface, deg F
L = Average wall thickness, in.

b. Wall insulated with several materials in series having thicknesses L_1; L_2; L_3, etc., and conductivities k_1; k_2; k_3, etc., respectively:

$$Q = UA(t_1 - t_2)$$

$$U = \frac{1}{\frac{L_1}{k_1} + \frac{L_2}{k_2} + \frac{L_3}{k_3} \cdots}$$

where

U = Coefficient of transmittance, Btu per sq ft hr deg F

If the air-to-air temperature differential is known, an approximation of the surface-to-surface temperature may be obtained by taking it as 90%. The different surfaces will vary in temperature, due to the room air opposite these surfaces. For instance, if the refrigerating machine is located under the cabinet, the heat given off by the unit and condenser will cause bottom and back surface temperatures to run several degrees warmer than the sides, door, and top. It is good practice to proportion the insulation to keep out heat from the unit.

Heat leakage around the doors is difficult to calculate, and varies greatly with the construction. Such losses are best obtained from heat leakage tests. A refrigerator as shown in **Fig. 2** would have a leakage

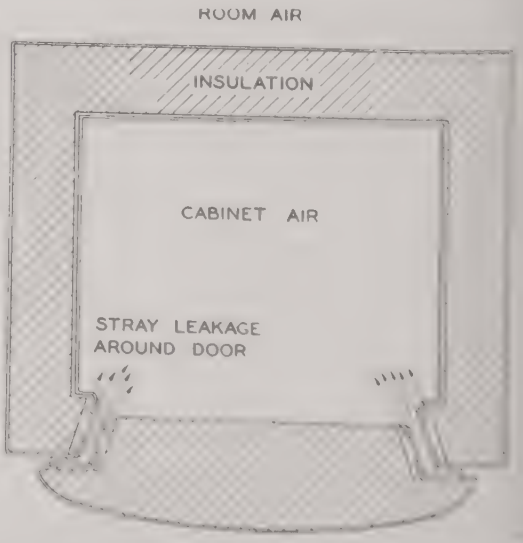


Fig. 2. Cabinet Section Showing Heat Channels Difficult to Calculate

factor of about .05 Btu per hr F for each linear foot of door perimeter, the temperature drop in this case being from room to cabinet air. This value may be used for general design purposes, but only as a rough approximation.

The maximum permissible leakage is frequently determined by the capacity of an existing refrigerating machine and the minimum by the amount of evaporator surface it contains. This method is, of course, more difficult than designing a unit and evaporator to match an existing cabinet of good appearance and high insulating value. Where the leakage rate is thus fixed by the machine to be used, the thermal design of the cabinet must be arrived at by manipulating wall thicknesses, wall areas, and conductivities until the desired cabinet proportions and leakage rate have been established.

Keeping the Insulation Dry

3. Since the efficiency of cabinet insulation is greatly reduced when it becomes wet, it is of the utmost importance to prevent moisture from accumulating in it. Water usually enters in vapor form from the room air through small cracks or openings in the outer case. It passes through the outer layers of insulation and condenses on reaching the dew point in the insulation next to the cold wall, where it is absorbed. This flow of vapor takes place as soon as the absolute humidity in the insulated space drops below that of the room, thus producing the necessary vapor pressure differential.

The ideal way to keep the insulation dry would be to seal hermetically the walls of the cabinet or the individual pieces of insulation, so as to make them vapor tight, but in practice this is too costly.

The practical way of keeping the insulation dry, where the conditions permit, is to construct the cabinet so that the **resistance to moisture** flowing into the insulation is considerably greater than the resistance to its escaping, so that at no time will the rate of moisture flowing into the insulated space exceed that leaving it. This may be accomplished by sealing the outer case of the cabinet as nearly air tight as is commercially practical, and leaving openings from the insulation space to the food compartment. The vapor pressure difference thus set up causes the water in the insulation, if any is present, to evaporate and the vapor to be deposited on the evaporator surface. **Fig. 3** shows a graphical description of how this takes place.

Unfortunately, all designs do not lend themselves to this simple method. For example, when the storage compartment liner itself is refrigerated, and designed for sub-freezing temperatures, keeping the insulation dry is not so simple. Since it is commercially impractical to seal the insulated space hermetically, the only alternative then is to incorporate a means whereby this accumulated condensate or frost may be periodically removed (see **Fig. 4**). In this manner the insulation may be kept practically dry, despite a few small openings into the insulated space.

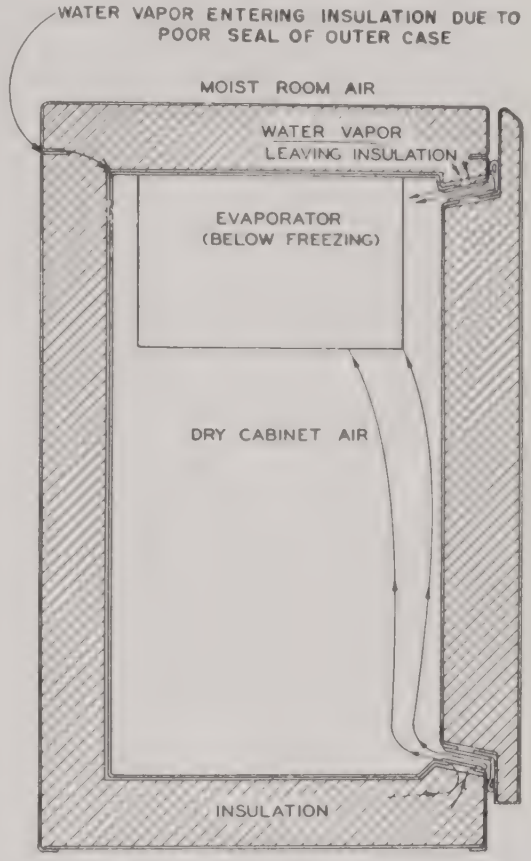


Fig. 3. Diagram of Cabinet Construction Permitting Water in Insulation to Evaporate

Handling Insulation in Production

4. In determining the adaptability of an insulating material to modern production

methods, such properties as the following must be given due consideration: Density, rigidity, flexibility, shrinkage, settling, and effect of handling on the health and comfort of workers (see Chap. 15).

During recent years a major part of the insulating materials used in refrigerator

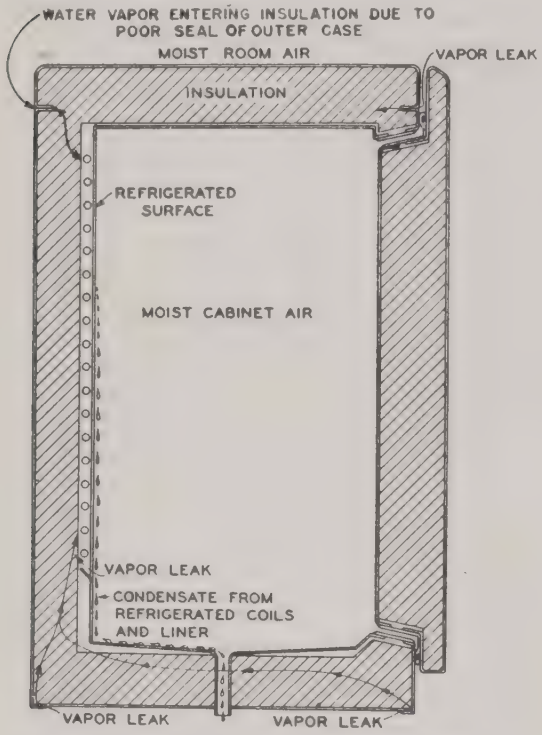


Fig. 4. Cabinet Construction Permitting Removal of Condensate or Frost

cabinets has been of the rigid or semi-rigid type, which required fabricating into slabs or batts to specified drawing dimensions. These types of materials have been very satisfactory as to speed of handling and installation, the major disadvantages being high fabrication and transportation costs, and lack of interchangeability of supply stock, which result in the carrying of a large inventory and a considerable storage space.

Bulk insulating materials suitable for blowing into the cabinet walls offer a possible solution to the problem of lowering the insulation and transportation costs, and simplifying the factory handling and stock supply problems. Such materials are now in use, being shipped in bales and then refluffed and blown into the cabinet walls by the cabinet manufacturer. To do this, special handling and blowing equipment is needed which is likely to be costly.

Insulation **settling** in the cabinet walls in transit has proved a major difficulty with bulk insulating materials. To overcome this, it has been necessary to use higher densities than are otherwise desirable from a cost standpoint. Such factors as fiber sizes, distribution, felting, brittleness, and weight influence the settling.

Testing Insulation and Cabinets

5. Tests which are of value in controlling the thermal quality of the product follow:

a. Hot plate test on insulation, developed by the Joint Committee of the A.S.T.M., A.S.H.V.E., A.S.R.E., and National Research Council Standards on Thermal Conductivity of Materials.
b. Heat leakage test on insulated cabinets, developed by H. W. Eagles, ("Thermal Testing of Refrigerator Cabinets Past and Present," *Refrig. Eng.*, June 1931).
c. No-load performance tests using calibrated test units, as prescribed in NEMA Household Electric Refrigerator Standards.
d. No-load performance tests on production refrigerators, by the same agency.

In addition to the above tests, it is desirable to check the thermal durability of the insulated cabinet, and for this purpose the following **life test** has proved valuable. In a test room at 90 F and 90 to 95% relative humidity, with cabinet air at 43 F, and no load, this performance schedule is followed:

a. One week of no-load performance at start of test.
b. One year or more of combined no-load and door-opening load performance. Four days per week forty 30-second door openings per day; two days per week of no-load performance; and one day per week for defrosting and no-load performance.
c. One final week of no-load performance for comparison with original week of no-load results.

Results of room, evaporator, and cabinet air temperatures, and room relative humidity, together with the power consumption rate and percentage running time of the unit, should be plotted on curve sheets daily. By reference to these curves, it is easy to follow the day-to-day progress of the test and the trends which indicate loss or gain of **overall efficiency.**

A good cabinet design should show no appreciable upward trend in power consumption or running time during a one-year test period.

31. SMALL REFRIGERATING UNITS

THE term mechanical condensing unit is applied to a complete high side of a small refrigeration system, from $\frac{1}{8}$ hp up. It is produced by a score of companies and evolves from both industrial and household machine practice. The refrigerants used are sulfur dioxide, methyl chloride and Freon-12. The single-acting reciprocating type compressor is used almost entirely, for both domestic and commercial applications, although the rotary unit also finds use (**Fig. 1**). Mounting a compressor with its drive and motor, condenser-receiver and controls on a common base has been common up to 60 or 75 hp.

Air-cooled condensers usually are confined to units below 3 hp. In small sizes, two-cylinder compressors predominate. Larger units tend to favor either three cylinders in line, or a four-cylinder V-type. There are compressors with four cylinders in line, but it has been found that such units tend to develop more vibration than the V-type. V-belt drives are common, which may or may not be provided with automatic adjustable motor bases for proper belt tension. In the smaller sizes, it has been found satisfactory to use a multiple-belt drive which gives a very low loading to the belt, and any stretch is kept to a minimum. With such drives there is no need to apply an automatic belt take-up. Compressor speeds range from 300 to 700 rpm, but higher speeds are sometimes used. Motors are of high torque, low starting current and in most instances run at 1,750 rpm.

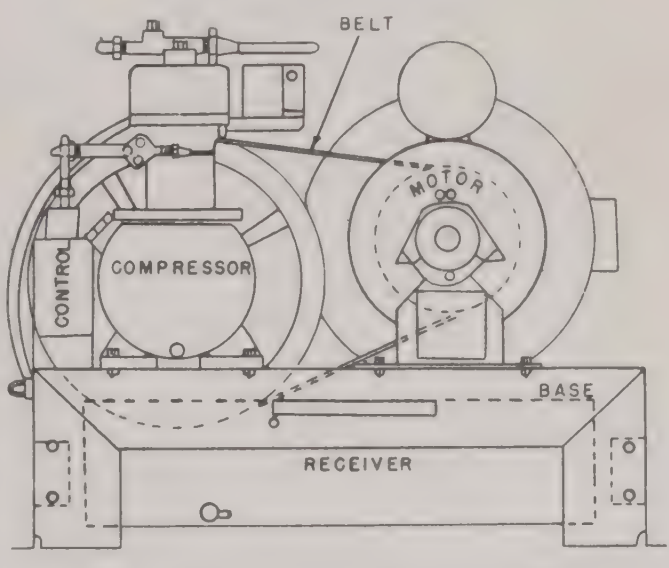


Fig. 1. Typical Commercial Condensing Unit

Domestic Refrigeration Units

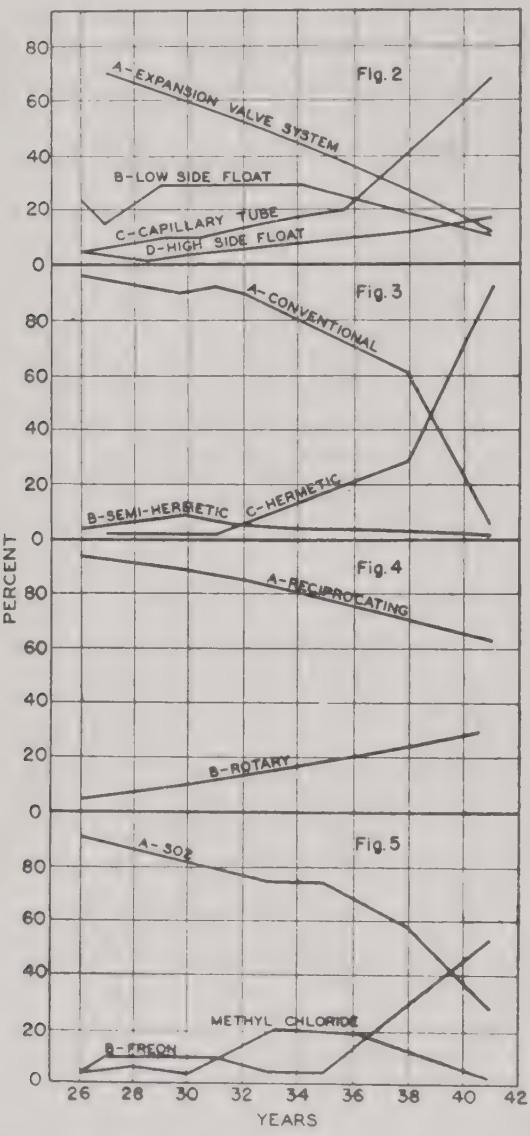
1. Small automatic **self-contained refrigerating systems** as used for domestic purposes can be classified according to the type of control: (a) high-side float; (b) low-side float; (c) expansion valve; and (d) capillary tube (see Chap. 22).

Fig. 2 shows the approximate extent of use of each type, by years. Another classification is those using reciprocating compressors, against those using rotary compressors.

Still another classification is made with regard to sealing—between the conventional system, the semi-hermetic system, and the hermetically sealed system.

The **semi-hermetic system** is a compromise between the conventional and the hermetic systems. The main difference in this unit from the conventional type, is that it operates at the full speed of the motor and both motor and compressor are contained in a gas-tight housing. Provision is made in the semi-hermetic unit for access to valves and pistons so that some repairs can be made in the field. However, the fits and tolerances in both the semi-hermetic and hermetic are such as frequently to make it impossible for the usual field shop

to do a satisfactory job. Units of this type should be returned to the factory for repair.



Figs. 2, 3, 4, 5 (top to bottom). Extent of Use of Various Types of Domestic Refrigeration Units

Fig. 2. Classified according to type of control
Fig. 3. Classified according to sealing
Fig. 4. Reciprocating or rotary compressors
Fig. 5. Classified according to refrigerant

The **hermetic system** has its motor and compressor completely contained within a sealed housing. No repairs can be made in the field, as it is necessary to cut this housing in order to gain access to the interior. Systems using hermetic and semi-hermetic units for domestic refrigerators are fabricated so that the complete high-side and low-side equipment is connected, sealed, dehydrated, charged and tested as a finished assembly before mounting within the refrigerator compartment.

2. Trends in compressor design. The **reciprocating compressor** will give the best overall efficiencies because of the more positive action of its component parts. It has the best volumetric efficiency, the lowest leakage loss per unit of volume handled, and the best power consumption of any other compression means. The reciprocating compressor, however, is at a disadvantage when extremely large displacements are required, because cylinder bores and strokes or speeds then have to be excessively large.

The reciprocating compressor has less leakage because of the increased effective sealing surface between the piston and cylinder, aided by oil as the sealing agent. It also lends itself to manufacturing because the design is adaptable to the usual drilling, boring, honing, milling and broaching operations.

The **rotary compressor** has worked out very satisfactorily for the small domestic assembly because of its fewer running parts and its adaptation to high speed production. It is not so efficient as a reciprocating unit, but if exceptionally large condenser area is used, lower head pressures result and the design is very satisfactory. The rotary compressor takes very little space, can be operated at high speeds and starts very easily.

The conventional reciprocating compressor, driven by a belt at moderate speeds, is usually lubricated by splash lubrication. Semi- or full-hermetic compressors, either rotary or reciprocating, are usually lubricated with a combination of splash and force feed. Some designs use a modified form of gear pump which conveys the oil to the running parts, while other designs use a form of rotary pump on the end of the shaft, dipping into the oil sump and flooding working parts with a bath of oil.

The rotary compressor does not use valves but the reciprocating compressor uses a reed-type valve assembled in a valve plate in the cylinder head. Some types of reciprocating compressors vary from the conventional design; that is, the cylinder is actuated by the crankshaft while the

piston is held stationary. Some types combine the connecting rod and piston by using an oscillating cylinder, and others use the Scotch yoke drive rather than the conventional crank or eccentric.

Two types of rotary compressors are being used at the present time. One type incorporates a rotating eccentric and one blade which is held in the compressor housing. The other type uses a rotating shaft which holds two or four blades, properly spaced, and rotates them in an eccentric housing. The oil performs an important function in maintaining a good seal in all rotary compressors.

Refrigerants

(See also Chap. 8)

3. Sulfur dioxide was once the most popular refrigerant **(Fig. 5),** although giving way to others in recent years. Factors favoring sulfur dioxide are that it is very stable, it simplifies lubrication problems, has good specific volume, effective latent heat, and good operating pressure characteristics. However, due to its irritating odor and corrosiveness when combined with water, its greatest use is in hermetic units.

Methyl chloride operates at higher pressures and has considerably less corrosive action in the presence of moisture, a good operating range, and does not require special compressor designs.

Freon-12 is popular because of its low specific volume which permits a small displacement compressor. When evaporator temperatures are very low (approach −40 F), it is desirable to use **Freon-22,** which has characteristics resembling those of ammonia and which permits the low-pressure designed compressor to develop lower evaporator temperatures. **F-21, F-114** and **methylene chloride** are sometimes used in rotary units. **Isobutane, ethyl chloride** and **methyl formate** refrigerants have been used occasionally. **Ammonia** is used in the absorption machine.

The **lubricant** used in domestic refrigerating machines has been the subject of considerable study. The oils now used are usually of the mineral base and must be chemically pure, with special attention paid the allowable maximum amounts of water and sulfur contents. The viscosity requirements are similar to those for small commercial machines, differing somewhat from the refrigerants used because of the thinning action produced in the crankcase by dilution with the refrigerant. Oil must be heavy enough to provide a good body at higher temperatures and still remain fluid at temperatures at least 10 to 15 deg lower than the lowest temperature likely to occur in the evaporator. Shown below are desirable oil **viscosities** for the three most popular refrigerants, S. U. at 100 F.

Sulfur dioxide	75–100
Freon-12	150–200
Methyl chloride	150–300

4. Controls. The pressure-operated **expansion valve** is popular for domestic refrigeration as it takes up very little room and is easily adjusted. However, it is not easy to make such a valve cheaply without compromising its performance, and for this reason it is not used as much as formerly. This type of valve also presents a hazard in connection with a hermetic system, because the system must be opened to the atmosphere to repair such a valve.

The **low-pressure float** has the advantage of simplicity of construction, operation and manufacture. Also, the float may be part of the evaporator and it is adapted to quantity production.

The **high-pressure float** in its early design was too cumbersome for domestic refrigeration application, but with the growth of the package hermetic system and refinements in design, it has become increasingly popular. To mount the high-side float remote from the cooling coil necessitates the use of an auxiliary refrigerant control to prevent the liquid line from frosting between the control and the cooling coil. Two of these auxiliary controls are in common use, namely, a **capillary tube** or weighted **check valve.** These secondary throttling devices permit the installation of the liquid line at room temperatures without the possibility of frosting or sweating. With the high-pressure float system, very accurate control of the refrigerant charge must be maintained; otherwise a very slight overcharging will cause excessive head pressures.

The capillary tube control system has

many favorable features, particularly with hermetically sealed domestic units, because of simplicity of construction, low cost, absence of moving parts and the permitting of balanced pressures for ease in starting. When capillary tubes are used for refrigerant control, the general practice is to use an accumulator which prevents liquid spillover to the compressor and thus eliminates difficulty in starting of the condensing unit. No receiver is used with a capillary system. **Table 1** shows the general practice of capillary tube size for various capacities.

Table 1. Capillary Tube Data for Various Conditions (Freon-12)

hp	Btu/hr	Suction temp F	Size of accumulator, cu in.	Capillary tube	
				Bore, in.	Length, ft
1/8	800	18	8	.032	6
1/8	975	18	8	.040	17
1/5	1,550	26	8	.040	6
1/5	1,550	26	8	.050	9½
				.040	4
1/4	2,125	26	12	.050	6½
1/4	2,125	26	12	.040	3
1/3	2,800	26	12	.050	5
1/5	800	0	8	.040	15
1/4	1,250	0	12	.040	12
1/3	1,620	0	12	.040	10
1/3	1,620	0	12	.050	12

5. Where quietness is essential, as in the domestic unit, the **capacitor-type motor** has proved most acceptable. Its characteristics lend it to application in hermetic-type compressors. The cooling of the motor is usually by fins on the housing cover, by circulating oil over the motor windings, or by permitting the cool return suction gas to come in contact with the windings. Considerable research was expended to discover a satisfactory insulation for the motor windings exposed to the refrigerant. Neoprene-coated wire, plain copper wire insulated with bleached long-fiber cotton, and a synthetic enamel known as Formex, have proved most successful. The assembly must be very thoroughly dehydrated and baked under a vacuum.

All domestic machines use a **motor control** to permit the automatic regulation of temperature. Two types have been used, the pressure-actuated and the thermostatic, the latter being more popular because of more accurate regulation. The newer controls, in addition to their motor control function, usually incorporate a manual switch, a calibrated temperature adjustment, an overload cutout and a defrosting switch.

6. The early domestic **evaporators** were made of copper tubes spun around a tray compartment, for direct expansion or flooded control. The trend was later toward pressed steel parts, hydrogen brazed. This design made it quite simple to fabricate serpentine cups in steel, so that when the parts were pressed together, they formed passages for the refrigerant. While this construction permitted fabrication in punch presses, there was difficulty with rusting until porcelain coatings and stainless steel materials were specified.

In design, the humidities and temperatures necessary to handle stored foods must be kept in mind. Generally, the temperature maintained in the evaporator is 18 F and in the refrigerator from 38 to 45 F, with relative humidity of from 70 to 80%. Since so much is expected of a domestic refrigerator, from making ice cubes and ice cream to preserving various foods, some dehydration is unavoidable. Special compartments to maintain proper temperatures and humidities are used.

All refrigerators are thermostatically controlled, either by having the temperature bulb of the thermostat strapped to the evaporator or to one or the other of the compartments in the box. The **thermostat** is provided with means for regulation of box temperature above or below the normal setting. Arrangements are made to defrost the evaporator periodically by shutting off the compressor. Some controls automatically restart the unit after defrosting for about an hour or two.

7. The newest model refrigerators have space where **frozen foods** may be stored at temperature around 0 F. This imposes still further temperature regulations upon the condensing unit. If a single condensing unit is to maintain both frozen food and medium temperatures, obviously additional controls will have to be provided

to regulate these diverse requirements. The condensing unit will have to operate at the lowest temperature and be fed and maintained at suitable medium temperatures for other services in order not to reduce humidity.

Some dual-service refrigerators utilize a secondary refrigerant circuit to develop medium temperatures which is claimed to simplify the controls and give better regulation. Many of the new refrigerators have coils surrounding the inner liner so the usual evaporator does not appear. The liner type of evaporator tends to improve relative humidity because of less air movement and reduced temperature differential between air and refrigerant.

It is likely that many dual-service refrigerators having space for frozen foods will have two separate and distinct condensing units. One will take its usual 40 F temporary storage compartment with ice cubes, while a separate unit will handle the longer storage 0 F frozen-food compartment.

Commercial Compressor Performance

8. The theoretical **performance** of a compressor in a condensing unit is the same as that of any compressor, as discussed in Chaps. 2 and 23. It is possible to predict performance with compressors for sulfur dioxide, Freon-12 and methyl chloride, as compared with air or ammonia. Understanding of compressor performance can be had by reference to the $P-v$ relationship of the indicator card. The theoretical **volumetric efficiency** of a compressor is discussed in Section 2 of Chap. 2.

It is obvious that if n is small, volumetric efficiency will be lower for the same per cent clearance volume. The value of the exponent n for several refrigerant gases is given in **Table 2.** It is seen that low-pressure refrigerants have rather low values of n. It can be expected that the volumetric efficiencies will be of a lower order than air or ammonia. Careful attention must be given to the reduction of excess cylinder clearances.

The theoretical **power** required to compress a cubic foot of gas is expressed as follows:

Table 2. Value of Specific Heat Ratio
($C_p/C_v=n$) for some refrigerant gases

Air	1.40
Ammonia	1.31
Carbon dioxide	1.3
Ethyl chloride	1.19
Freon-12	1.13
Methyl chloride	1.20
N-Butane	1.11
Propane	1.13
Sulfur dioxide	1.29

Theoretical adiabatic horsepower

$$= \frac{144}{33{,}000} \frac{n}{n-1} P_1 v_1 \left[\left(\frac{P_2}{P_1} \right)^{(n-1)/n} - 1 \right] \quad (1)$$

Where,

P_1 = psia suction pressure

v_1 = cu ft of gas entering compressor section

If the formula is based on one cubic foot, v_1 is eliminated, and the theoretical adiabatic horsepower becomes

$$\text{T.H.P.} = P_1 \left\{ \frac{144}{33{,}000} \frac{n}{n-1} [(R)^{(n-1)/n} - 1 \right\} \quad (2)$$

T.H.P. $= P_1$ (hp constant).

Values of horsepower constant are plotted in **Fig. 6,** for various compression and specific heat ratios. To obtain the adiabatic horsepower per cu ft, it is necessary to multiply the horsepower constant by the absolute suction pressure.

9. Experience indicates that the theoretical volumetric efficiency will give only a rough approximation of the true value because of many factors which influence it and which are difficult to measure or predict. Further tests indicate that leakage plays such an important part and is so difficult to measure that any extensive practical analysis must rest on actual test. The effect of cylinder superheating has been discussed by Crampton.[1] In design, the probable volumetric efficiency can be calculated. The actual volumetric efficiency will be as much as 15% less. On open-type compressors, the displacement can be estimated and arrangements should be made to adjust the overall capacity by increasing speed as developed by actual tests. (See Chap. 2 and 23.)

A composite curve of volumetric efficien-

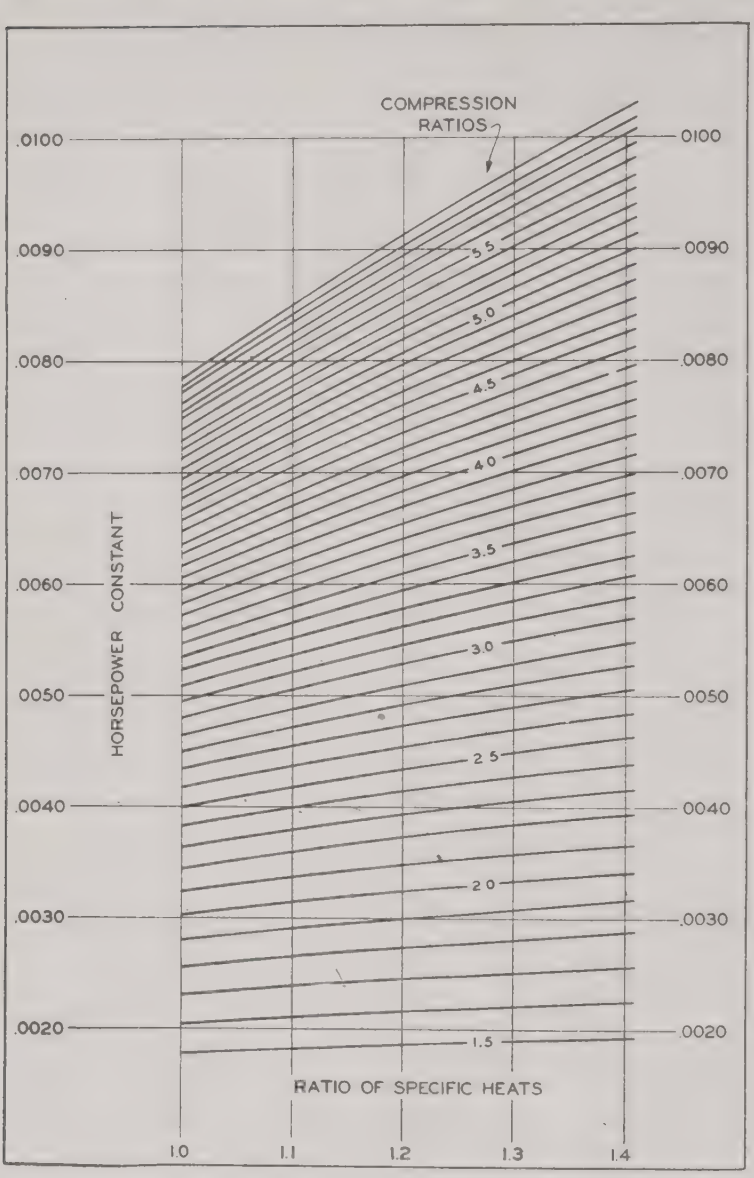


Fig. 6. Horsepower Constant for Various Compression and Specific Heat Ratios

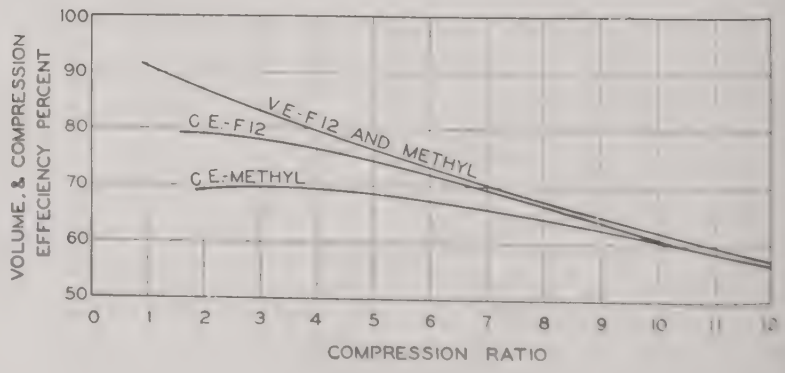


Fig. 7. Volumetric (V.E.) and Compression (C.E.) Efficiencies of Compressors

cies of all compressors would be hard to define, but a representative curve which gives very close results is shown in **Fig. 7,** for Freon-12 and methyl chloride. Values for sulfur dioxide would approximate the methyl chloride curve. An important factor in testing for volumetric efficiency is superheating of the suction gas, which will be considered in more detail later.

The power required to drive the compressor is found by the following:

$$\text{B.H.P.} = \frac{\text{T.H.P.} \times \text{D} \times \text{V.E.}}{\text{C.E.} \times \text{M.E.}} \qquad (3)$$

Where

B.H.P. = brake horsepower

T.H.P. = theoretical adiabatic horsepower per cu ft

D. = displacement of compressor, cfm

V.E. = volumetric efficiency, $\frac{\text{per cent}}{100}$

C.E. = compression efficiency, $\frac{\text{per cent}}{100}$

M.E. = mechanical efficiency, $\frac{\text{per cent}}{100}$

Representative values of compression efficiency are shown in Fig. 7, for Freon-12 and methyl chloride. Mechanical efficiency should not be constant, but should depend upon the actual loading of the compressor at the conditions. It would be preferable to find the friction horsepower first, or estimate it. If the compressor were designed for a 5-hp maximum load, the friction horsepower could be expected to be about 0.40, based on a peak load mechanical efficiency of 92%. If the compressor were only loaded to 3 hp, the brake horsepower should be 0.40 and the mechanical efficiency would be only 88%. The friction horsepower will vary almost directly with rpm.

Since practically all condensing units must start against pressure, it is desirable to estimate the starting torque required in lb ft. Since the following equation is for estimating purposes, it neglects the friction of the compressor, the additional torque required to force discharge gas out of the cylinder, and the fact that the tangential force at the crankpin is not always equal to the normal force at the piston.

$$\frac{(P_2 - P_1)As}{24D/d} \qquad (5)$$

or

$$\frac{(P_2 - P_1)As}{24\frac{r_1}{r_2}}$$

Where

P_2 = discharge pressure, psi
P_1 = suction pressure, psia
A = area of cylinder, sq in. (Use only one area for a two- or four-cylinder compressor.)
s = stroke of compressor, in.
D = diameter of compressor pulley, in.
d = diameter of motor pulley, in.
r_1 = motor rpm
r_2 = compressor rpm

This formula shows that the direct-connected compressor will need more motor torque to start it than a belted arrangement. The starting torque available in a motor is obtained as follows:

$$= \frac{5{,}250 \text{ hp}}{\text{rpm}} \frac{B'}{100} \text{ lb ft} \qquad (6)$$

Where

hp = horsepower rating of motor
rpm = full load speed of motor
B' = percentage starting torque = 400 to 600% for single-phase motor, or 225 to 325% for polyphase.

Because of the difficulty of obtaining indicator cards for small compressors, breakdown efficiencies are seldom attempted. Actual tests are essential and the comparisons are usually made on the basis of capacities in terms of Btu per hr capacity per watt hr of motor input. Representative curves of the performance of a $\frac{1}{2}$-hp air-cooled condensing unit are shown in **Fig. 8.** Typical manufacturers' performance specifications are shown in **Table 3.** Since, at low suction pressures, the compressor needs more displacement to load the motor, it is commercial practice to split the

operating temperature range into three sections, usually designated as low, medium or high temperature, and operate the compressor at three different speeds, corresponding to each range. Sometimes, a larger compressor is necessary to satisfy the low-temperature ranges. This makes possible the most effective use of the available motor power.

The hermetic compressor, because of inability to adjust its speeds to such operating ranges, cannot produce as much capacity at low temperatures as the belt-driven unit. This is because the hermetic unit is normally designed for high suction pressures and motors are applied for that rating. This influences design in many ways, as explained previously; that is, when the capillary tube is used, balanced pressures at the start result in reduced starting requirements.

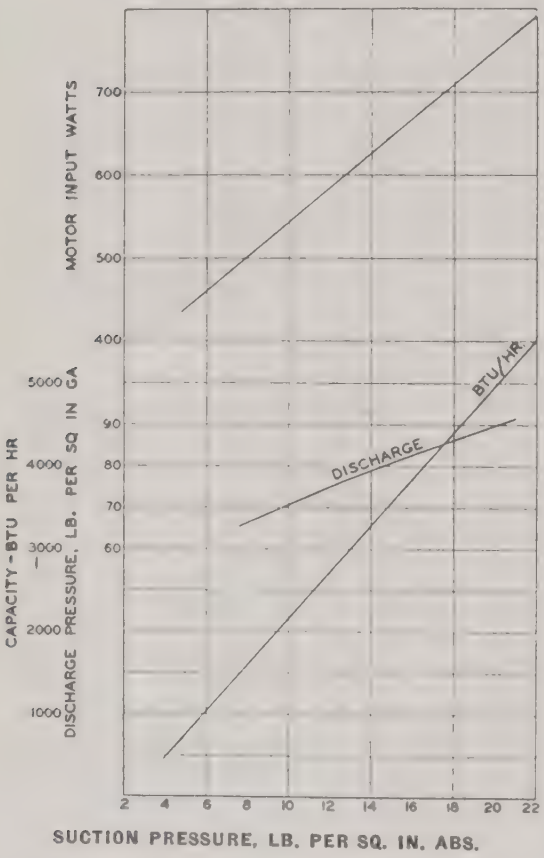


Fig. 8. Air-Cooled Sulfur Dioxide ½-hp Compressor, 2-cylinder, 2⅛ in. bore × 1⅝ in. stroke, 510 rpm, 90° room

Table 3. Typical Specifications, 1-hp Air-Cooled Condensing Unit

(2-cyl, 2½-in. bore × 1¼-in. stroke, 1-hp motor, 2 V-belts; overall dimensions 32 × 24 × 21 in. high.)

Condenser	**Continuous fin**
Receiver capacity, lb, methyl chloride	**17**
Receiver capacity, lb, Freon-12	**25**
Liquid line fitting	**⅜ in. S.A.E.**
Suction line fitting	**⅝ in. S.A.E.**
Oil charge, pints	**3¾**
Weight, lb	**294**

Capacity with methyl chloride, 90 F ambient

Evap temp, F	rpm	Capacity Btu/hr
−40	640	1,900
−30	640	2,700
−20	640	4,050
−10	640	5,500
0	640	7,390
−10	530	4,650
0	530	6,050
10	530	7,800
20	530	9,750

Capacity with Freon-12

Evap temp, F	rpm	Capacity Btu/hr
−40	570	2,775
−30	570	3,775
−20	570	4,900
−10	570	6,350
0	570	8,100
−10	500	5,200
—	500	6,700
10	500	8,300
20	500	10,200
30	365	9,225
40	365	11,050
50	365	13,100

Compressor Design

10. Low-pressure refrigerants permit less restrictions regarding materials than ammonia. All forms of steel, brass, copper, zinc, aluminum and die castings can be used with Freon-12 or sulfur dioxide. The same is true of methyl chloride with the exception of aluminum. There is a very pronounced chemical reaction between aluminum and methyl chloride and, therefore, these two materials should never be used together. Die castings in which a certain percentage of aluminum is used have been a part of methyl chloride systems with apparently no ill effects.[2,3]

To design a compressor for low-pressure

refrigerants requires a knowledge not only of the gas handled and its characteristics, but the way in which the actual system functions. It is not only necessary to provide the proper cylinder proportions and valve openings, but also the proper arrangement for returning gas and oil to the cylinder.

11. A sulfur dioxide compressor will have certain differences from a Freon-12 or methyl chloride compressor, while the differences between Freon-12 and methyl chloride will not be apparent. These differences center about the pressures involved and the affinity of the gas for the oil used. Sulfur dioxide operates at below atmospheric suction pressures in most instances and is very critical regarding oil pumping. It is desirable to have some form of catch chamber to separate the oil slugs and return them to the crankcase, before the gas is admitted to the cylinder. It is usually desirable to operate the sulfur dioxide compressor at a somewhat slower speed, in order to give added protection against damage to the compressor. While it is true that the hermetic compressor operates on sulfur dioxide at high speeds, this application is mostly restricted to the domestic unit, where small capacities can be more readily handled. In addition, large separating chambers are provided to give the necessary protection.

Freon-12 and methyl chloride compressors must also be provided with arrangements to permit all oil that returns with the suction gas to be drained to the crankcase. Experience indicates that the requirement for separator chambers on the suction is not so critical as with sulfur dioxide. This is apparently due to the complete mixing of the oil with the gas, permitting a more uniform and constant return at all times.

12. The design of **crankshaft** and **bearings** follows the usual compressor practice. It is usual to consider 200 psi pressure difference as the basis for Freon-12 and methyl chloride piston loads. If condensing units are to pass the Underwriters Laboratories' tests, the crankcases must be designed for hydraulic test pressures of five times the refrigerant pressure at 70 F, and the cylinders, heads, and all high-side fittings at five times the setting of any pressure limiting device. The usual practice is to test both low and high sides at least to 195 psi pressure before shipment.

13. It is common practice in many designs to harden the **shafts** to a Scale C Rockwell index of about 50 to 60. The hardened shaft serves as a riding surface for the seal and provides a superior wearing surface for bearings, especially where leaded bronze bearings are used. Experience indicates the inadvisability of using a crankshaft having any chromium content if a shaft shoulder is used as a sealing surface. Chromium specks in the steel cause an imperfect surface for the seal. Steel of Class S.A.E. X3140 should be used only where seals do not ride directly on the shaft. S.A.E. X1315 steels properly case-hardened are better for shaft riding seal designs.

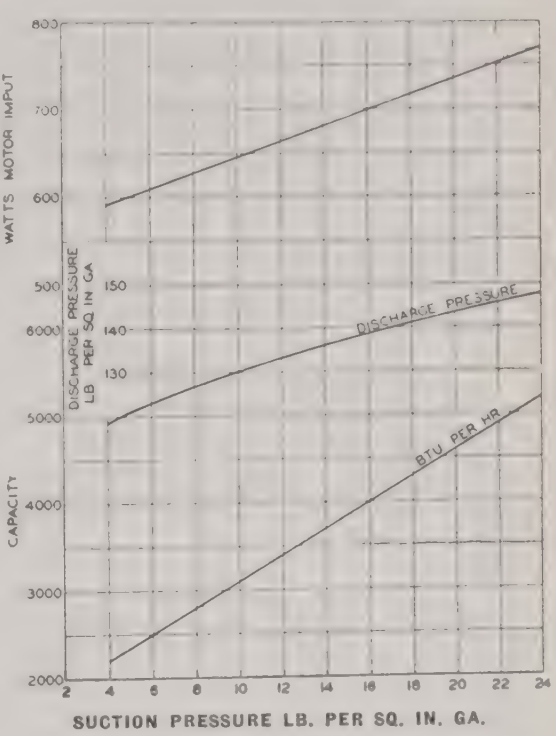


Fig. 8a. Air Cooled Freon-12, ½-hp Compressor, 2-cylinder, 1-13/16 in. bore×1⅜ in. stroke, 525 rpm, 90° room

14. Diaphragm and bellows **seals** have been most popular until recent years. A construction which eliminates the bellows and instead floats the seal nose in a syn-

thetic rubber shaft sealing collar, while the nose bears against a stationary seal plate, has proved very satisfactory and is gaining wider acceptance. Seal noses have been

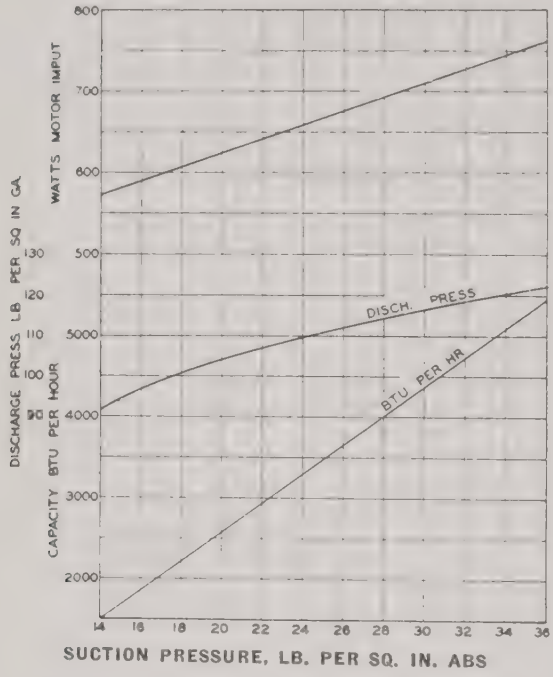


Fig. 8b. Air Cooled Methyl Chloride ½-hp Compressor, 2-cylinder, 1-13/16 in. bore × 1⅜ in. stroke, 545 rpm, 90° room

made of special bronzes and metal impregnated carbon. In seal designs, when the shaft diameter exceeds 2 in., it is advisable to use a multiple-spring design to maintain contact of seal against the shaft. This is especially true of the type of seal which comprises a separate hardened or alloy cast iron nose-piece, sealed to the shaft with a Neoprene or synthetic rubber gasket. This type tends to tip or cock, unless the nose piece is compressed evenly.

Engineers are varied in their opinions regarding a one-piece crankcase and cylinder. Properly tooled, a good shop can handle either effectively. There is no difficulty in securing alignment with the divided construction. The cast iron used must be dense and close-grained, preferably with a nickel content of from 1 to 2%.

It is not advisable to provide eccentric drives for strokes greater than 4 in. Static friction of a larger diameter eccentric increases the starting torque requirements.

15. Cast iron **pistons** are used in most designs since they permit extremely close fits and give good efficiencies. In small compressors, leakage may become a large percentage of the total gas pumped and close tolerances must be observed, with all pistons and cylinders ground and honed to mirror finishes. The usual piston and cylinder clearance is 0.0003 in. per in. of cylinder bore. There is no advantage in using rings on pistons smaller than 2-in. bore. The plug type, no-ring piston is common practice in domestic and commercial units of ½ hp and smaller.

Piston rings, where used, are generally fitted between top of piston and wrist pin, according to automotive practice. Two compression rings and one oil ring are used, although in Freon-12 and methyl chloride units an oil ring is hardly needed, as oil flow through the system is encouraged. The trunk-type piston design admits suction gas through the center of the piston, and may have an additional compression or combination compression and oil wiper ring on the piston skirt. The trunk-type piston design is especially useful in high-speed compressors, as placing the valve in the piston gives greater space for compressor valves, permitting lower valve velocities. Where moderate speeds are used, suction and discharge valves can be located in a common head plate and this construction provides certain desirable simplifications in servicing.

16. For small compressors, high volumetric efficiencies can only be obtained by small re-expansion volume. **Valve velocities** for both suction and discharge valve at about 3,000 fpm should not be exceeded, especially for Freon-12 units. For estimating purposes the velocity calculation is

$$\frac{2S \times \text{rpm} \times \text{piston area}}{12 \times \text{area of valve opening}}$$

where S = stroke, in.

The actual valve velocity must be based on the actual percentage of the stroke during which time the valve is open. Ingenious designs are required to get these velocities at minimum re-expansion volume. The reed valve, **Fig. 9,** works well on small units.

In larger compressors, the ring-plate or

disc valve gives performance and durability which cannot be obtained with the reed type, where the suspension between supports becomes too great **(Fig. 10).** Swedish steel gives best durability and grain structure for valve elements. Utilizing the usual bending formula, for disc and flat plates and for flat springs as beams, it is advisable to limit the allowable working stress to 50,000 psi. A good practice to follow in all valve design is to prevent the reed, disc, or plate from impinging on any backing springs, either flat or coiled, which form a part of the valve assembly. A stop discharge velocities as low as those of the suction.

Compressors larger than 2 hp require ample gas space above the discharge valves, to iron out violent pulsations. Unless this is done, valves suffer reverse flexes, causing early failure, and compressor efficiencies will be affected. In shell condensers, pulsations are absorbed within the large volume, but with evaporative condensers and long discharge lines, an external expansion chamber must be located at the compressor outlet. An oil separator often serves this purpose.

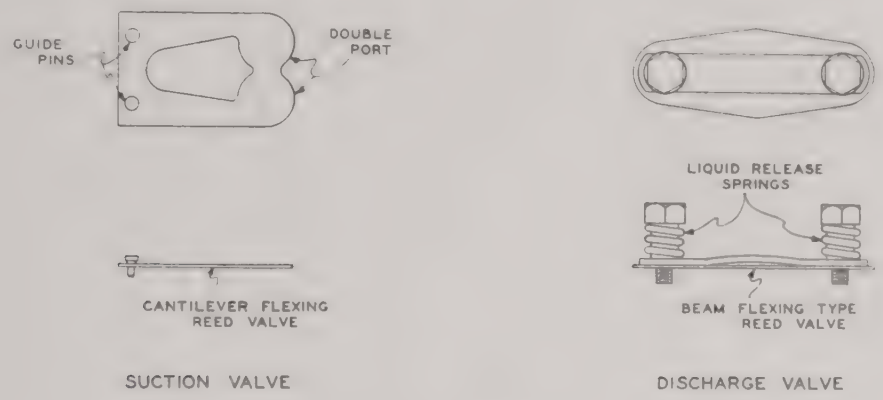


Fig. 9. Reed Valves

to prevent a curved spring from being compressed flat, or a coil spring being compressed until it meets an adjacent coil, should be provided.

Some designers provide a liquid relief in the discharge valve, opening when liquid slugs enter the cylinder.

17. Streamlining passages or ports, or placing fins on the heads of the cylinders, seems to make negligible changes in efficiencies in sizes below 1 hp; in fact, the effects are only slightly evident in units as large as 50 hp. Even water-jacketed heads are being discarded. Gas passages through the heads should have about the same or lower velocities as those through the compressor valves. In some instances, assumptions have been made that discharge valve and port velocities can be higher than those of the suction valve, because the discharge period is shorter than that of the suction period. When consideration is given to greater friction because of the higher pressure, it is desirable to maintain

18. Main **bearings** can be of any well-known material, such as cast iron, leaded bronze or steel shell Babbitt. The favored bearing is leaded bronze, as it is non-scoring. Bearing clearances are approximately 2.5 thousandths per inch of shaft diameter. Needle, roller, or ball bearings have been used very successfully in main bearings. Needle bearings are often selected for wrist pin bearings. Such types, however, are somewhat noisy, and because of the usual requirement for quietness, have not been widely applied to condensing units in sizes below 15 hp.

19. Connecting rods are of the conventional type and have been specified either as forged or cast iron construction. The cast iron should have certain malleable qualities with sufficient ductility to absorb shock. Eccentric straps, where used, are generally cast completely of a bearing bronze, making it unnecessary to add bushings at the bearings. Wrist pins are hardened and ground, usually of a case-

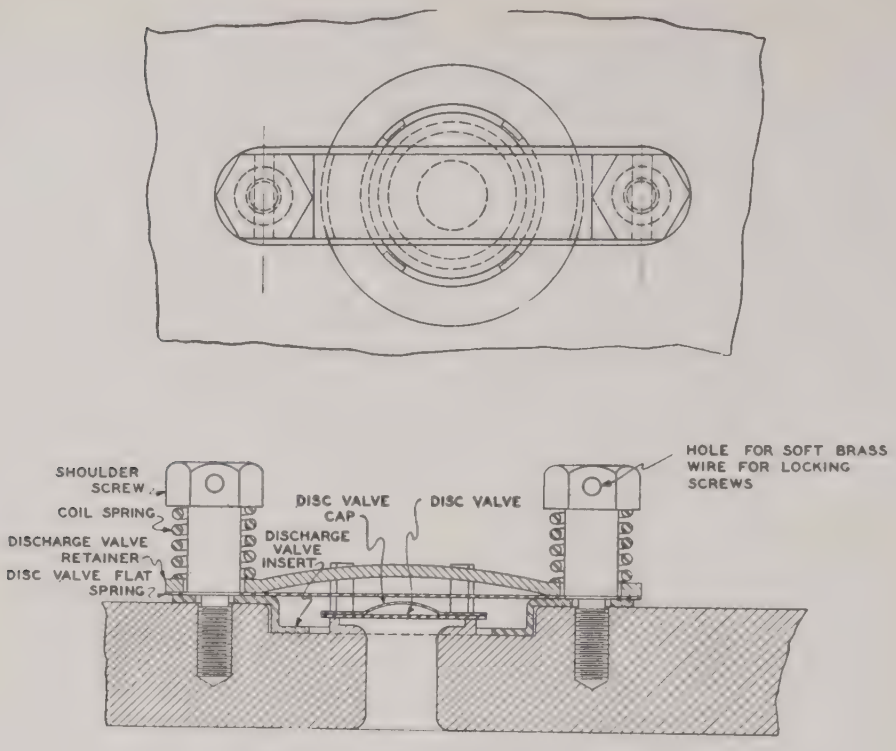


Fig. 10. Discharge Valve Assembly

hardened material, such as S.A.E. X1315 and then hardened to a Rockwell Scale C—50–60.

Recent research and experience indicate that to secure least wear between rubbing or sliding surfaces, there should be a Rockwell hardness difference between the two parts of at least 50 points.

20. Lubrication of small compressors up to about 25 hp is handled very satisfactorily by splash lubrication, although some manufacturers use force feed even in very small sizes. There is no indication that any reduction in oil pumping or friction horsepower occurs where force feed is used. If speeds are increased beyond 800 to 900 rpm, force feed is desirable because it eliminates violent splashing. Oil pumps must be carefully fitted, in order not to lose prime if gas enters the pump. Gear pumps are favored. In small sizes up to 10 gpm they can be connected to the end of the shaft, as ample lift is developed by a good gear pump to pick up the suction between the base of the crankcase and the center line of the shaft. A strainer should be inserted in the suction line to keep out foreign substances from the pump and bearings.

In designing a cylinder, some provision must be made to return oil from the suction port to the crankcase. As there is always leakage of gas upward from the crankcase to the port, the velocity must be low in order that oil may drop against the upward gas flow. At least 5% of the

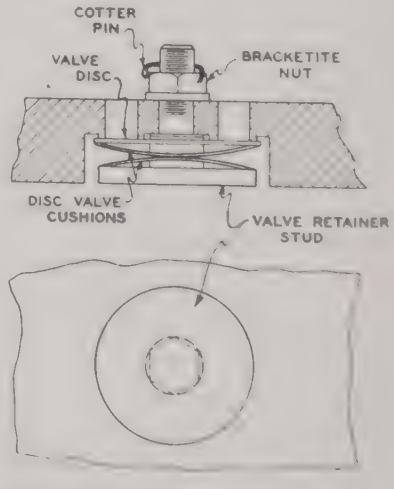


Fig. 10a. Suction Valve Assembly

total piston displacement is leakage. This must escape through the return oil opening at a velocity not to exceed 200 fpm. A separating chamber can be built into the cylinder to help separate oil from the gas, before it enters the suction port.

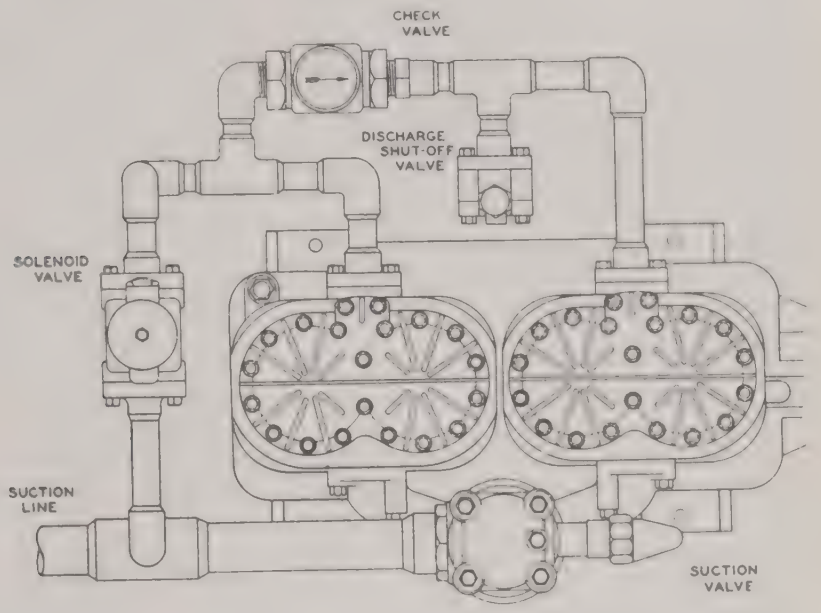


Fig. 11. Cylinder Cutout Control Piping

In many designs, a check valve is inserted at the bottom of the oil return port to prevent a surge of crankcase oil entering the suction. A bypass must be provided in this check valve, which is always open, to permit the check valve to open wide after the oil surge has passed. Otherwise, the check valve would close tightly. When a separating chamber is used, the oil surge is trapped before it can enter the suction port, and a check valve is thus not essential.

21. Regulation of compressor **capacity** can be accomplished in three ways:

1. By throttling suction
2. By bypassing cylinders
   a. Using an external solenoid bypass valve
   b. By holding open the suction valves
3. By variable speed motors.

Throttling the suction is inadvisable, because pulling down the suction pressure causes crankcase oil to froth and be pumped from the compressor. If violent enough, it can break valves or bend connecting rods. In addition, operation at reduced suction pressure is uneconomical in power consumption. Therefore, the first method is seldom used, but in combination with the second gives a rather close regulation, reducing the hazard of throttling.

The **solenoid** is often connected as shown in **Fig. 11.** Care must be exercised to mix the hot bypass with cool return gas, else excessive cylinder temperatures will be built up in the inoperative cylinder. This method is used to give 50% capacity reduction. The check valve should have a discharge area equal to that of the compressor discharge valve. It is a good plan to use the same type of valve in the check as that in the compressor, it being less noisy than a swing check.

Power requirements at 50% capacity will run from 65 to 90% of the full load requirements, depending on the free passage provided through the bypass. Method 2b eliminates the need for external bypass piping and a check valve, as shown in **Fig. 12.** There are other complexities, however. A conventionally designed suction valve must be provided to permit attachment of unloading fingers. These fingers are actuated by a plunger, and the pressure to this plunger is controlled by a three-way solenoid valve, actuated by a

temperature or pressure switch. The cylinder operates cooler than with method 2a and can be applied on a four-cylinder compressor to 75% unloading with only one cylinder operating. There must be at least one cylinder in operation all the time, in order to provide condenser pressure to actuate the unloader plungers. One danger is that the unloader fingers snap down on the suction valve and add stresses to an already overworked part. The force of the blow has also to be absorbed by whatever backing retainer holds the suction valve in position. Power requirements will be about the same as those for method 2a.

The **variable speed motor** is the most efficient means of varying capacity. Usually a two-speed motor, giving 50% capacity reduction, is sufficient for most applications. Power requirements for 50% capacity will be from 55 to 60% of the full load.

In all these unloading methods, to insure an adequate return of oil to the compressor at all times there must be no traps in the evaporator or the suction line. Gas velocities in the suction line at reduced capacity must be above 1,000 fpm.

Commercial Motors

22. The majority of **single-phase motors** used on condensing units are of the repulsion-induction type. This motor has low current inrush per horsepower and extremely high starting torque. These factors favor this design although there has been a trend in some instances toward the capacitor-start induction-run motor. Inconclusive service records show that capacitor motors have lower maintenance cost. As for efficiencies, equal performance can be expected. A capacitor motor larger than 1 hp draws an extremely heavy starting current, which may affect contact points of pressure and temperature controls (see Chap. 20).

Polyphase motors are of the double squirrel-cage type, having high starting torque, with low starting current inrush. Two-speed motors are usually of the constant torque, single-winding or consequent pole type, 1,800 to 900 rpm or 1,200 to 600 rpm. Dc motors should be compound wound. If specified with inter-poles, better commutation is assured.

23. Controls. Single-phase motors, up to and including 1 hp, are usually furnished with built-in overload elements. In larger sizes, or on polyphase motors, across-the-line starters are provided with overload and under-voltage protection, by the use of thermal elements. Built-in over-load protection is of the automatic reset type, but starters have to be reset by hand.

It is not desirable to furnish **automatic resets** in a starter without some form of temperature device buried in the windings of the motor, because the overload elements might reset while the motor is still at a dangerous temperature. It is possible, however, to get motors in very large sizes with built-in thermal elements controlling external starters, thereby obtaining the desirable features of automatic reset.

The usual practice in refrigeration is to provide **overload elements** which will trip after a period of two hours, at 140% of the full load current of the motor, when the motor and starter is of 5 hp and below, and operating in an ambient temperature of 40 C (104 F). For motors larger than 5 hp, overloads are selected at 125%.

Automatic operation is usually accom-

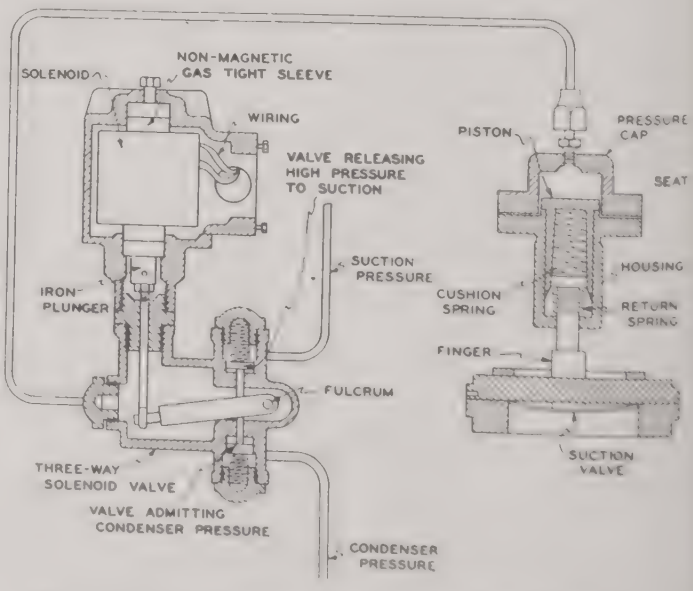


Fig. 12. Use of Solenoid Valve without Bypass Piping

plished by bellows or diaphragm-type switches. Water-cooled condensing units are supplied with a dual control, one part of which, the high pressure cutout, will stop the motor if the condenser pressure becomes excessive. This may occur if there is any lack or stoppage of water. The second part of the control, low-pressure or temperature cutout, starts or stops the motor, depending on the condition at the evaporator.

The air-cooled condensing unit under 3 hp is seldom furnished with a high-pressure cutout because air is always present to prevent excessive head pressures. The overload protection provided in the motor is a sufficient safety limiting device; because of fouling of the condenser, there will be excessive current drawn from the line, eventually stopping the motor. With this design, a straight pressure or temperature control regulates the motor operation.

Table 4. Compressor Ventilation Requirements

Compressor hp	Natural circulation, cu ft room volume needed		Forced circulation, cfm fresh air entering room	
	Air-cooled	Water-cooled	Air-cooled	Water-cooled
¼	200		100	
⅓	400	190	145	130
½	600	260	220	175
¾	800	360	330	275
1	1,000	475	440	300
1½	1,500	665	660	425
2	2,000	850	880	550
3	3,000	1,100	1,320	780
5		1,300		1,100
7½		1,450		1,400
10		1,600		1,700
15		1,850		2,250
20		2,000		2,750
25		2,150		3,150
30		2,250		3,500
40		2,500		4,250
50		2,750		5,000

In room temperatures below 20 F, it may be necessary to restrict air flow to the condenser. This should only be done if a dual control is specified, in order to protect against too high head pressure. If the head pressure is too low, insufficient pressure difference is developed to force liquid through the expansion valve, unless the orifice is made excessively large. With low room temperatures a pressure control cannot operate, since the suction pressure is below that corresponding to room temperature. In such instances, it is desirable to use a temperature control to maintain evaporator temperatures.

Air-cooled units must be furnished with sufficient **ventilation** to maintain their efficiencies. **Table 4** indicates the minimum size of room in which air or water-cooled condensing units should be installed. When smaller rooms are used, some form of forced ventilation should be provided.

24. Tests indicate that dry gas is needed at the compressor suction to obtain best efficiencies. (Refer to Chap. 18, for further data on dehydration.) This is true of all gases with low heat of vaporization, as particles of liquid on striking the cylinder are flashed into gas, restricting the entrance of further gas. **Fig. 13** shows that at least 30 to 35 deg of superheat are needed to assure all liquid entrainment being eliminated. By using a heat exchanger, it is possible to obtain the total temperature at the compressor in the neighborhood of 65 F.

Heat Exchangers

25. The **surface area** required in a heat exchanger for a refrigerating system can be estimated by the following formula:

$$\frac{Q}{(T_1 - T_s)25} \tag{8}$$

Where

Q = capacity of the condensing unit $\times 0.30$, Btu per hr

T_1 = temperature of liquid entering the exchanger, deg F

T_s = temperature of suction gas entering the exchanger, deg F

Care is to be exercised that the velocity of gas through the exchanger is not less than 1,200 fpm. No low points or traps are desirable, as oil may collect and suddenly surge into the compressor. In systems of ¼ and ⅓ hp, soldering the suction

and liquid lines together for some length makes an effective exchanger.[6]

Automatic Oil Separators

26. The purpose of an oil separator is to prevent oil from entering the evaporator. Oil separators are not generally used on small units as the compressor is designed to pump small percentages of oil with the refrigerant. The effects in the evaporator are shown in **Tables 5** and **6.** When the velocity of gases passing through a coil is more than 1,200 fpm, oil will usually return to the compressor crankcase and not greatly affect the capacity.

In flooded evaporators of large internal volume, velocity is far below this requirement. Oil is not returned to the compressor, but loads up the evaporator. In such applications, it is essential that an automatic oil separator be used.

When oil accumulates in a refrigeration system, the lower the refrigerant temperature, the greater effect it has on the evaporative capacity. This is shown in Table 6 by the rise in refrigerant temperature over that of pure Freon-12 refrigerant. Oil circulation in the compressor proper, due to better sealing of the suction and discharge valves, pistons and other moving parts, reduces kw input and increases the volumetric efficiency as indicated by referring to **Fig. 14.**

Condensers

27. Air-cooled condensers are generally made of copper tube and copper fins, copper tube and steel fins, or steel tube and steel fins. Steel fins and tubes are recommended because the price is less than that of copper, and the likelihood of electrolysis is eliminated.

Fins are usually made of .006 to .015 in. steel or copper, sufficiently kinked for strength and generally set five to seven per inch, depending on the spacing of the tube and fin thickness.

The fins are forced on the tubes and properly bonded by dipping in a bath of lead or tin, to make a direct bond to the tube. The outside surface is either tinned or painted with some pigment paint, suitable for temperatures up to 250 F. Painting does not affect heat transfer.

Table 7 shows the Btu heat removal required per ton of refrigeration, when rated in accordance with the ASRE method using 65 deg suction vapor temperature.

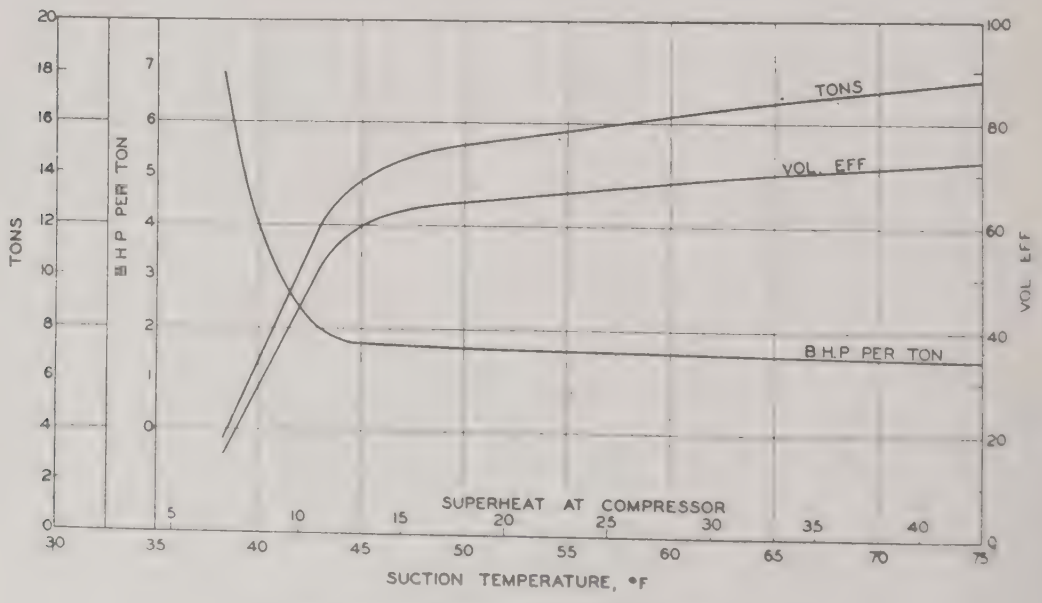


Fig. 13. Variable Superheat at Constant Pressure

(Refrigerant—Freon; suction pressure 31 psig; discharge pressure 140 psig; barometer 29 in at 80 F)

The practice as to adequate condenser surface, when the primary and secondary surface is considered, is to use 150 sq ft of total surface per ton of refrigeration with the average air velocity of 500 fpm. This applies to full-length condensers, covering the whole condensing unit assembly. When a condenser is placed directly over the fan, proportionately less surface and higher air velocities are required, from 800 to 1,000 fpm. Higher air velocity increases the noise factor. The edge of the fan should be placed $\frac{3}{4}$ to $1\frac{1}{4}$ in. away from the condenser. The choice of fan is a problem of experimentation.

Table 5. Capacity as Affected by Oil

Percentage of oil in Freon-12 by weight	Percentage reduction of capacity in evaporator due to oil mixture
0	0
1	$\frac{1}{2}$
2	$1\frac{1}{4}$
3	$2\frac{1}{4}$
4	3
5	4
6	$4\frac{3}{4}$
7	$5\frac{1}{2}$
8	$6\frac{1}{4}$
9	7
10	8
15	13
20	$18\frac{1}{2}$
30	$32\frac{1}{2}$
40	48
50	63

Table 6. Increase in Boiling Point Due to Oil in Freon-12, deg F

Boiling point of pure Freon-12, F	Part of oil in circulation by weight, %				
	10	20	30	40	50
0	1.50	3.50	5.50	8.0	11.50
5	1.36	3.3	5.3	7.8	11.2
10	1.2	3.12	5.15	7.6	10.87
15	1.05	3.0	5.0	7.36	10.53
20	.9	2.8	4.7	7.15	10.25
25	.75	2.6	4.65	6.9	9.9
30	.6	2.46	4.5	6.7	9.6
40	.3	2.12	4.15	6.25	9.0

Condensers, when applied to low-pressure refrigerants, are used from −40 F to +50 F. Manufacturers have adopted and catalogued commercial condensing units as low-temperature applications, ranging from −40 F to 0 F; medium-temperature, −10 F to 25 F; and high-temperature, 30 F to 50 F.

28. Water-cooled condensers are generally classified into three types as follows:

a. **Shell-and-tube** type is similar to the familiar ammonia construction, with tubes rolled into tube sheets. Steel shells and tube sheets with copper tubes, usually finned, are common. For marine use, shells are copper with cupronickel tubes and tube sheets. The gases to be condensed are on the outside of the tube. The water travels in a multipass flow inside of the tubes. Heads can be removed to clean tubes when necessary; $\frac{5}{8}$-in. O.D. copper tubes with .042-in. wall, finned, make a compact tube design.

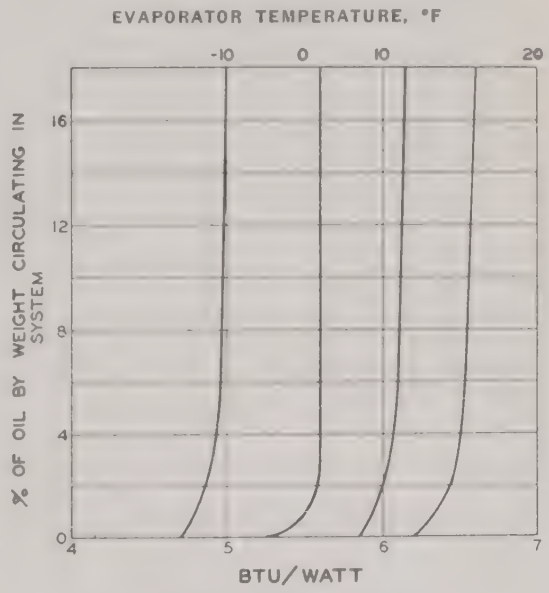


Fig. 14. Effect of Oil Circulation on Compressor Efficiency

b. The second type, the **double-pipe**, usually made of copper, is a tube within a tube, using the counterflow principle. The refrigerant passes from the head of the compressor between the outer and inner

tubes, while water passes from the bottom up, within the inner tube. Tube sizes may be $\frac{3}{4}$-in. O.D., .042-in. wall outside, and $\frac{1}{2}$-in. O.D., .035-in. wall inside; or 1-in. O.D., .05-in. wall outside, and $\frac{3}{4}$-in. O.D., .042-in. wall inside.

c. The third type, known as the **shell-and-coil** type, uses an outer shell and a water tube bundle usually finned, placed inside with inlet and outlet water nipple projections. This type of condenser is built with a primary surface of copper tubing, and has a secondary surface made of fins, usually spaced from four to seven fins per inch. The water tube bundle is made either as a helical coil or a nest of straight tubes suitably manifolded, inserted inside the shell and rigidly fastened in place. The shell is then closed and welded. Common tube sizes are $\frac{1}{2}$-in. O.D., $\frac{5}{8}$-in. O.D., or $\frac{3}{4}$-in. O.D. copper tubes. This arrangement does not permit cleaning of tubes except by chemical action.

The surface area required per ton of refrigeration for the shell-and-tube type is 12 sq ft, for the double-pipe type 9 sq ft, and for the shell-and-coil type 24 sq ft. These values are based on the capacity of the condensing unit at 50 F evaporator temperature, or at the peak temperature for the particular operating range.

There is a tendency to maintain low pressures on condensing units without considering the balance between cost of power and cost of water. **Fig. 15** shows the quantity of water and power required for a $\frac{1}{2}$-hp Freon condensing unit. It is to be seen that with water at 25¢ per 1,000 gal or $1.87 per 1,000 cu ft and power at five cents per kilowatt, it is more economical to operate with head pressures from 130 to 150 psi.

Table 7. Heat Removed from Freon-12 by Condenser, Per Ton of Refrigeration, Btu
(65 F suction vapor entering compressor)

Condensing temp, F		90	100	110	120
Condensing pressure, psig		99.6	116.9	136.0	157.1
Suction temp, F	Suction pressure, psig				
0	9.17	16,320	16,930	17,600	18,360
10	14.15	15,620	16,220	16,850	17,550
20	21.05	14,950	15,550	16,100	16,750
30	28.46	14,350	14,900	15,400	16,000
40	36.98	13,830	14,320	14,750	15,250
50	46.69	13,400	13,800	14,150	14,600

Evaporative Condensers

29. An evaporative condenser for the small condensing unit may comprise only a coil with a propeller fan. Into the fan is directed a stream of water and no water is recirculated. Larger sizes provide fans drawing air up through the wetted condenser coil, while water is sprayed over the coil by a recirculating pump. The following formulas permit ready check of the evaporative condenser requirement for the small condensing units. The amount of air required is

$$\frac{\text{Condensing unit capacity, Btu/hr}}{3.5(h_2 - h_1)} \qquad (9)$$

Where,

h_2 = total heat in the leaving air, Btu per lb at a wet-bulb temperature of $T_c - 15$

T_c = refrigerant condensing temperature, deg F

h_1 = total heat in entering air, Btu per hr at atmospheric wet-bulb temperature of T_A

T_A = wet-bulb temperature, atmospheric air, deg F

The air required ranges between 250 and 300 cfm per ton. Surface requirements

for finned tubing can be approximated by the following formula:

$$B_f = \frac{\text{Condensing unit capacity, Btu/hr}}{6(T_C - T_A + 15)} \quad (10)$$

Where B_f = surface area, sq ft, including prime and fin surface. (The ratio of fin to prime must not exceed 10 to 1.)

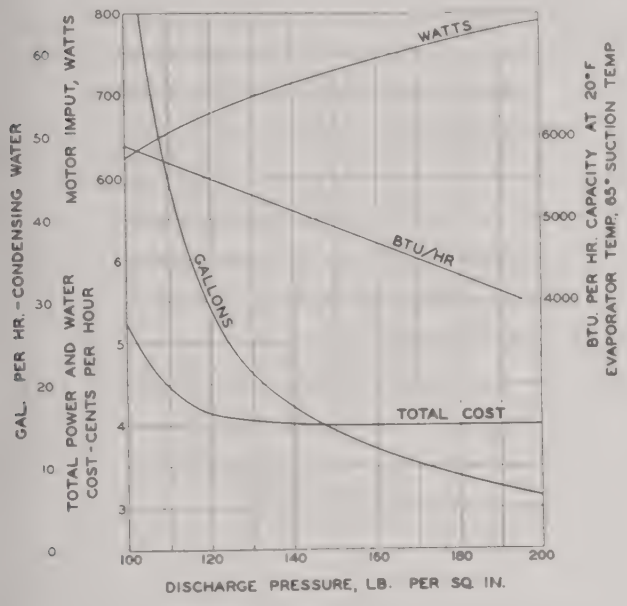


Fig. 15. Water-Cooled Freon-12, ½-hp Compressor; 2-cylinder, 1-13/16 in. bore × 1⅜ in. stroke, 580 rpm

Surface requirements where all the surface is prime will be approximated as follows:

$$B_p = \frac{\text{Condensing unit capacity, Btu/hr}}{19(T_C - T_A + 15)} \quad (11)$$

Where B_p = area of prime surface, sq ft.

The size of the casing for induced air flow will be approximated by the following:

$$\frac{\text{Air required, cfm}}{400} \quad (12)$$

The amount of water circulated for the sprays will closely approximate 3.5 × area of casing. The amount of make-up water necessary will be:

$$\frac{\text{Condensing unit capacity, Btu/hr}}{300{,}000} \quad (13)$$

Gasoline Engine Drive

30. Whenever electric power is unavailable, such as on the farm, or for bus or truck transportation, an air-cooled gasoline engine is mounted on the condensing unit in place of the motor. The engine ratings should be at least 2.5 to 3.0 times the hp of the motor it replaces, to take care of overloads.

The engine should be a constant-speed type, controlled by a fly-ball governor connected to a butterfly valve on the carburetor. It is advisable to provide a clutch pulley on the engine drive, so that the engine can be cranked without turning the compressor.

Condensing units are usually furnished without controls and are started by hand. More elaborate installations for train and bus air conditioning applications provide automatic starting and stopping. Engines can be started by hand and stopped with a reverse-acting pressure or temperature control, with contacts to short circuit the magneto and stop the engine. On air-cooled condensing units, the gasoline engine should be arranged so that its flywheel fan blows air away from the condenser, for otherwise hot air from the engine will raise the head pressure. Some manufacturers prefer to use a jack shaft at the point where an electric motor would ordinarily be mounted, and arrange the gasoline engine on a separate base adjacent to the condensing unit.

The oil in air-cooled gasoline engines of small size should be changed every 50 hr. These engines consume lubricating oil at the rate of approximately 1 gal every 300 hr. Engine speeds vary from 1,800 to 3,600 rpm. Fuel consumption is about 1.2 lb per bhphr, or 1.66 pints per bhphr.

An interesting addition to the engine-driven condensing unit is the propane cycle refrigeration unit, developed by Schlumbohm.[7] In this system, gas from the condenser operates the engine. The quantity removed is automatically recharged into the system from a charging drum.

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32. COMMERCIAL LOW SIDES

AN EVAPORATOR is that part of a mechanical refrigeration system in which the liquid refrigerant is evaporated and the heat required for evaporation is absorbed from the medium to be cooled. The medium cooled may be a gas such as air, a liquid such as water, or a solid such as packaged food.

Evaporator Materials and Refrigerants

1. Evaporators are usually made of iron, steel, brass, copper, or aluminum, depending upon the use of the evaporator and the refrigerant employed.

The most commonly used **refrigerants** in commercial installations are Freon-12, methyl chloride, sulfur dioxide, and ammonia. Freon-12 and sulfur dioxide can be employed with iron, steel, brass, copper, or aluminum evaporators. An aluminum-magnesium alloy is not recommended with Freon-12. Ammonia can be used with iron, steel, or aluminum evaporators. It is dangerous to use copper alloy with ammonia, since any moisture present will form ammonium hydroxide, which attacks and corrodes copper. Methyl chloride can be used with iron, steel, brass, or copper, but is not used with aluminum due to the reaction between methyl chloride, lubricating oil, and aluminum, which finally reduces the lubricating oil to a heavy sludge.

Refrigerant Control

2. Methods of refrigerant flow control are as follows:

a. Flooded
   1. High-pressure float
   2. Low-pressure float
   3. Injector nozzle
b. Automatic expansion valve
c. Thermostatic expansion valve.

These methods are discussed in Chaps. 20, 22 and 25.

The combination of thermostatic expansion valve and dry-expansion type evaporator usually is lower in cost, easier to construct to the required shape, less complicated to install, more dependable in returning oil to the compressor, and requires less space than a float-controlled, flooded-type evaporator with its auxiliaries. For these reasons, the **thermostatic expansion valve** is the most commonly used control for commercial applications. In applications where the level of the evaporator is constantly shifting, as on boats, trucks, and trains, the flooded types will usually not give as satisfactory operation as the dry-expansion type. The quantity of refrigerant required is much greater with the flooded type than with the dry-expansion type.

However, better heat transfer can be obtained with liquid flowing through the entire evaporator as in a flooded type. This means that a smaller and lighter evaporator may be used than with expansion valve control.

Defrosting

3. In this chapter, defrosting will be referred to as one of two types: (a) natural and (2) artificial. **Natural defrosting** is that type which occurs during the idle periods of the system and which requires no automatic or manual attention. Its use is limited to evaporators in rooms with temperatures above freezing. **Artificial defrosting** is the type where the defrosting operation is performed either manually or by automatic controls. Artificial defrosting includes scraping off the frost manually, running water over the evaporator, using hot gas or applying some external heat.

A typical hot gas defrosting hookup is shown in **Fig. 1.** During normal operation the valves A, B, E, and F are closed and C and D are open. In order to defrost the left evaporator valve C is closed and the valves A and E are opened. The hot compressed gas then leaves the compressor and flows through the discharge line to this evaporator through the valve A. The heat

which is absorbed melts the frost, and the gas is condensed. The liquid then flows from the evaporator through the valve E and the liquid line to the expansion valve of the right evaporator. When the ice has been completely removed from the first evaporator, it is returned to normal operation by closing valves A and E and opening C. The second evaporator is defrosted in the same manner.

Types and Purposes

4. Evaporator construction may be generally classified as to three types:

a. Bare pipe or tubing is most generally applied where the air temperature to be maintained is below 34 F, and for liquid cooling where the coils are submerged.

b. Plate surface evaporators usually consist of two pieces of flat sheet metal soldered or welded together at the outside edges, with either welded internal construction or formed tubing to provide refrigerant passage. Those with welded internal construction have the refrigerant in **direct contact** with sheet metal surfaces. The other type, in which the refrigerant passage is through formed tubing, usually has a **vacuum** or a **eutectic solution** in the space between the tubes. The vacuum permits atmospheric pressure to force the outside flat sheets against the tubing, thereby providing a better, more positive heat transfer bond between them.

The direct refrigerant contact type and the vacuum type are used for shelf surfaces in freezer rooms and display cases, in place of bare pipe in low-temperature (below 34 F) storage rooms, in farm and home freezers, and in liquid chilling tanks. The type containing a eutectic solution is generally used in conjunction with a central refrigerating plant, such as with refrigerated trucks, where the plates in the truck are connected to the refrigerating system during the night, thereby freezing the eutectic solution to supply sufficient holdover refrigeration for the truck the following day.

Sheet-metal plates with refrigerant tubing soldered to one side are used as salad pans and in various commercial cabinets and fixtures and in some freezer cabinets.

The corrugated plate surface, commonly termed an **aerator,** is constructed of two pieces of corrugated sheet metal welded together at the outside edges and welded together between each corrugation, the corrugations then providing a passage for the refrigerating medium, either a cooled liquid, such as brine or water or some refrigerant. Aerators are used for liquid cooling only, chiefly in dairies for milk cooling and bottling plants for water cooling. The aerator is mounted in a vertical position with the corrugations horizontal, the liquid to be cooled being fed by gravity from a distributor above the aerator. The liquid flows down over the aerator surface in a thin sheet. These evaporators are usually made of stainless steel for sanitary reasons.

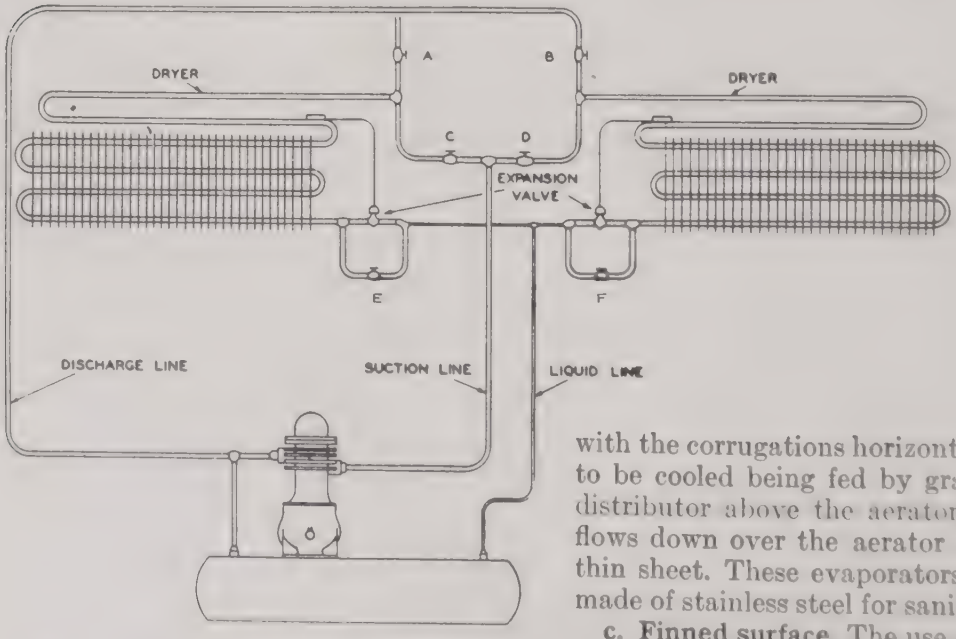


Fig. 1. Typical Hot Gas Defrosting Hookup

c. Finned surface. The use of dry finned surface evaporators is generally limited to

air-cooling applications where the temperature maintained is above 34 F, especially with respect to coils operating on gravity air flow circulation, since most commercial installations work on a natural defrosting cycle. There are several types of forced air coolers using finned coils on the market designed to operate in low-temperature spaces. Most of these units have incorporated in their design some method of artificial defrosting. Finned coils as well as bare pipe coils are used in brine spray units.

Bare Pipe and Flat Plate Surfaces for Air Cooling

5. a. Bare pipe and flat plate evaporators, while giving satisfactory operation at all temperatures, are generally limited to applications requiring 34 F and below. **Bare pipe evaporators** are usually constructed of scale-free pipe on the larger installations and copper tubing on the smaller jobs. While bare pipe evaporators can be made in practically any shape or size, they are usually found in one of three common designs (see **Fig. 2**). The most common type is the flat evaporator which is either hung on the walls or ceiling or sometimes used as a shelf. Where the space is limited, zig-zag pipe coils are often employed. For fixtures having overhead evaporator bunkers, the oval pipe coil is used.

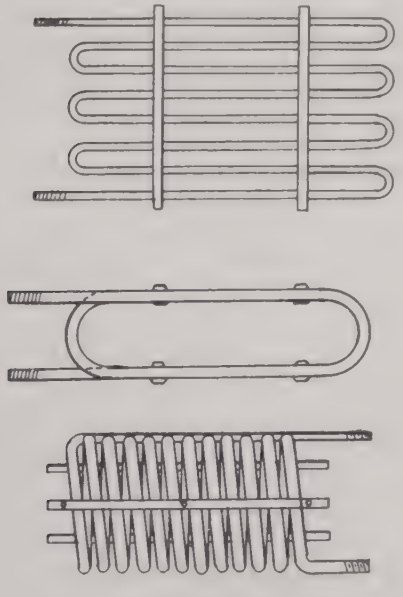


Fig. 2. Bare Pipe Coil Types

The **flat plate type evaporator** is often used in place of the wall or ceiling type pipe coil. For shelves where products are stored or frozen, the flat plate type is superior to the pipe coil due to the increased surface of the evaporator in contact with the product to be frozen.

A considerable thickness of frost can be accumulated on pipe and plate surface before the capacity is materially reduced. However, periodic artificial defrosting will be necessary, the length of a period depending upon the moisture available from the product, varying between open, moist products and canned products. Defrosting is accomplished either by hand scraping or by the application of heat.

Table 1. Pipe Coil Capacities, Gravity Air Circulation

(Btu/hr/lin ft of 1-in. standard or extra heavy pipe)

Refrig temp, F	Fixture temperature, F				
	−20	0	20	30	40
25					11.1
20					16.2
15				9.3	21.0
10				12.7	25.8
5			7.5	16.3	30.4
0			11.0	19.9	34.9
− 5			14.6	23.5	
−10			18.4	27.5	
−15		7.5	21.6		
−20		11.0	24.8		
−25		14.6			
−30		18.3			
−35	7.3	21.5			

b. Heat transfer values for pipe coils with gravity air circulation, as recommended by most manufacturers, vary from 1.6 to 2.3 Btu per sq ft hr deg F. **Table 1** indicates pipe capacities for 1-in. pipe. These capacities are based on heat transfer values closely approximating the above figures. The heat transfer of gravity air circulation pipe coils improves with greater temperature difference, due to the increased air circulation. Where the coils are used for shelves in contact with the product to be cooled, heat transfer will not only take place by convection but also by

conduction. This will result in a higher rate of heat transfer, which is generally considered to be 15 to 25% greater than that for gravity air circulation coils.

Most of the plate surface manufacturers give ratings from 2.0 to 2.5 Btu per hr per sq ft per deg F, considering both sides of

Table 2. Recommended Drier Pipe Lengths for Bare Pipe Coils

Fixture temperature, F	Length of pipe coil, ft				
	50	100	150	200	250
	Length added for drier, ft				
−20	20	33	46	56	64
−10	18	30	42	50	57
0	16	27	38	45	51
10	14	24	33	40	45
20	13	21	29	35	39
30	11	18	24	30	33
40	9	15	19	24	27
50	7	12	15	18	20
60	5	9	11	13	15

the plate effective when used in "still air" at 40 F. These are based on frost-free plates. Values of 2.0 and less are given when used in 0 F air, assuming some frost accumulation. When the plates are used as shelf surfaces with the product to be chilled or frozen resting thereon, the heat transfer rate (to the product) is increased 15 to 25%.

c. Drier surface is recommended with all gravity air circulation pipe coils and plates controlled with thermostatic expansion valves. The drier surface consists of an additional length of pipe or plate surface. **Table 2** shows the added lengths sometimes used. When two spaces are cooled with one condensing unit and the bare pipe evaporators are connected in series the drier is added to the end of the last coil. In this case the drier coil obtained in Table 2 is usually increased by the amount shown on **Table 3.** Some manufacturers recommend that either the bare pipe evaporator be increased 25% to provide drier surface or a liquid-gas heat exchanger be used.

d. The **thermal bulb** of the thermostatic expansion valve should be located on the condensing unit side of either the drier coil or the heat exchanger. With the low heat

Table 3. Drier Pipe Coil Factors for Use with Evaporators in Series

Per cent of total load in last evaporator	Multiplier
20	2.1
30	1.9
40	1.7
50	1.5
60	1.3
70	1.1
80	1.0

transfer rate obtained from gravity air circulation pipe coils, satisfactory operation will be obtained with the thermal bulb on the condensing unit side of the heat exchanger.

Table 4 shows the recommended length of pipe or tubing per feed with various suction pressures and loadings. On installations requiring more than one feed controlled by thermostatic expansion valves, a thermostatic valve is usually put on each feed. Two or more lines can be fed with one valve, but quite often this leads to refrigerant distribution problems.

Bare Pipe and Flat Plate Surfaces for Liquid Cooling

6. a. Submerged pipe coils are made chiefly in the flat or the oval shapes shown in Fig. 2, the cylindrical shape shown to be used in small chambers.

b. The following **heat transfer** values for a non-agitated bath are obtainable when coils of the flat or oval shape are mounted vertically in the liquid bath for an approximate minimum height of two feet. Mounting the coils this way provides a satisfactory natural circulation of the bath. If the coils were mounted in a horizontal position, there would be little natural bath circulation and the rate of heat transfer would be reduced. This factor should be considered on those applications where it applies and a safety factor used. In a non-agitated water bath with thermostatic expansion control, the heat transfer for bare pipe or copper tubing is 30 Btu per sq ft hr deg F. When using this value, no drier surface is required. With a flooded evaporator and the other conditions the same as above, the heat transfer rate is 40. **Figs. 3a, 3b, 3c,** and **3d** indicate some rec-

Table 4. Maximum Load, Btu per hr in Freon-12 Evaporators for Each Thermostatic Expansion Valve*

Equivalent length of pipe or tube per valve, ft	O.D. size of tube, in.							Standard size of pipe or tube, in.						
	⅜	½	⅝	¾	⅞	1	1⅛	⅜	½	¾	1	1¼	1½	2
40	1,130													
50	1,010													
60	920	2,400						3,400						
70	850	2,200	4,300					3,120						
80	795	2,050	4,000					2,920	5,100					
90	755	1,950	3,800					2,750	4,850					
100	720	1,850	3,550	5,800	8,400			2,600	4,650	9,900				
110	685	1,770	3,410	5,550	8,000			2,500	4,400	9,500				
120	655	1,690	3,270	5,300	7,700			2,400	4,200	9,050				
130	625	1,620	3,120	5,100	7,400			2,300	4,050	8,700				
140	605	1,550	3,010	4,900	7,120			2,200	3,900	8,400				
150	585	1,500	2,900	4,750	6,850	11,800	13,100	2,150	3,780	8,100	14.600	30,300		
200	510	1,295	2,500	4,100	5,950	10,250	11,400	1,850	3,290	7,000	12,600	26,300	38,300	
250	455	1,150	2,225	3,650	5,300	9,200	10,100	1,650	2,900	6,250	11,300	23,500	34,200	63,000
300	415	1,050	2,030	3,320	4,900	8,350	9,200	1,500	2,670	5,700	10,300	21,400	31,400	57,500
350	385	965	1,900	3,075	4,500	7,700	8,500	1,390	2,480	5,250	9,500	19,900	29,000	53,000
400	360	910	1,770	2,860	4,250	7,250	7,950	1,290	2,300	4,930	8,900	18,500	27,100	49,800
450	340	850	1,670	2,700	4,000	6,850	7,500	1,220	2,170	4,620	8,350	17,500	25,600	47,000
500	320	810	1,500	2,500	3,700	6,400	7,100	1,150	2,050	4,400	7,900	16,500	24,400	44,500
550	310	770	1,490	2,430	3,550	6,100	6,800	1,100	1,970	4,175	7,500	15,700	23,200	42,500
600	295	740	1,430	2,320	3,400	5,900	6,500	1,045	1,900	4,000	7,200	15,100	22,100	40,500
650	285	705	1,370	2,220	3,300	5,700	6,250	1,010	1,830	3,850	6,980	14,500	21,300	39,000
700	272	680	1,320	2,150	3,200	5,450	6,000	970	1,750	3,700	6,700	14,000	20,500	37,500
750	263	650	1,270	2,080	3,100	5,300	5,800	940	1,700	3,550	6,550	13,500	19,800	36,500
800	225	630	1,230	2,015	2,990	5,100	5,620	910	1,650	3,460	6,250	13,100	19,200	35,200
850		610	1,200	1,950	2,900	4,990	5,450	880	1,600	3,350	6,100	12,600	18,600	34,200
900		596	1,160	1,905	2,800	4,850	5,300	850	1,550	3,280	5,900	12,300	18,100	33,200
950		580	1,135	1,850	2,750	4,725	5,150	830	1,510	3,180	5,750	12,000	17,650	32,500
1,000		570	1,120	1,820	2,700	4,600	5,050	810	1,475	3,100	5,600	11,700	17,200	31,350
1,100			1,070	1,730	2,550	4,400	4,850		1,400	2,950	5,300	11,200	16,400	30,000
1,200			1,015	1,650	2,450	4,200	4,620		1,340	2,800	5,075	10,700	15,700	28,800
1,300			970	1,580	2,350	4,030	4,450		1,290	2,700	4,900	10,300	15,100	27,700
1,400			940	1,525	2,260	3,900	4,300		1,240	2,600	4,700	9,900	14,600	26,500
1,500			905	1,480	2,190	3,750	4,150		1,200	2,500	4,550	9,600	14,200	25,800
1,600				1,430	2,110	3,625	4,000			2,430	4,400	9,250	13,700	25,000
1,700				1,380	2,060	3,525	3,900			2,370	4,290	9,000	13,300	24,200
1,800					2,000	3,450	3,780			2,290	4,150	8,800	12,900	23,600
1,900					1,960	3,350	3,680			2,230	4,050	8,500	12,600	22,900
2,000					1,900	3,260	3,580			2,170	3,950	8,250	12,200	22,300
2,100						3,180	3,490				3,850	8,100	11,900	21,700
2,200						3,100	3,400				3,725	7,900	11,650	21,200
2,300						3,020	3,330				3,650	7,750	11,400	20,800
2,400						2,960	3,250				3,580	7,600	11,200	20,400
2,500						2,910	3,200					7,450	11,000	20,000
2,600												7,300	10,800	19,700
2,700												7,150	10,600	19,300
2,800												7,050	10,400	19,000
2,900												5,900	10,200	18,700
3,000												6,800	10,000	18,300
3,100													9,820	17,900
3,200													9,700	17,600
3,300													9,550	17,400
3,400													9,400	17,200
3,500													9,250	16,900
3,600														16,700
3,700														16,500
3,800														16,300
3,900														16,100
4,000														15,800

* Reference should be made to Tables 5 and 6, and the example shown on page 544.

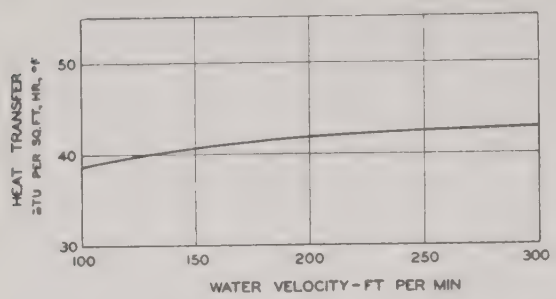


Fig. 3a. Heat Transfer of Bare Pipe or Copper Tubing Submerged in Water Bath, No Agitation of Bath, Water in Tubing

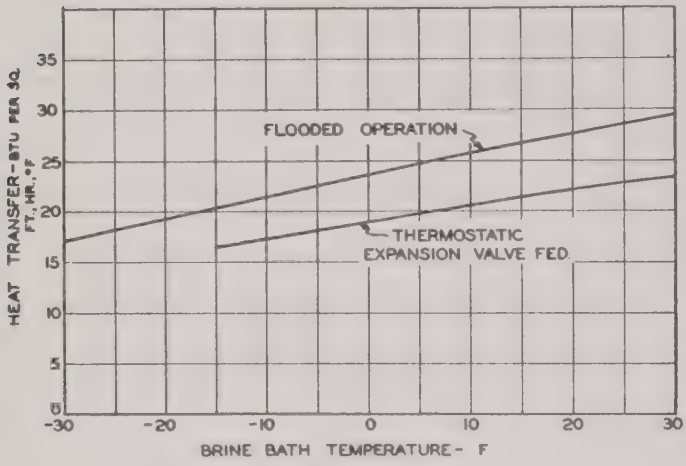


Fig. 3b. Heat Transfer of Bare Pipe or Copper Tubing Submerged in Brine Bath, No Agitation of Bath, Refrigerant in Tubing

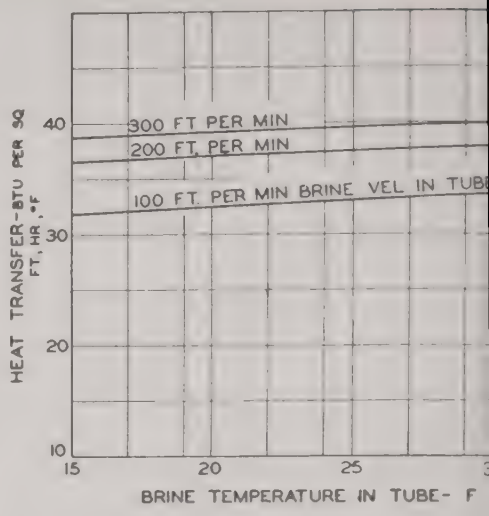


Fig. 3c. Heat Transfer of Bare Pipe or Co Tubing Submerged in Water Bath, No Agit of Bath, Brine in Tubing

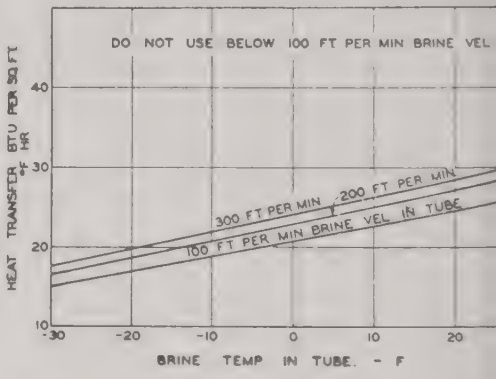


Fig. 3d. Heat Transfer of Bare Pipe or Co Tubing Submerged in Brine Bath, No Agitatio Bath, Brine in Tubing

ommended heat transfer rates for bare pipe or copper tubing in water, or brine baths with refrigerant, water, or brine in the pipe. For length of pipe or tubing per refrigerant feed see Table 4.

Finned Surface with Gravity Air Circulation

7. a. Finned coils with gravity air circulation are employed in walk-in coolers, reach-in or grocery refrigerators, display cases, and counter cabinets. Although unit coolers have been developed and are partially replacing the gravity-type coils, for applications requiring minimum dehydration and loss of color, the gravity coil is usually recommended. For this reason gravity coils are ordinarily used in wholesale storage rooms and aging rooms. Gravity coils are also used where space or electric power limitations eliminate the use of forced air units.

Satisfactory **coil** and **baffle arrangements** are shown in **Figs. 4, 5,** and **6.** With the arrangements of Figs. 4 or 5, the coils are one or two rows deep, that is, approximately 3 to 7 in. deep. Shallow coils spread over the ceiling are preferable for best air circulation over the products and for greatest unit surface efficiency. As the coil depth is increased, the air temperature range through the coil is increased, thereby reducing the mean temperatures, which finally results in less overall coil capacity.

The side wall coil of Fig. 6 delivers approximately the same heat transfer as the overhead coils due to the chimney effect of the baffles. These coils without the baffles

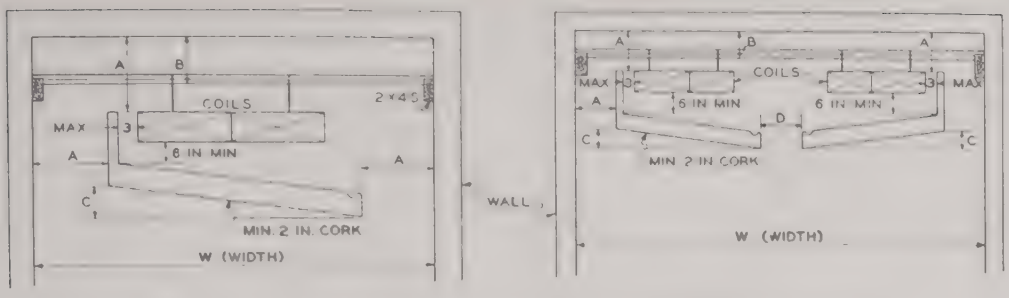


Fig. 4. Coil Arrangement for Single L Bunker (left); Coil Arrangement for Double L Bunker (right)

will have about 50 to 60% of the capacity they will have when baffles are used.

The effect of distance from ceiling for a one-row coil with five fins per inch without baffles has been studied. With a 40 deg temperature difference the coil capacity increased 33% in moving it from 1 in. to 3 in. below the ceiling. It increased an additional 17% when moved to 7 in. below the ceiling.

With arrangements of Figs. 4 and 5 the warm air flue should be located on the door side of the cooler in order that the warm air entering the door will pass through and drop its excess moisture to the coil before flowing over the products. Otherwise this moisture would condense on the cold products.

Several coil and baffle arrangements for grocery or reach-in refrigerators and display cases are shown in Figs. 7 and 8. In general, the coils and baffles are supplied and installed by the cabinet manufacturers.

In selecting finned-type evaporators the 24-hr daily load is first determined. A condensing unit is then selected on the basis that it will absorb this load in 12 to 16 hr

Table 5. Multiplier Table

Temp diff	Evaporator Temperature −20	−15	−10	−5	0	5	10	15	20	25	30
10	.69	.77	.86	.96	1.06	1.18	1.31	1.45	1.59	1.74	1.90
15	.86	.97	1.08	1.20	1.32	1.46	1.61	1.78	1.95	2.13	2.32
20	1.00	1.12	1.23	1.37	1.51	1.68	1.87	2.06	2.27	2.49	2.71
25	1.14	1.28	1.42	1.57	1.73	1.90	2.10	2.31	2.55	2.81	3.08

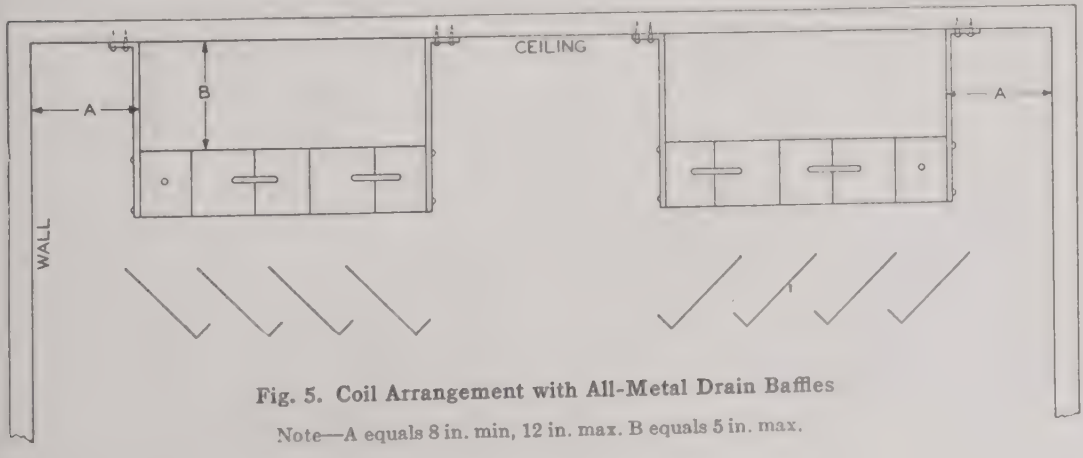


Fig. 5. Coil Arrangement with All-Metal Drain Baffles

Note—A equals 8 in. min, 12 in. max. B equals 5 in. max.

of operation, in order to provide sufficient off periods for natural defrosting and excess capacity for peak loads. An evaporator is chosen which has a capacity equal to

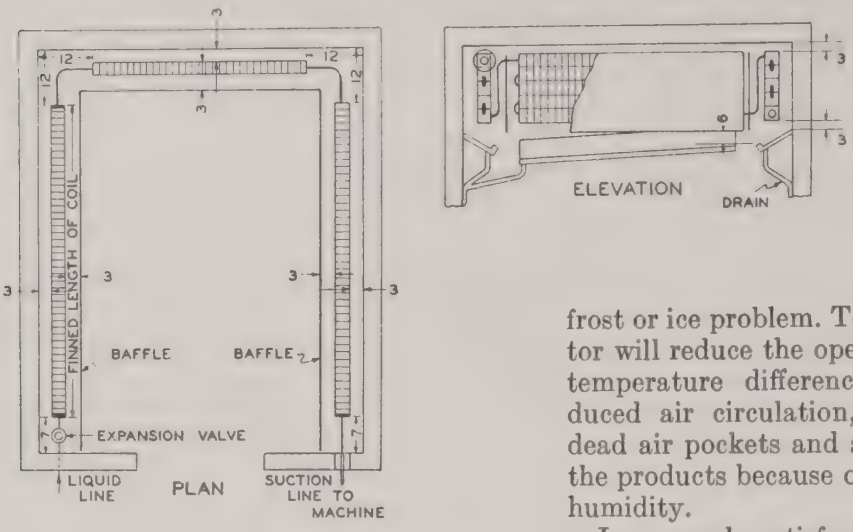


Fig. 6. Coils for Side Wall Installation

(Low ceiling storage rooms)

or slightly more than the condensing unit capacity. An evaporator too small will result in low relative humidity, product dehydration, greater temperature differences between refrigerant and air, difficulty in maintaining desired fixture temperature, longer running periods, and possibly a frost or ice problem. Too large an evaporator will reduce the operating time and the temperature difference, resulting in reduced air circulation, which may cause dead air pockets and a slimy condition of the products because of too high a relative humidity.

In general, satisfactory storage is obtained in retail store fixtures with a temperature difference between 15 and 25 F. For wholesale storage, usually for a longer period than retail service, the temperature difference can be broken down into three groups according to the minimum dehydration desired:

1. 15 to 20 F for cheese, cut meats, meat in quarters or halves, eggs, fresh fruit, fresh vegetables, poultry, and the like;
2. 20 to 25 F for meats, dried fruits, and vegetables, iced fish and beer in wood kegs;
3. Any temperature difference is satisfactory with products that cannot give up their moisture, such as canned or bottled goods and beer in steel kegs.

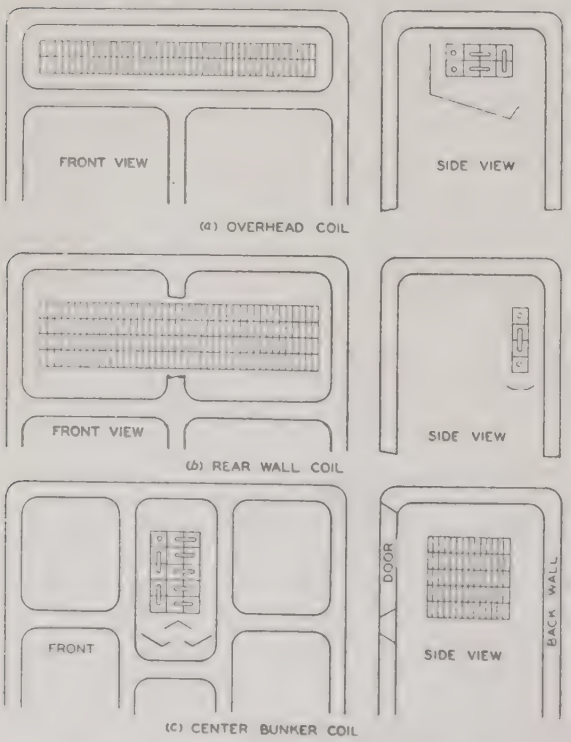


Fig. 7. Grocery or Reach-in Refrigerators

b. The rate of **heat transfer** for gravity air flow over direct expansion finned coils varies chiefly with the air velocity and air distribution over the coils. The velocity and distribution will vary considerably with each type of fixture depending upon the bunker, baffle and coil arrangement. It will also vary with the size of warm and cold air flues and the head of air pressure available for generating circulation as well as with the fin spacing, rows of tubes wide and deep, fin size and fin design of the coil.

Hence it is difficult to offer specific heat transfer values that can be validly used in selecting the size evaporator required for a given load. Values can be taken from manufacturers' data obtained in accordance with the ASRE Standard 25-44.

Tests of transfer from air to finned surface have been made by Askin. Variables include the ice on the surface, variation in suction temperature, amount of superheat and the like. A coil was placed in a calorimeter with constant suction pressure maintained by means of an automatic expansion valve, refrigerant vapor being superheated a fixed amount. **Fig. 9** shows the results, from which it was concluded that the greatest value is obtained in a coil that is as shallow as possible, with a wide fin pitch. In other words, it is better to use a 2-row, $\frac{1}{2}$-in. pitch than to use a 4-row, $\frac{1}{4}$-in. pitch, with the same surface. Another factor is length; it is more economical to select a length of 80 in. and a width of 18 in. than a length of say 40 in. with a width of 36 in. It must be remembered that the heat transfer values of Fig. 9 are applicable only for the fixture tested and are included here only for the purpose of showing the effect of tubes high and fin spacing.

Generally speaking, the heat transfer of a specific type of surface will vary progressively from high to low in the following applications: (a) overhead and side wall coils in walk-in coolers; (b) overhead coils in a reach-in refrigerator; (c) side wall coils in reach-in refrigerators and display cases; (e) center bunker coils in a reach-in refrigerator and back bunker coils in display cases.

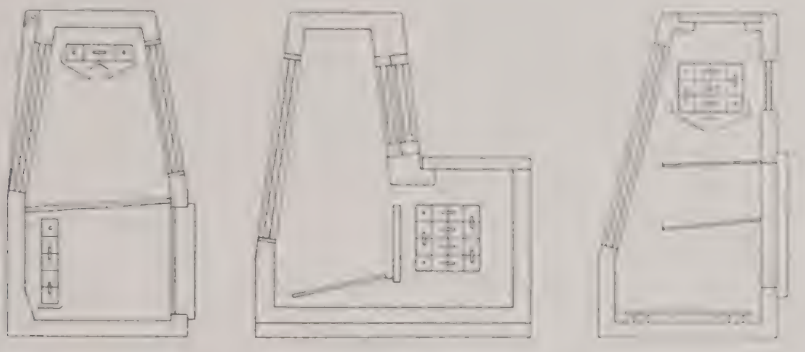


Fig. 8. Coil Arrangement for Display Cases

c. Drier coils are frequently recommended with gravity air flow finned surface, the drier coil consisting of a length of bare tubing located above the evaporator in the warm air. One manufacturer recommends 15 to 20 ft of drier coil, with the

Table 6. Return Bend Table

Size of tubing or pipe, in.	Equiv ft of tubing or pipe per return bend
$\frac{3}{8}$	1.25
$\frac{1}{2}$	1.50
$\frac{5}{8}$	1.50
$\frac{3}{4}$	1.75
$\frac{7}{8}$	1.75
1	2.50
$1\frac{1}{8}$	2.50
$1\frac{1}{4}$	3.00
$1\frac{1}{2}$	4.00
2	7.00

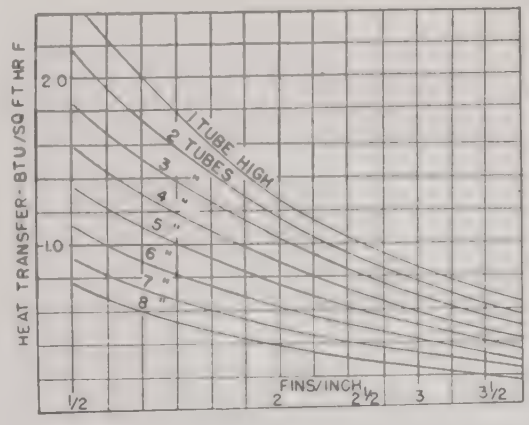


Fig. 9. Performance of Finned Coils under Natural Circulation (fins 24 in. wide × $2\frac{3}{4}$ in./tube high)

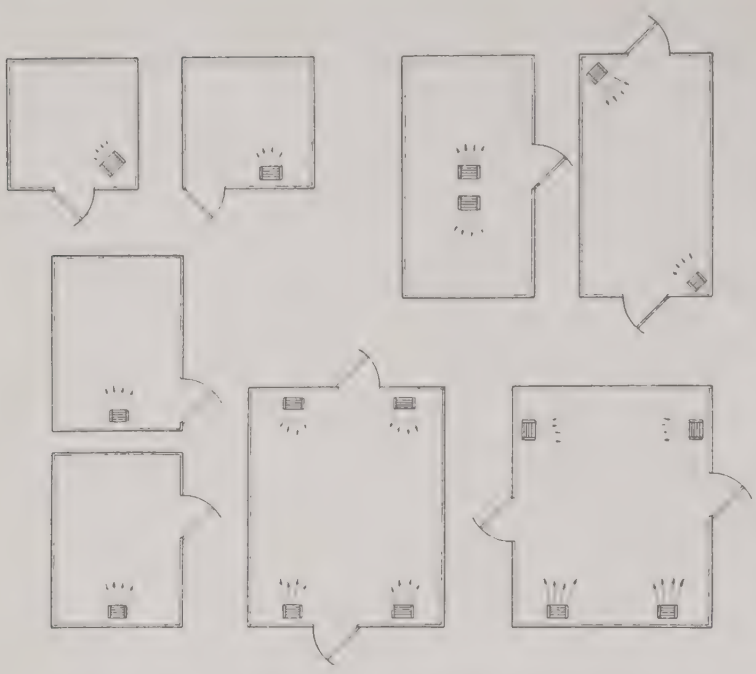


Fig. 10. Suggestions for Location of Unit Coolers in Walk-in Refrigerators

thermal bulb of the expansion valve located on the drier coil midway between the evaporator outlet and the point where the suction line leaves the fixture. The drier coil is recommended by this manufacturer for the purposes of providing superheating surface and for assurance that outside heat will not be conducted back along the suction line to the thermal bulb during the off cycle, otherwise the conducted heat would open the expansion valve and flood the evaporator.

On installations where the suction line must pass through spaces in which sweat dripping from the suction line would be objectionable, a liquid-gas heat exchanger located inside the fixture will reduce and quite often eliminate this problem. The use of the heat exchanger will also increase the capacity of the condensing unit due to the liquid subcooling obtained. Such a heat exchanger should also be used if the evaporator is located much above the condenser.

d. On most finned surface commercial applications the refrigerant flow is controlled by **thermostatic expansion valves.** From Table 4 the maximum recommended length of tubing for one refrigerant line can be determined in order to operate with a reasonable refrigerant pressure drop through the coil. In the event that more than one feed is required, it is considered preferable to use a thermostatic expansion valve with each feed. Two or more feeds can be controlled by one expansion valve, if the proper refrigerant distribution can be attained. There are multi-outlet expansion valves on the market that perform satisfactorily for use with properly designed multiple-feed evaporators.

With finned surface on any evaporator operating with a comparatively high rate of heat transfer, the thermal bulb should be located on the evaporator side of liquid-gas heat exchangers for satisfactory operation, otherwise excessive pressure cycling will result.

Unit Coolers

8. a. During the past few years, unit coolers have gained wide acceptance for use in retail commercial refrigeration fixtures. From the many models and sizes on the market, a unit cooler can be selected for practically any application where gravity air flow finned coils are applicable, such as walk-in coolers, reach-in refrigerators, display cases and beverage cooling cabinets.

In general, these coolers are applied to maintain temperatures above 34 F, in order to operate on a natural defrosting cycle. There are, however, models that are designed for low-temperature applications, having included in their design a method for artificial defrosting. Three common methods of artificial defrosting are: (a) the use of hot discharge gas with the arrangement of Fig. 1; (b) city water distributed from a tube above the coil; and (c) electric strip heaters built into the unit, usually with adjustable panels for closing the supply and return air openings during defrosting.

From observation of the frost and ice

Table 7. Recommended Maximum Temperature Difference and Air Velocity for Various Commodities

Commodity	Maximum temperature difference, F	Maximum air velocity, ft/min
Beef (quarters)	12	40
Beer (wooden kegs)	12	40
Berries	12	40
Butter	12	40
Cheese	12	below 25
Citrous Fruits	12	40
Cut Meats	12	below 25
Dried		
Fruits	16	60
Meats	16	60
Nuts	16	60
Vegetables	16	60
Eggs (crated)	12	40
Fish (iced)	16	60
Flowers (cut)	12	below 25
Fruits		
Dried	16	60
Fresh	12	40
Lamb (halves)	12	40
Lard	20	100
Malt	20	100
Maple sugar	20	100
Maple syrup	20	100
Meats		
Dried	16	60
Hams (fresh)	12	40
Loins (fresh)	12	40
Sausage (bulk)	12	below 25
Smoked	16	60
Nuts (dried)	16	60
Onions	16	60
Oysters	16	60
Pork (halves)	12	40
Poultry (fresh)	12	40
Sausages	12	40
Veal (halves)	12	below 65

formation of a unit operating in a 35 F space, it might appear that the ice would stop the air flow in a short time on a unit operating in a 0 F space. Actually this is not true, since the frost formed in a 35 F space is wet and sticks to the fins, while the frost formed in a 0 F space is snow-like in form, and a considerable quantity of it is blown off the surface.

There are many opinions as to the correct location of the unit in the cooled space for the most satisfactory performance. For walk-in coolers the recommendation of one manufacturer aims to draw the hot, moist air entering the door up through the coil in order to remove the excess moisture from the air with the coil instead of having this moisture condense upon the cold products. In selecting the location of the unit, sufficient space must be provided between the rear of the unit and the wall so that the air flow will not be restricted (**Fig. 10**).

In a specific fixture with fixed evaporator and fixed air temperatures, the dehydration of a moist product is considerably greater with a unit cooler than with gravity air flow finned surface, due, of course, to the more rapid air circulation with the unit cooler. For this reason most manufacturers list maximum permissible temperature differences for various products in their application literature (**Table 7**).

Recommended minimum design evaporators vary between 20 to 25 F in spaces to be kept at about 34 F. Lower evaporator temperatures generally result in an icing problem.

b. The **heat transfer** rate of finned surface with forced air circulation is affected by so many variables that it is difficult to suggest any specific values that could be depended upon. Manufacturers' values based on unbiased tests made in accordance with the ASRE Standard 25-44 should be acceptable.

c. Drier coils are sometimes used to provide superheating surface. They will not eliminate suction line sweating, since the gas temperature can be raised only to the fixture temperature and in most instances this temperature is lower than the dew point of the outside space.

d. The refrigerant flow through the unit cooler is generally controlled by a **thermostatic expansion valve.** When liquid-gas heat exchangers are used, the thermal bulb of the thermostatic expansion valve should be located on the evaporator side of the heat exchanger to reduce pressure cycling to a minimum. The heat exchanger does not increase the efficiency of the evapora-

tor, but does increase the condensing unit capacity due to liquid sub-cooling. Also, the liquid in most cases will raise the suction gas temperature leaving the fixture to the point where the line temperature will be above the dew point of the outside space and little or no sweating will occur.

Most engineers recommend that the unit cooler fan be operated continuously in order to defrost more rapidly during the off cycle.

Brine Spray Units

9. Brine spray units are used for many applications where coil temperatures below freezing are required. These units use either bare pipe or finned type coils. They usually are floor mounted. A pump draws the brine from the drain pan and forces it through nozzles over the coil. These units, built much like evaporative condensers, have replaced pipe coils and bunker spray units in packing houses, breweries and other installations. When using them it must be remembered that sodium brines have relatively high freezing temperatures. Calcium brines have lower freezing temperatures but it is not desirable to use calcium brine units for spaces containing exposed food products.

Example of Use of Tables in Design

Assume a storage room with a load of 7 tons at −10 F evaporator and 10 F room temperature, no heat exchanger being used; calculate the following using 2-in. pipe in 16-ft lengths with an overall heat transfer factor (U) of 1.6 Btu per sq ft, hr, deg F.

1. *Required length of pipe.* The area required is $(7 \times 12{,}000)/(1.6 \times 20) = 2{,}625$ sq ft. The linear feet per sq ft being 1.61, this calls for 4,226 ft, to which an allowance of 25% may be added for safety, making 5,280 ft.

2. *Length allowing for the return bends.* This amount of pipe in 16-ft lengths will total to $5{,}280/16 = 330$ passes. For 2-in. pipe, each return bend by Table 6 equals 7 ft, making a total equivalent length of pipe of 2,310. The total equivalent length is now $5{,}280 + 2{,}310 = 7{,}590$.

3. *Design with three expansion valves.* The equivalent length per feed in this case will be $7{,}590/3 = 2{,}530$ ft, the load per valve amounting to 28,000 Btu per hr. In Table 4 it is found that the maximum allowance is 19,900 ft, subject, however, to the multiplier of Table 5 which for the conditions stated is 1.23. Thus the allowable load is $19{,}900 \times 1.23 = 24{,}500$. The actual being greater, it appears that more than three expansion valves will be needed.

4. *Design with four expansion valves.* In this case the equivalent length per feed is 1,897 and the load 21,000 Btu per hr. Table 4 shows that 22,900 is permissible while the multiplier is still 1.23, making a total of 28,200, or more than the actual.

The minimum length per feed is to be based on the minimum gas velocity from the valve of 400 fpm, with the exception of $\frac{3}{8}$ and $\frac{1}{2}$-in. O.D., where it should be 200 fpm.

33. COMMERCIAL REFRIGERATORS

LARGE refrigerators may be divided into eight classes: restaurant, large home, grocery, bottle, bakery, biological, frozen food and florist.

Restaurant Refrigerators

1. Restaurant refrigerators are built in 15, 20, 30, 40, 60 and 80-cu ft sizes. Because many of the smaller sizes are used behind counters the depth must be limited to about 28 to 30 in., depth of larger sizes being 33 to 34 in. The popular height is 72 in., allowing accessibility on the top shelf as well as on the lower shelf.

The **interior** of metal refrigerators should be lined with acid-resisting porcelain as it is easily cleaned and not affected by moisture or food. The exterior, where exposed to handling or rubbing, especially the doors and corners, should be made of porcelain, stainless steel, or other material not easily affected by abrasion.

Many restaurant men prefer **wood construction** refrigerators. The interiors are usually spruce-lined, sanded smooth and finished with clear odorless alcohol shellac, and the exterior is golden oak. Hardware should be rustproof, and of the self-closing type. Some restaurants require fasteners that can be padlocked, and most modern fasteners have this feature. Reach-in doors should be equipped with grease-proof gaskets and fitted to make an air-tight seal.

In all wood refrigerators, wood slat shelves are used, and in metal refrigerators porcelain slat or wire shelves are used. These have an adjustable feature so that they can be raised or lowered, giving flexibility to the refrigerator. An automatic light that turns on when the doors are opened is a very convenient feature on modern refrigerators.

The **coiling** should be designed to maintain temperatures of 36 to 42 F and the circulation should be designed to prevent interchange of odors. A fairly high humidity (70 to 85%) is required for all cut and uncovered foods to avoid excessive dehydration.

Gravity coiling is fast giving way to **forced air coiling.** Some manufacturers use a panel-type coil on the back wall, drawing warm air in at the top front and forcing it out at the bottom. Other manufacturers use a coil behind the door mullion which takes the warm air in at the top and forces it out at the bottom, and still others install a ceiling type unit in the center of the ceiling. This unit draws the air in at the center and forces it out the sides. The most satisfactory and successful unit is one that does not blow directly on the products in the refrigerator, but gives uniform circulation and is not greatly affected by service.

In view of the fact that restaurant refrigerators usually go in a hot kitchen, may stand next to a hot cook stove and are subjected to heavy service, they should be designed to operate satisfactorily in an ambient as high as 120 F.

Large Home Refrigerators

2. Refrigerators for large homes are usually of the restaurant type, depending upon the requirements. Many large homes require a refrigerator larger than the domestic type, but having the same general arrangement of ice-freezing trays, frozen-food storage capacity, and a large high-temperature storage compartment.

The best refrigeration equipment for some of the larger homes is the combination of a restaurant-type refrigerator for high-temperature storage and a frozen-food refrigerator or chest for ice-making and low-temperature storage.

Grocery Refrigerators

3. The general construction and sizes of grocery refrigerators are about the same as those of restaurant refrigerators. They differ in that they are equipped with glass doors. Another type of grocery refrigerator has sliding sash on the upper front and

storage doors in the bottom. It is brilliantly lighted with fluorescent lights, and because of the sliding doors is frequently used in self-service markets.

Products usually displayed in grocery refrigerators include milk, butter, cheese, eggs, lard, oleomargarine, yeast, soft drinks, etc. The milk compartment should be maintained below 40 F, whereas temperatures up to 45 F are satisfactory for other items. Cut items require a high humidity and a gentle circulation.

Heat loads for grocery refrigerators are usually figured for a 90 F ambient, and the machine specified to operate an average of 16 hr per day in an average ambient temperature of 90 F.

Bottle Refrigerators

4. Bottle refrigerators are used extensively where beer is sold in bottles or cases. Grocery-type refrigerators are often used as bottle refrigerators. However, when this is done, it is necessary that the coiling and machine be specified for bottle loads. The bottle refrigerator has these advantages over other types of bottle storage: (1) it will hold a large quantity of bottles; (2) the bottles are easily accessible on all shelves; and (3) if glass doors are used, they make a good display.

The heat load and machine specifications for bottle refrigerators will depend upon the customer's requirements. If only one load of bottles is to be cooled every 24 hr, the machine should take care of the refrigerator heat gain, including service, and at the same time cool the bottles. If the refrigerator is to be refilled two or three times a day, the machine and coils should be selected to fit these requirements. Since humidity is not a factor in bottle cooling, a greater temperature difference can be used than in restaurant refrigerators, in many cases making it unnecessary to redesign the coil for bottle cooling. Forced air coils are fastest and most efficient for this application due to positive, continuous circulation of the cooled air.

Bakery Refrigerators

5. It has only been in recent years that bakers have recognized the value of refrigeration for retarding dough as well as for keeping ingredients. The bakery refrigerator may be two or three compartments wide, or multiples of these. General construction follows closely the design and construction of restaurant refrigerators. The compartments for retarding doughs are equipped with slides for supporting the trays, the average tray being 18×26 in.

Coiling in bakery refrigerators is of the utmost importance. Coils must be designed to maintain humidities approaching 100%, to take heat out of the dough quickly, and maintain an average temperature of 36 F. In addition, the circulation of air must not be directed on the dough.

The capacity of the condensing unit required to cool the bakery refrigerator will depend upon the room temperature, size of refrigerator, amount of dough to be retarded, and service on the refrigerator.

Biological Refrigerators

6. Biological refrigerators are used for the refrigeration and storage of biological supplies, serums, etc. The general size and construction are similar to those of restaurant refrigerators except that instead of shelving drawers are used for storing the supplies. This helps in segregating the different supplies and at the same time increases accessibility. Average temperatures of 35 to 36 F are required. The humidity should be held below 50% to prevent labels coming off the bottles and packages. A thermostatic control is desirable and the bottoms of the drawers should be perforated to help maintain uniform temperatures. The down draft type of coil is quite satisfactory for this usage.

Because the preservation of biological supplies is of the utmost importance and refrigeration must be maintained under all conditions of ambient temperature, the machine should be specified to give satisfactory refrigeration in an ambient temperature as high as 120 F.

Frozen-Food Refrigerators

7. Frozen-food refrigerators are newcomers to the commercial refrigeration market. They are so new, in fact, that there are no standard sizes. Greater care must be taken in the design and construction of these refrigerators than with higher temperature refrigerators. They may be wood exterior and interior, but it is likely they

will follow the trend of the higher temperature refrigerators and be metal exterior and interior.

The **insulation** should be at least 4 or 5 in. thick, depending on the type used, and must be sealed vapor-tight to prevent infiltration of moisture. The coiling may be built into the liner, but the popular arrangement is to use plate coil shelves which serve two purposes.

Since cold air will spill out like water when the door is opened on an upright type refrigerator, means must be taken to prevent this by the use of **inner doors** made of metal, glass, plastic, fibreboard, or drawers. Great care must be taken that the refrigerator door seals tightly all around. It should be equipped with freeze-proof type gasket. The hinges and fastener must be heavy enough to prevent sagging, and the insulation of the door must be sealed the same as in the rest of the cabinet.

The condensing unit should be selected to maintain an average temperature of 0 F in the refrigerated compartment with not more than 16 hr running time per day in a 90 F room. If there will be a freezing load, it must be considered when specifying the condensing unit.

Frozen-food chests. The simplest form of frozen-food container is a chest with lift lid. The construction should be similar to that of the upright type frozen-food refrigerator. Inner doors are not necessary, as the cold air will stay in the well, a definite advantage which saves refrigeration and simplifies the design. The disadvantage is that food stored in the lower part is not easily accessible.

Here again the coiling may be built into the liner or it may be made up of plate coil dividers. Each arrangement has its advantages. Proponents of the liner-type coiling claim that the coiling picks up the heat before it can get to the stored product and also that it allows more flexibility of the interior arrangement than do stationary plates. On the other hand, the liner coil increases the temperature difference between the two surfaces of the insulation, resulting in greater heat leak.

The condensing unit for this application should also be selected to run not more than 16 out of 24 hr in a 90 deg room, and if there is a freezing load it should be considered when specifying the condensing unit.

Florist Refrigerators

8. Basically, florist refrigeration is divided into three classes: display; storage of flowers, greens, and materials ready for use; and storage of bulbs and plants. No retail florist should be without a refrigerated, well-lighted display.

Since the flowers are sold from the display refrigerator, accessibility is important. The doors must open easily and be so arranged that flowers can be put in and taken out of the refrigerator quickly, and with the least disturbance to other stock already in it.

Many florists require both **display** and **storage,** in which case the display refrigerator is installed in the front part of the shop, and a walk-in cooler installed in the workshop in the rear. The storage cooler should be arranged with shelving and equipped with high humidity coils. It can be used for retarding bulbs, for extra stock, finished pieces, and miscellaneous items.

The wholesaler of florist stock may require both display and blind storage, depending upon his operation. For blind storage, walk-in coolers equipped with shelves and high humidity coils are recommended. The construction may be wood or metal. In the case of wood construction, care should be taken to prevent swelling and warpage due to high humidities inside the refrigerator. Florist display refrigerators should have 3-in. insulation in the walls and three thicknesses of glass in the display windows.

If gravity coils are used, they must be designed to maintain high humidity conditions. This is accomplished by operating them at a high back pressure and consequently a high coil temperature. At the same time, they must be baffled so that the intensity of the circulation does not strike the petals. Certain flowers such as gardenias will not stand circulation of air and must be kept in an enclosed space. The favored method is to keep them in a small glass or cellophane box on the shelf inside the refrigerator.

Forced air coils should preferably be of the panel type, and designed to maintain a high humidity and not blow on the flowers.

Certain precautions are necessary to

prevent damage by the refrigeration itself If a pressure switch is used for control, a small leak or valve freeze-up could cause the machine to run continuously and freeze the stock.

To guard against damage by freezing, the florist refrigerator should be equipped with a thermostat which automatically turns off the machine when the temperature for which it is set is reached. Refrigerants such as sulfur dioxide and ammonia are detrimental to flowers and should be avoided where they are likely to come in contact with the flowers should a leak occur.

Until recently temperatures of 45 to 50 F were considered satisfactory for florists' stocks. It has been demonstrated, however, that some flowers hold better at 40 to to 45 F and that temperatures approaching 32 F are desirable for retarding bulbs.

Machines for florist refrigerators should be of the high back-pressure type, and specified to operate approximately 50% of the time in a 90 F room.

Walk-in Coolers

The refrigerated walk-in cooler originated more than fifty years ago. Wood coolers with cork insulation are still popular and are difficult to improve upon for all-round efficiency, although white porcelain, aluminum, stainless steel, or Dulux exteriors and lighter but efficient insulations are now widely used.

Most coolers are built in sections so they may be easily transported and installed. The present trend is toward metal exterior and interior coolers built up of standard small sections and finished in Dulux or other suitable finish. This type of construction allows for efficient production, easier handling, easier cleaning and more flexibility of sizes.

Standard size coolers have been 4×6, 5×6, 7×5, 8×8, 10×8, 10×10 and 10×12 ft. With the new trend the depth dimension is limited only by the length of the top and floor sections, and the width may be any desired dimension as long as it is a multiple of the standard sections.

Three inches of insulation for wood construction coolers and five for metal construction coolers have been generally accepted to be the most efficient for general storage temperatures of 35 to 40 F, considering insulating efficiency and maximum storage space. The height depends upon the coiling and the wall thickness. The clear height of the cooler inside should be at least 6 ft so that tall men can walk in without stooping over.

The hardware on the doors should be of heavy non-rusting metal and of the self-closing type. The fastener on the entrance door should be arranged so that it can be easily opened from the inside. Some coolers are equipped with a bull's-eye light with switch on the outside door panel controlling the light on the inside of the coolers.

Meat Coolers

9. In meat markets reach-in doors with glass display are popular, both for their display value and facility. The entrance door may be on the end or in the front.

Shelving is placed behind the reach-in doors for the storing of lard, butter, eggs, and smaller pieces of meat. The walls of the cooler are equipped with meat rails, usually two per wall, for hanging halves and quarters, or with shelving.

The most successful coiling provides a constant, gentle circulation of air that does not blow on the meat with intensity. At the same time a relatively high humidity and a small temperature differential between cut-in and cutoff of the machine should be maintained. The coiling should take care of maximum load requirements, yet it should not cause excessive humidity and consequent slime during the off cycle. Coolers with overhead coiling are made $8\frac{1}{2}$ to 9 ft high and are equipped with suitable louvered drain pans properly insulated to prevent condensation. New modern coolers are being built with forced air circulation, some with ceiling type units in the center of the cooler ceiling, and others with a down draft type unit on the cooler wall.

Vegetable Coolers

10. Vegetable cooler design and construction are essentially the same as for meat coolers, the main difference being that here the shelving takes the place of meat rails. The floor should be metal covered, preferably with galvanized iron, to prevent slipping. Floor racks over the

metal floor are necessary to prevent slipping.

In the past vegetable coolers have been held at higher temperatures than meat coolers, a satisfactory procedure for short-time storage. However, it has been proved that temperatures ranging from 34 to 40 F are more suitable for most vegetables, especially if they are to be stored for more than several days.

Both gravity and forced air coiling are satisfactory. The coils must be selected to maintain a high relative humidity, and in the case of the forced air coil, the intensity of circulation should not be directed on the vegetables. If large quantities of warm vegetables are placed in the cooler at one time this extra load should be considered when specifying the coil and machine equipment.

Beer Coolers

11. Frequently walk-in coolers are used for cooling bottled and keg beer. Coolers of this type are known as blind storage coolers as they have no windows or small service doors. They may be of wood construction or of the new metal construction, both inside and out, with three inches of insulation and equipped with large capacity forced air coils for quick cooling. Since air velocity, direction of circulation and humidity are not factors, forced air coils blowing the circulation on the bottles or kegs are quite efficient. Some coolers in which a faucet and drip catcher are attached to the wall are also used as dispensers, and the beer is drawn directly from the kegs in the cooler.

If keg beer is kept in the cooler, normal heat load requirements are satisfactory. However, if bottled beer is kept in the cooler, the additional load of the number of bottles stored per day must be considered.

Restaurant Coolers

12. Walk-in coolers are a necessity in large restaurants, hotels and institutions. A restaurant of normal size has need for only one cooler for storing meats, vegetables, and dairy products. Larger restaurants, hotels, and institutions should have a three-compartment cooler of sufficient size to store meats in one compartment, vegetables in another, and dairy products in the third. The general construction is the same as for meat coolers but without display windows.

Wood exterior and interior, porcelain exterior and wood interior, and metal exterior and interior coolers are used, although many restaurant men still prefer the wood exterior coolers. In general, the construction is the same as for meat coolers. Since these coolers will be installed in or near the kitchen, which may be very hot and will be subjected to heavy service, the heat load requirements are greater than for other types of coolers. The machine and coils should be selected to maintain proper temperature in the cooler in ambient temperatures as high as 120 F.

Frozen-Food Coolers

13. The increasing popularity of frozen foods has created a need for a low-temperature cooler suitable for storing quantities of frozen foods. This cooler is necessarily of heavier construction, at least 6 in. of insulation being desirable. Great care must be taken to seal it with hydrolene, vaporproof paper, or other suitable sealing material against the penetration of moisture vapor. The joints must be designed to make an air-tight fit and at the same time minimize conduction of heat through the framing.

The entrance door must be designed so that it will seal perfectly without danger of freezing shut. To accomplish this some manufacturers are using an outside overlap door. The hinges and hardware must be extra heavy to carry the load of the door.

Wood floor racks are required to allow circulation under the packages piled on the floor and at the same time prevent slipping.

Cooling may be accomplished with plate coil stands, overhead plate banks, plates on the side walls, water defrost blower coils, or hot gas defrost blower coils.

If blower coils of either the water defrost or hot gas defrost are used, they should be specified as not more than 10 deg temperature difference to prevent excessive accumulation of frost on the coils.

If there will be a freezing load, this should be added to the heat load of the cooler when specifying the machine equipment. The condensing unit should be specified to have sufficient capacity at the suction pressure at which it will operate.

Unless the condensing unit has a built-in separator, it should be equipped with one.

Bakery Coolers

14. Bakers are fast learning the benefit of dough retarding, and are making use of it on a larger scale. Many of them started out with a small dough-retarding refrigerator, and now require a larger bakery cooler, with an entrance door and ramp on one side for running racks and troughs into the cooler, and a row of reach-in doors and bakery slides on the other side for retarded dough. While the design of this cooler is different, the general construction is the same as that of the meat coolers.

The coiling should be of a type that does not direct the intensity of the air on the surface of the dough, but will maintain temperatures above 32 F and below 36 F, will maintain the humidity above 85%, and will take the heat out of the dough quickly.

Chill Bars

15. Chill bars are counter-type beverage dispensers used for cooling and dispensing bottled beer or other beverages. They come in various lengths, the most popular being 6, 8, 10, and 12 ft. The average height is 36 to 38 in. so that they will slip easily under a standard bar. Unless installed under a bar, they are usually equipped with Formica or similar acid-resistant and burn-proof top. The exterior may be Formica, metal or wood and the interior rust-proof metal. The back is equipped with sliding sash set at an angle to provide easy access to the interior.

Most chill bars are of the dry refrigeration type in which bottled beverages are cooled by plate or forced air coils. Forced air cooling permits easier cleaning and provides more flexibility in the arrangement of the bottles. Since many states restrict sales of alcoholic beverages to restaurants, a water-cooling attachment should be optional.

Refrigerated Display Cases

Refrigerated display cases are used for displaying meats, vegetables, delicatessen items, dairy products, candy, and frozen foods, each requiring its own special design and arrangement.

There has been a revolutionary trend in display case design during the past six or eight years. There are now available two distinct types of display cases—the closed and the open self-service type.

Closed-Type Meat Display Cases

16. In general, there are three types of closed-type refrigerated meat display cases; the **single-duty type** which has a display compartment only, the **double-duty type** which has a display and a storage compartment, and the **super-market case** which is similar to the single-duty type but is deeper and provides more display.

Display case lengths run from 4 to 18 ft but the standard or popular lengths are 5, 6, 8, 10, 12, 14 and 16 ft. The standard height for meat cases is 50 in. and the standard depth 34 in., with the exception of super-market cases which are 36 to 42 in. in depth.

The **contour** of the case is important. Consideration must be given to good visibility of the product displayed, whether it be on the main shelf or mezzanine shelf, and at the same time the slope of the front glass must be such that undesirable reflections will be at a minimum. Accessibility for the butcher is also important. The drain arrangement should be such as to facilitate cleaning of the case and cleaning of the trap.

Among the first problems that confront the design engineer are the best **wall thickness** and **thickness of insulation.** The thicker the wall, the less will be the heat flow. However, there is a practical minimum, for shelf space is a vital factor. Thickness of insulation is also important for cases with gravity circulation. Sufficient heat must leak through the walls, when there is no service on the case, to cause the machine to cycle frequently enough to maintain circulation and prevent the meat from sliming in normal temperatures. On the other hand, too much leak will raise operation costs. Three inches has been found to be the most efficient thickness of insulation, when available shelf space, preservation of stock, and cost of insulation in heat units per dollar are all considered.

The number of **panes of glass** to use in the front of the case is also important to

the design engineer. The effectiveness of the display is reduced by the amount that the light is reduced in intensity or distorted in color in passing through the panes of glass. Each polished surface disperses 4.6% of the light passing through it, so there is a definite reason for using as few panes of glass as economy and good refrigeration will permit. Since glass is not a good insulator itself, dependence for insulation must be placed upon the air spaces between the panes. It is generally accepted, and tests have proved, that an air space of $\frac{1}{2}$ to $\frac{3}{4}$ in. gives maximum insulating value without permitting circulation between the glasses. As indicated by **Fig. 1,** a design

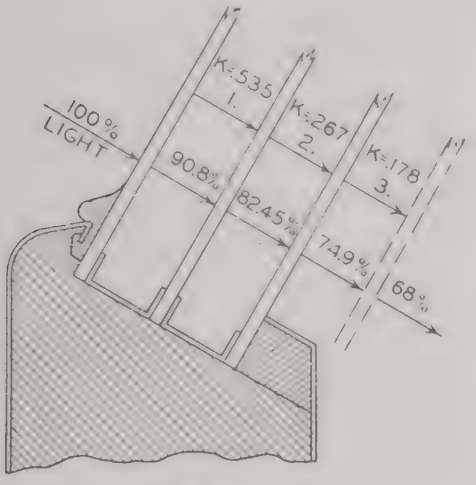


Fig. 1

employing three panes of glass allows good visibility and is economical.

According to the modern trend in lighting, display cases are supplied with **fluorescent lights.** Since fluorescent lamps give off much more light at a fraction of the wattage input taken by incandescent lamps, it is possible to install them inside the display fixture, a procedure which allows more efficient illumination and does away with undesirable reflections. Cases with mezzanine shelves cast a dark shadow on the main shelf when the top lights are turned on, so most manufacturers supply lights to be fitted underneath the mezzanine shelf. Care should be taken that heat from these lamps does not interfere with the circulation inside the case. Both hot cathode (low voltage) lamps and cold cathode (high voltage) lamps are used.

Glazing is an important construction detail, for if the front glasses become fogged, the display value of the fixture is nullified. It is difficult to seal the front glasses so that fog will not collect between them. Two methods of doing this are: (1) to seal the glasses tightly; and (2) to leave an opening to the inside of the case at the top and bottom, to allow breathing to the inside of the case. Because of its polished surface and freedom from imperfections, $\frac{1}{4}$-in. plate glass has been used extensively. However, some manufacturers use one sheet of $\frac{1}{4}$-in. plate glass on the outside and two sheets of $\frac{3}{16}$-in. sheet glass on the inside. Hard rubber sash glazed with two or three thicknesses of double-strength glass are generally used on the back of the case.

The interior of the case should be constructed of porcelain enamel on steel, stainless steel, or other material that can be washed easily and will not scratch, stain, or corrode. The drain trap and drain pipe must be sealed to prevent leakage of moisture into the insulation.

Most double-duty display cases are cooled with direct-expansion type finned coils in the top and in storage compartment. Single-duty types use bare tube or plate coils underneath the trays on the main shelf.

In the meat display cases employing **gravity circulation** with a top coil it is important to have a drip pan underneath the top coil, not only to catch the defrost water from the coil but also to guide the circulation. The efficiency of the entire system depends greatly upon how well the drip pan allows the warm air to circulate through the coil. In modern cases this drip pan is split or louvered, to allow free passage of air from the top coil. The drip pan must also be insulated to prevent condensation on the under side when the doors are opened. Most drip pans employ an air space of $\frac{3}{8}$ to $\frac{1}{2}$ in. between two thicknesses of metal having a moisture-resistant finish on the bottom surface.

Some case manufacturers use **forced air circulation** in which finned coils are installed in the bottom of the case and air is forced upward through the coils to the top of the case and then back to the fan through ductwork. Another method is to install a blower coil at one end of the case.

Air is drawn in along the horizontal length of the case in the top compartment and dispersed into the bottom compartment. Still another method is to install a blower coil in the center of the case, force the cooled air out on the floor, and then pull it vertically into a flue at the top of the case. The main point to keep in mind is that a high humidity must be maintained and the intensity of the air cannot be directed on the products.

Single installations of one condensing unit and one case usually give best results. The refrigerant is fed to the coils through a metering device, such as a thermostatic expansion valve or capillary tube similar to that used on domestic refrigerators. The control may be either of the pressure or temperature type. The recent development of a control that cuts in by coil temperature and cuts out by case temperature is being used successfully.

Temperatures generally accepted for meat display cases are 40 to 45 F on the mezzanine shelf, 36 to 38 F on the main shelf, and 34 to 36 F in the base. Frequent cycling with small variation in temperature and humidity between on-and-off cycles is desirable for best results. A long on-cycle tends to pull the moisture out of the meat and a long off-cycle tends to put it back on the surface of the meat. The first results in dehydration and the second in sliming.

The heat load on a refrigerated display case will vary with its construction. Those cases well insulated and with the sash properly sealed will have less heat loss. The accepted method of determining heat loads is to figure the actual heat leakage through the walls and through the glass, and add an allowance of from 25 to 60% to take care of service. Thirty-five per cent added for a service load is average. Another method to determine the heat leak is to place a bank of lamps inside the case in a controlled ambient with an integrating watt meter connected with the lines to the lamps. The amount of power required to maintain a definite temperature difference between inside and outside temperature will determine the heat leakage through the walls.

Delicatessen Cases

17. Delicatessen cases are distinguished from meat cases in that a varied type of product is kept in them requiring additional shelf space. For this reason, delicatessen cases are usually 54 in. high and are equipped with two mezzanine shelves in addition to the main shelf. They also have a storage compartment for storing meats, butter, milk, cheese, and bottled goods. Standard lengths are 6, 8 and 10 ft.

General construction details are similar to those of meat display cases; however, due to the added height the contour is different. The front glasses are wider and the sliding sash on the rear are more nearly vertical, allowing easy access to all the shelves. An average temperature of 40 to 42 F in the display compartment and a temperature of 38 to 40 F in the storage compartment is required.

Closed-Type Dairy Cases

18. Increased attention paid to the sale of dairy products has brought about a new design called dairy cases. The popular type is a self-serve upright model with large sliding display sash in the upper compartment and smaller sliding or reach-in doors in the storage compartment. Some manufacturers arrange the sash to raise vertically through the use of counter balances.

Dairy cases require good lighting and frequently an advertising sign is incorporated into the lighting fixture.

Temperatures of 36 to 45 F are satisfactory, depending on the product displayed. The coiling may be gravity type or forced air, and the machine should be selected to operate 16 hr in a 90 F room with service on the fixture.

One type being used in both large and small stores is shown in **Fig. 2.** This case is a semiportable, self-contained type, which can be moved from place to place in the store with little difficulty.

Closed-Type Vegetable Cases

19. Closed vegetable display cases are usually of the self-service type with sliding sash giving accessibility to the display compartment. They are built in 6, 8, 10, and 12-ft lengths. Most vegetable cases have storage compartments, equipped with solid sliding or swinging doors, underneath the display compartment.

One arrangement is similar to the upright self-service dairy case with the display sash vertical or almost vertical. An-

other popular arrangement is a low case with the main vegetable display shelves about 36 in. off the floor so that the customer has to reach forward only slightly to examine the contents of the shelf conveniently. The sliding sash on this case are almost in a horizontal position.

The construction of the vegetable case is similar to that of meat cases. If a porcelain liner is used, it should be acid-proof. Sliding sash may have hard rubber frames or a small metal binding edge around the glass, depending on the application.

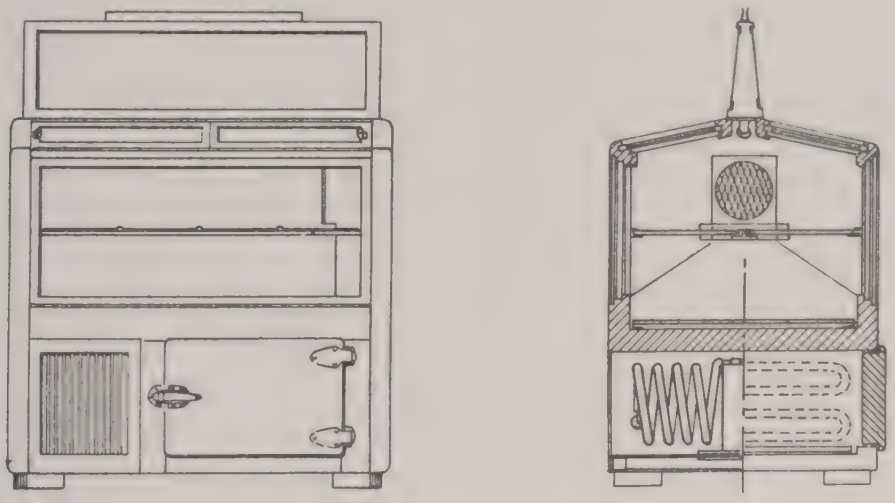


Fig. 2. Two Views of Portable Dairy Display Case

High levels of illumination are desirable to the proper display of vegetables in order to bring out the vivid colors. For this purpose fluorescent lights have proved very satisfactory.

The coiling may be gravity type or forced air. A very high relative humidity, and an average temperature of 38 to 45 F in the display compartment and 36 to 40 F in the storage compartment, are required. To accomplish this the machine is usually specified to operate not more than 16 hr per day in a 100 F ambient.

Closed-Type Candy Case

20. The closed-type candy case is a small self-contained type case 5 to 6 ft long, 30 in. deep and 36 in. high. It is designed to set against the wall or out in the store. The refrigerated storage compartment is accessible from a front reach-in door. It has a small, well-lighted display compartment for displaying candy. The coiling is usually forced-air type designed to maintain temperatures of 65 to 67 F and a relative humidity of approximately 70%. In view of the higher temperatures, insulation of 1 to 2 in. is considered satisfactory. The condensing units are usually the small hermetic type.

Closed-Type Frozen-Food Case

21. With the increasing popularity of frozen foods, display cases are a "must" in food stores. There are two types of closed frozen-food display cases—the type which displays the frozen foods but is not accessible to the customer, and the self-serve type which provides display and at the same time allows the customer to help himself. Both types are similar in construction to the frozen food chest.

Cases with an inclined display glass should have at least four thicknesses of glass to prevent fogging over. They should be well lighted and the coiling should be designed to keep the products displayed at the proper temperature.

The self-serve type case may have flat or inclined sliding doors, friction type or on rollers. The doors should be designed to prevent freezing and sticking. Good lighting is essential to good display and most manufacturers incorporate advertising and price display merit into the lighting fixture.

The closed-type frozen-food case with inclined glass display usually has a plate coil shelf for displaying the merchandise, and the coiling for the storage compart-

ment as well as for the self-service case is similar to that of frozen-food chests. A temperature of 0 F is required for the safe storage of frozen foods.

The condensing unit should be specified to operate not more than 16 hr per day in a 90 F room.

Open-Type Self-Serve Display Cases

Open-type display cases are essentially wells of cold air into which the products are placed.

Open Vegetable Cases

22. The open type of self-service vegetable and fruit display and storage case is a development of the last decade. It is built both single-duty and double-duty in varying lengths, 6 to 12 ft long, and is approximately 40 in. high and 36 in. deep from front to back. It consists of little more than an open-top compartment or tank with insulated ends and back wall, and with glass front for display. Mirrors are placed above the back wall in a sloping position to multiply the display and give the impression of a greater mass of produce. Coiling may be plates placed around the inside lining, including the floor, or as dividers. Some manufacturers use a finned or plate coil on the back wall in conjunction with the liner coils to create a very slow and gentle circulation in the cabinet. Another arrangement is to install a fin coil in a vertical duct back of the mirrored section so that a thermal syphon system is created. The coils are so arranged as to maintain a line of refrigeration even with the top of the display.

The vegetables are displayed at such a level as to be most convenient for inspection by the customer, and at the same time to be seen for a long distance by any prospective customer.

With this type of case it is necessary to maintain a very high relative humidity to keep the product in its original condition and avoid dehydration.

Vegetables inside the refrigerated zone become chilled and the top, being entirely open to the warm air of the room, allows moisture in the warm air to be condensed on the vegetables. Contrary to most expectations, there is no great exchange between the warm and the cold air.

Since there is a gradient of temperature between the bottom and the top of the display section in the open type of case, due to the very little amount of circulation allowed, vegetables requiring lower temperatures should be stored in the lower part of the compartment, and those requiring the higher temperatures placed higher in the case. In general it can be said that the temperature range will vary from 35 to 46 F. A short running time and comparatively long off period on the operation cycle of the compressor are essential for proper refrigeration conditions.

High levels of illumination are desirable for the proper display of vegetables in order to bring out vivid colors. For this purpose fluorescent lights are popular and have proved very satisfactory. The same light for illuminating the display can be used to light an advertising display design.

Open Meat Cases

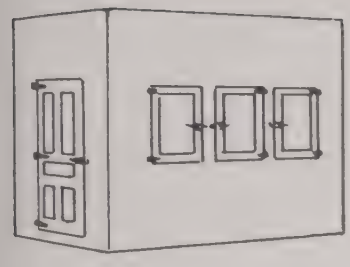
23. Most meat cases are 10 and 12 ft in length and are equipped with mirror top sections to enhance the display. The refrigeration may be accomplished with a coil located in a vertical duct behind an insulated mirror section, making a thermal syphon system, with plate coils in the bottom and around the sides, or with a combination of finned and plate coils. The temperature gradient from the bottom shelf to the top edge of the display glass will depend upon the type of coiling and the depth of the well. The top six inches is usually the buffer zone between the storage temperature and the room temperature and should be used only for short time display.

Since meats will be wrapped in moisture-proof paper a high humidity is not required. Lighting should be of the fluorescent type and of high intensity. This type of case should not be installed where fans or drafts will blow over the case, forcing the cold air out and the warm air in.

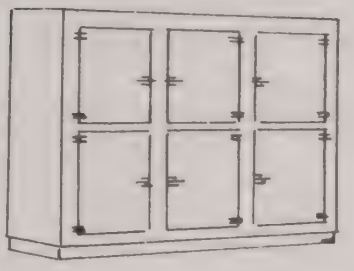
The condensing unit should be of sufficient size to maintain proper temperatures and at the same time cycle sufficiently to allow defrosting even under maximum conditions.

Open Dairy Case

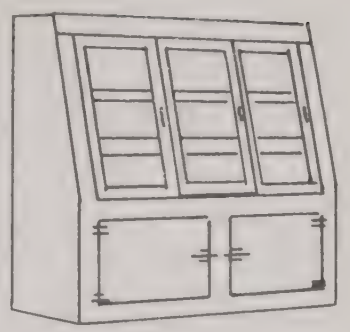
24. The open-type dairy case is similar to the meat case. In many cases it is the same with special step-type shelves to al-



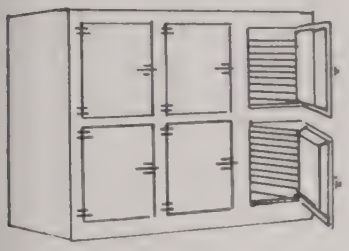
Walk-in Cooler



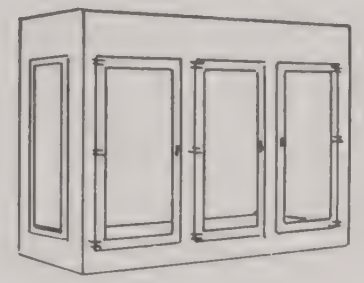
Restaurant Refrigerator



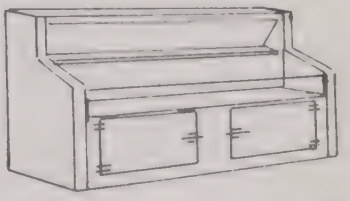
Dairy Refrigerator—Sliding Sash



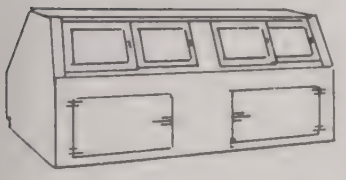
Bakery Refrigerator



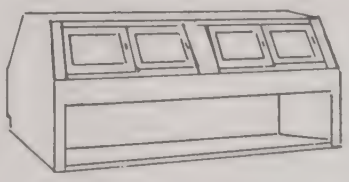
Florist Refrigerator



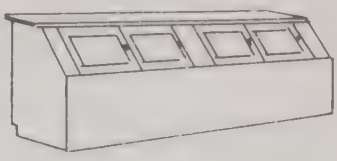
Fruit and Vegetable, Dairy, and Meat Case—Open Self-Serve Type



Meat Case—Double-Duty



Meat Case—Single-Duty



Chill Bar

Fig. 3. Various Types of Commercial Refrigerators

low building the display up higher. Dairy products such as milk and cream should be stored in the lower temperature region of the display compartment, whereas cheese, butter, etc., may be stored in the higher temperature area.

Later style dairy cases have two or even three refrigerated shelves.

Open Candy Cases

25. Open-type self-service candy cases are a recent development. They are very similar in construction to the closed type with the addition of an open-type display above the storage compartment. Refrigeration in the display compartment is accomplished with plate coils on the floor and along the sides or with a fin coil at the back. Care must be taken to prevent condensation of moisture on the packages and also to protect the bottom of the packages from condensation or defrost water.

Open-Type Frozen-Food Display Cases

26. The advance made by the frozen-food industry during the past few years has made a definite demand for frozen-food cases. Freezer chests and ice-cream cabinets have been used for storing frozen foods, but they do not provide display or accessibility.

The design of an open-type frozen-food display case is similar to that of an open-type meat case. Since frozen foods require temperatures of -5 F to $+5$ F, the construction of these cabinets is necessarily

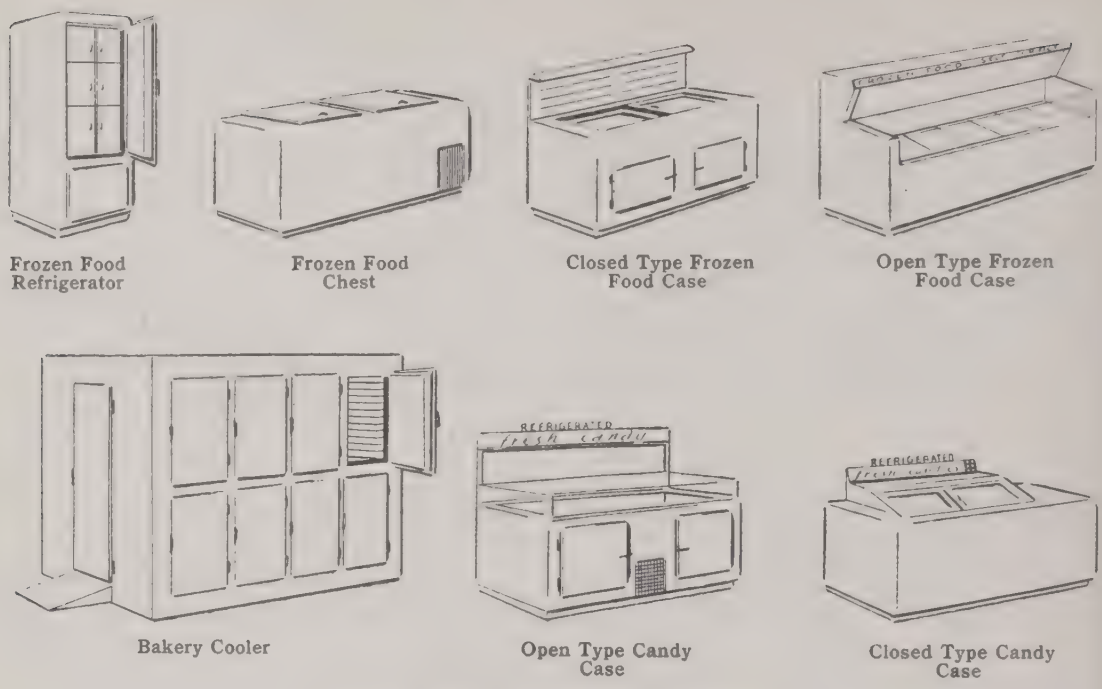


Fig. 4. Types of Refrigerated Cabinets

much heavier. The insulation should be from 4 to 6 in. thick and must be sealed in so it is moisture tight. The exterior and interior lining around the top must be properly separated with suitable breaker strips to prevent heat flow.

The coiling should be designed to maintain proper temperatures and at the same time provide a buffer temperature between the outside air and the top of the display so that there will not be an excessive amount of condensation on the plates. Ice-cream freezer plate type coils, some equipped with eutectic hold-over solutions, are used. These coils may be made up of tubing which is soldered or mechanically fastened to plates, or they may be of the all-steel welded-plate type. Some manufacturers use a fin coil in conjunction with the plate coils to cool and dehydrate the air before it comes in contact with the plates.

The frozen food stored in the case should be stored low enough in the case to avoid being affected by the higher temperatures in the buffer area and the room temperature above. Some cases are provided with a lid that may be closed over night and over weekends. Several manufacturers are experimenting with an automatic defrosting arrangement that periodically defrosts the coils.

Since the product is already frozen when placed in the case it is not necessary to figure a product load. Only the heat leak through the walls and service are considered when selecting the condensing unit. It is most important to protect the case from drafts and fans. High intensity fluorescent lighting is used to light the display brilliantly; it may also be enhanced through the use of a sloping mirror section at the back or with vividly colored lithographs showing pictures of the various frozen foods.

34. SERVICING

ONCE the source of trouble in a refrigeration system is located, it is usually an easy matter to correct it, the main difficulty in servicing work being the location of the trouble. It is important that a systematic method of analysis be followed. It would be considerably better for the service man to take the first half hour to observe the arrangement and operation of the machine than to take a blind shot at the cause of the trouble and make any adjustment in the hope that it will correct the trouble. Such a procedure often adds trouble because the adjustment was not necessary or proper.

When the service man is following a systematic method of trouble shooting, he will first listen carefully to the complaint of the owner or operator of the equipment. This will give him his first clue as to the cause of the trouble. He should then carefully check the complaint to see if it is justified.

In some instances the cause of the trouble will be immediately apparent but in the large majority of cases some measuring of pressures, temperatures and running time will have to be made. In this latter case, it will be necessary to determine the type of machine, since the method of locating the trouble will depend upon the type of the refrigerating machine and the method of control.

Classification of Complaints

For the purpose of this discussion, refrigerating machines have been classified both by the method of refrigerant control and motor control. The refrigerant control methods include thermostatic expansion valve, automatic expansion valve, low-side float, high-side float and capillary tube. Motor controls are classified as to temperature and pressure.

Owner's or operator's complaints may also be classified. These complaints are, in the order they are taken up; (1) Refrigerated space too warm, (2) refrigerated space too cold, (3) refrigerating machine runs too long, (4) water bill too high, (5) electric bill too high, (6) refrigerating system too noisy and (7) odors in refrigerated space.

Before a man does any trouble shooting he should be familiar with the operation of the various systems under all kinds of conditions. The basic systems of controls as shown in Table 1 are especially important. He should know how a system normally operates and what pressures, temperatures and running times to expect under such conditions so that he will readily recognize the abnormal.

To determine the source of the trouble it is desirable to make observations as follows:

1. The temperature of the refrigerated space or of the substance cooled
2. Temperature of the cooling unit
3. Suction pressure
4. Head pressure
5. Temperature of liquid and suction line
6. Sound of refrigerant control valve
7. Running time
8. Noise.

Temperature of Refrigerated Space

The temperature of the cooling unit cannot be determined exactly, but an approximation can be secured by clamping an accurate thermometer to the surface. It will be found that the temperature of the surface will be about 10 deg F higher than the temperature of the refrigerant in the evaporator, as shown by the suction pressure when the condensing unit is running.

The suction pressure will give an indication of the temperature of the refrigerant in the cooling coil. If the suction pressure reading is made at the condensing unit and the suction lines are long, the pressure in the cooling unit will be from 1 to 5 psi lower than at the evaporator.

Table 1. Types of Refrigerant Controls

Refrigerant control	Application	Type evapo-rator	Condensing unit control	Features
Capillary tube	Domestic only	Flooded	Temperature	No moving parts; self-compensating; must have exact refrigerant charge.
Low-side float	Commercial Domestic Multiple systems	Flooded	Pressure-temperature	Simplicity; good heat transfer; evaporator must be kept level; requires comparatively large amount of space and refrigerant.
High-side float	Domestic Commercial	Flooded Dry	Pressure-temperature	Wear on needle and seat is slight because liquid is handled most of time; must have exact refrigerant charge.
Automatic expansion	Domestic Commercial	Dry	Temperature	Requires small space. Easily adjusted. Permits use of easily constructed cooling coil; must be used with steady loads.
Thermostatic expansion	Domestic Commercial Multiple systems	Flooded Dry	Pressure-temperature	Permits use of easily constructed cooling coil; utilizes full length of coil; requires comparatively small amount of refrigerant.

Normally it will be found that in retail fixtures the temperature differential between the refrigerant and air will be from 17 to 27 F. For wholesale storage of eggs, poultry, cheese and the like, the temperature differential should be for normal operation 14 to 17 F; for cut meats, vegetables, fruits, beer in wooden kegs 18 to 22 F; for meats in carcasses and some fruits 21 to 27 F; and for goods where humidity control is not important, such as bottled beverages, food in cans and beer in steel kegs, 27 to 37 F.

Head Pressure

In normal operation it will be found that the temperature of the condensing refrigerant will be about 30 to 35 F higher than the temperature of the air entering the condenser in an air-cooled condensing unit and about 20 to 25 F higher than the water entering a water-cooled unit when the units are operating. These figures are for suction temperatures of from 10 to 20 F. Lower suction temperatures will result in lower head pressures and temperatures while higher suctions will result in higher head pressures and temperatures.

Fig. 1 shows the approximate head pressure of air-cooled condensing units using Freon F-12, methyl chloride and sulfur dioxide.

Liquid and Suction Line Temperatures

The temperature of the **liquid line** will give an indication of the condition of the refrigerant within the liquid line. Normally the temperature of the liquid line is slightly above the temperature of the air or water leaving the condenser. If it is noted that the line is considerably warmer, this condition is probably due to the fact that the line is filled with gas rather than liquid due either to improper operation of the con-

denser or a shortage of refrigerant in the system. If, on the other hand, the line is much cooler than the leaving temperature of the condenser air or water, expansion of the refrigerant is probably taking place in the liquid line. This condition is caused by too great a pressure drop due to too long a line, too small a tube, excessive lift or a restriction such as a kink or partially closed valve.

The **suction line** temperature is normally a trifle lower than the room temperature but it will fall somewhat as the amount of refrigerant circulating is increased. If the line is considerably colder than normal or is frosted, it is an indication that liquid refrigerant is entering the suction line. Such a condition could be caused by improper operation of the refrigerant control valve.

The temperature of the **compressor** will give a good indication of the operation of the system. Normally the compressor head is fairly hot to the touch unless it is water-cooled. If it is not water-cooled and if it is fairly cool there is a good possibility that for some reason or other not enough refrigerant is being circulated. If it is very hot it will be found that the head pressure is excessive due to poor condensation or to air in the system.

The base of the compressor normally will be slightly above room temperature. A very warm compressor base may indicate a **shortage of oil** while a cool base may be caused by excessive oil or liquid refrigerant being slugged back through the suction line from the evaporator.

Normally the refrigerant **control valve** is fairly quiet in operation, there being only a slight noise due to the refrigerant passing through the orifice. However, when gas is passing through the valve instead of liquid a noticeable hissing noise will be heard. If the valve is completely quiet it is a good indication that no refrigerant is passing through.

Automatic condensing units will run anywhere from 5 to 20 hr or more, depending on a number of factors such as air temperature, design of unit and system and the setting of controls. Extremely short cycles or very long periods of continuous operation should be noted as they are indications of improper operation.

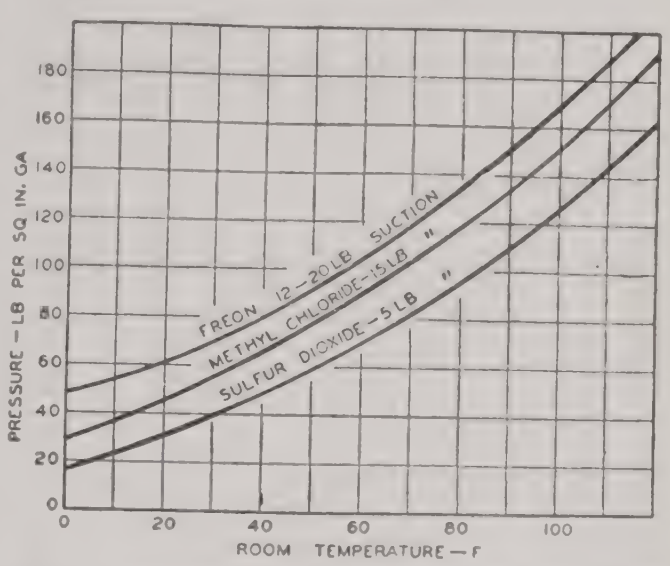


Fig. 1. Approximate Head Pressure of Air-Cooled Condensing Units Using Freon-12, Methyl Chloride and Sulfur Dioxide

Unusual noises in the compressor, motor, drive or refrigerant control valve should be noted as they serve as a good indication of trouble.

Trouble Shooting Charts

The following trouble shooting charts are not intended to serve as an inflexible guide for the service man, but they are designed merely to show one method of approach. Nothing can take the place of a fundamental knowledge of refrigeration and of the normal operation of refrigeration equipment in trouble shooting work. Yet these charts should be of considerable help not only to the man just starting in servicing but also to the more experienced man in pointing out how a logical method of approach will simplify trouble shooting.

The first observation to be made by the service man is how the condensing unit is operating, for example, whether it is operating at all, short cycling or operating

continuously. The next observation is made to determine the type of system, that is, whether it is a high-side or low-side float or thermostatic or automatic expansion valve system and whether it is pressure or temperature controlled.

Once the method of operation and type of system are determined it is then necessary to make detailed observations on the condition of the system. These include the suction pressure, head pressure, temperature of the evaporator, compressor, motor, suction and liquid line, and sound of refrigerant control valve.

It will be noted that the charts are so arranged that all the troubles which apply to any type of system are first taken up, then those applying only to systems with pressure controls, then ones applying to systems with thermostatic or temperature controls, and finally those applying only to particular systems. This arrangement not only considerably reduces the size of the charts but also makes them easier to use.

COMPLAINT: REFRIGERATED SPACE TOO WARM—I

First Observation: Condensing Unit Not Operating
Type System: Any

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—high superheat	No current to condensing unit due to:	
	1. Open switch	1. Close switch
	2. Fuse blown	2. Replace fuse
	3. Power lines dead	3. Have lines repaired
	4. Broken connection	4. Repair
	Suction, temperature, or head pressure control	
	1. Control out of adjustment	1. Adjust
	2. Control contacts corroded	2. Clean with sandpaper
	3. Broken mercury bulb	3. Replace bulb or control
	4. Spring broken or worn out —control stuck open	4. Replace spring or control
	5. Control out-of-level	5. Level
	6. Leaky bellows or joints	6. Repair or replace
	Motor overload switch open due to:	
	1. Belt driving compressor is too tight	1. Loosen belt
	2. Failure of holding mechanism or dirty contact points	2. Replace or repair
	3. Low line voltage	3. Decrease load on line or increase wire size
	4. Bearings too tight or lack of oil	4. Oil or repair
	5. Compressor stuck	5. Repair or replace
	Motor failure	Repair or replace

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—high Evaporator—high superheat	High head pressure cut-out or motor overload switch open due to high head pressures caused by: 1. High air temperature entering air cooled condenser 2. Condensing water temperature too high 3. Air in system 4. Clogged condenser	1. Use an evaporative condenser or cooler supply of air 2. Use an evaporative condenser 3. Purge system 4. Clean or replace condenser
Suction pressure—below normal cut-in point Head pressure—low Evaporator—high superheat	Space surrounding condensing unit is too cold	Partially enclose condensing unit in box. Block off part of condenser

Type System: Any with Pressure Control

Suction pressure—below normal cut-in point Head pressure—below normal Evaporator—high superheat	An obstruction preventing refrigerant from circulating	
	1. Closed liquid line service valve	1. Open
	2. Liquid line stopped by kink	2. Remove kink
	3. Strainer or dehydrator plugged	3. Clean
	4. Liquid line solenoid valve failure	4. Repair or replace
	5. Suction service valve closed	5. Open
	6. Suction line stopped by kink in line or at flare nut	6. Remove kink
	7. Ice, wax or dirt plugging refrigerant metering valve	7. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
	8. Refrigerant charge lost	8. Repair leak and recharge

Type System: Thermostatic Expansion Valve—Pressure Controlled

Suction pressure—below normal cut-in point Head pressure—below normal Evaporator—high superheat	1. Power element has lost its charge	1. Replace expansion valve
	2. Expansion valve out of adjustment	2. Adjust

Type System: High-Side Float—Pressure Controlled

Suction pressure—below normal cut-in point Head pressure—below normal Evaporator—high superheat	1. Float has leak in it and sinks in refrigerant	1. Replace

COMPLAINT: REFRIGERATED SPACE TOO WARM—II

First Observation: Condensing Unit Short Cycles
Type System: Any

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—high superheat Also:	If motor overload switch is tripping	
1. Motor hot	1. Belt driving compressor too tight	1. Loosen belt
2. Lights dim when motor starts	2. Low line voltage	2. Decrease load on line or use larger wire
3. Motor hot	3. Motor too small	3. Replace
4. Motor hot	4. Bearings lack oil	4. Oil
5. Motor may be hot	5. Ground or short circuit in motor winding	5. Repair or replace
Suction pressure—above normal cut-in point Head pressure—high Evaporator—high superheat	Head pressure cut-out or motor overload switch opening due to high head pressure caused by:	
1. Receiver pressure low. Motor and compressor head hot. Liquid line warmer than normal	1. Partially closed discharge service valve	1. Open valve
2. Motor and compressor head hot. Liquid line warmer than normal	2. Air-cooled condenser (a) Dirty (b) Air temperature too high (c) Poor air circulation	2. (a) Clean (b) Install evaporative or water-cooled condenser (c) Remove obstruction
3. Motor and compressor head hot. Liquid line warmer than normal	3. Water-cooled condenser	3.
(a) No water discharged to sewers	(a) No water	(a) Remove obstruction in water pipe
(b) Discharge water very warm	(b) Restricted water flow	(b) Increase pressure or pipe size, remove obstruction
(c) Discharge water very warm and quantity excessive	(c) Water temperature too high	(c) Install evaporative condenser
(d) Discharge water cool and quantity excessive	(d) Corroded condenser	(d) Clean or replace
4. Motor and compressor head hot. Liquid line warmer than normal	4. Evaporative cooled condenser	4.
(a) Hot dry air leaving condenser	(a) No water	(a) Correct trouble in water line or float valve
(b) Hot dry air leaving condenser	(b) Spray nozzle clogged	(b) Clean
(c) Hot dry air leaving condenser	(c) Pump not operating	(c) Repair
(d) Same as (c)	(d) Coil surface dirty	(d) Clean

Observations	Trouble	Remedy
(e) No air leaving condenser	(e) Air inlet or outlet obstructed	(e) Remove obstruction
(f) No air leaving condenser	(f) Fan not operating	(f) Repair
5. Motor and compressor head hot. Liquid line warmer than normal	5. Air in system	5. Purge system
6. Motor and compressor head hot. Liquid line warmer than normal and compressor may be jumpy and slug at start	6. Too much refrigerant	6. Release some refrigerant
7. Receiver pressure low. Motor and compressor head hot. Liquid line warmer than normal	7. Restricted condenser tube	7. Clean or replace
8. Motor and compressor head hot. Liquid line warmer than normal	8. Condenser too small	8. Replace
Suction pressure—normal Evaporator—normal superheat Head pressure—low	1. Evaporator too small	1. Install larger evaporator
	2. Evaporator surface dirty	2. Clean
Suction pressure—rises rapidly when compressor stops Evaporator—suction line frosted Head pressure—low	Compressor is removing refrigerant faster than it can be evaporated due to restricted air flow to evaporator or ice on evaporator.	Remove obstruction to air flow. Defrost

COMPLAINT: REFRIGERATED SPACE TOO WARM—III

First Observation: Condensing Unit Short Cycles

Type System: Any with Pressure Control

Observations	Trouble	Remedy
Suction pressure—rises rapidly when compressor stops Head pressure—low Evaporator—high superheat	An obstruction in liquid or suction line is reducing flow of refrigerant or there is a refrigerant shortage	Remove obstruction. Add refrigerant
Also:		
1. Hissing at metering valve	1. Shortage of refrigerant	1. Add refrigerant
2. Hissing at metering valve. Liquid line temperature decreases with height above receiver	2. Liquid line too small or too great a lift	2. Replace with larger line. Subcool liquid or install heat interchanger
3. Hissing at metering valve and temperature drop across obstruction	3. Strainer or dehydrator partially plugged or too small	3. Clean strainer. Replace dehydrator charge. Replace strainer or dehydrator
4. Hissing at metering valve and temperature drop across obstruction	4. Liquid line partially closed by kink or other obstruction	4. Remove kink or obstruction
5. Hissing at metering valve. Cold liquid line	5. Liquid service valve partially closed or too small	5. Open or replace

REFRIGERATED SPACE TOO WARM III (Continued)

Observations	Trouble	Remedy
6. Suction pressure in evaporator is much higher than at compressor	6. Suction line partially stopped by kinks or other obstruction, or too small	6. Remove obstruction. Replace with larger line
7 Hissing at metering valve and temperature drop across solenoid valve	7. Liquid line solenoid valve too small	7. Replace
	8. Ice, wax or dirt partially closing metering valve	8. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
	9. Differential on control too small	9. Increase differential

Type System: Any with Temperature Control

Suction pressure—rises rapidly when compressor stops Head pressure—low or normal Evaporator—high superheat	1. Differential on control too small	1. Increase differential

Type System: Thermostatic Expansion

Suction pressure—rises rapidly when compressor stops Head pressure—low or normal Evaporator—high superheat		
1. Test for ice—allow expansion valve to warm up to melt ice or tap lightly. Test for wax or dirt—flush valve by allowing large amount of liquid to flow through in a short time	1. Ice, wax or dirt partially closing expansion valve	1. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
2. Same	2. Expansion valve orifice too small	2. Replace with larger orifice
3. Try a short turn of expansion valve adjusting nut first. If this is no good, something else is wrong	3. Expansion valve out of adjustment	3. Adjust
4. Warming feeler bulb raises suction pressure and decreases superheat only slightly	4. Expansion valve power element has partially lost charge	4. Replace expansion valve
5. Warming power element with hands corrects trouble temporarily	5. Expansion valve in too cold a location and power element is cooler than feeler bulb	5. Move expansion valve to warmer location

Type System: High-Side Float with Pressure Control

1. Evaporator partially sweated or frosted	1. Shortage of refrigerant	1. Add refrigerant

COMPLAINT: REFRIGERATED SPACE TOO WARM—IV

First Observation: Condensing Unit Operates Continuously
Type System: Any

Observations	Trouble	Remedy
Suction pressure—above normal cut-in pressure Head pressure—low Evaporator—high superheat	Refrigerant is not circulating at proper rate due to obstruction, inefficient compressor or shortage of refrigerant	
1. Compressor head and liquid line cooler than normal	1. Compressor belt slipping	1. Tighten belt
2. Hissing at refrigerant metering valve. Liquid line cooler than normal	2. Liquid line service valve partially closed or too small	2. Open or replace
3. Hissing at refrigerant metering valve. Temperature drop across strainer or dehydrator	3. Strainer or dehydrator partially plugged or too small	3. Clean strainer. Install new dehydrator charge Replace
4. Hissing at refrigerant metering valve. Temperature drop across solenoid valve	4. Liquid line solenoid valve too small	4. Replace
5. Hissing at refrigerant metering valve. Temperature drop across obstruction	5. Liquid line partially closed by kink or other obstruction	5. Remove obstruction
6. Hissing at refrigerant metering valve and gradual drop in temperature of liquid line	6. Liquid line too small or too great a lift	6. Replace with larger line. Subcool liquid or install heat interchanger
7. Suction pressure in evaporator is much higher than at compressor	7. Suction line partially stopped by kinks or other obstruction, or too small	7. Remove obstruction. Replace with larger line
8. Hissing at refrigerant metering valve. Compressor head cooler than normal	8. Shortage of refrigerant	8. Add refrigerant
9. Test for ice—allow valve to warm up to melt ice or tap lightly. Test for wax or dirt, flush valve by allowing a large amount of refrigerant to flow through valve in short time. Repeat a number of times	9. Ice, wax or dirt partially closing refrigerant metering valve	9. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
10. Liquid line and compressor head cooler than normal	10. Inefficient compressor (a) Leaky compressor valves (b) Leaky piston rings (c) Leaky gaskets between cylinders	10. (a) Repair (b) Repair (c) Repair
11. Compressor speed too low	11. Low line voltage	11. Decrease load on line or increase line size
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—high superheat	Condensing unit capacity too small	Install larger condensing unit

REFRIGERATED SPACE TOO WARM IV (Continued)

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—high Evaporator—high superheat	Condensing unit has reduced capacity due to high head pressure caused by:	
1. Receiver pressure low. Motor and compressor head hot	1. Partially closed discharge service valve	1. Open valve
2. Motor and compressor head hot. Liquid line warmer than normal	2. Air-cooled condenser (a) Dirty (b) Air temperature too high (c) Poor air circulation	2. (a) Clean (b) Install evaporative or water-cooled condenser (c) Remove obstruction
3. Motor and compressor head hot. Liquid line warmer than normal	3. Water-cooled condenser	3.
(a) No water discharge to sewer	(a) No water	(a) Remove obstruction in water line
(b) Discharge water very warm	(b) Restricted water flow	(b) Increase pressure or pipe size, remove obstructions
(c) Discharge water very warm and quantity excessive	(c) Water temperature too high	(c) Install evaporative condenser
(d) Discharge water cool and quantity excessive	(d) Corroded condenser	(d) Clean or replace
4. Motor and compressor head hot. Liquid line warmer than normal	4. Evaporative cooled condenser	4.
(a) Hot dry air leaving condenser.	(a) No water	(a) Correct trouble in water line or water valve
(b) Hot dry air leaving condenser	(b) Spray nozzle clogged	(b) Clean
(c) Hot dry air leaving condenser	(c) Pump not operating	(c) Repair
(d) Same as (c)	(d) Coil surface dirty	(d) Clean
(e) No air leaving condenser	(e) Air inlet or outlet obstructed	(e) Remove obstruction
(f) No air leaving condenser	(f) Fan not operating	(f) Repair
5. Motor and compressor head hot. Liquid line warmer than normal	5. Air in system	5. Purge system
6. Motor and compressor head hot. Liquid line warmer than normal and compressor may be jumpy and slug at start	6. Too much refrigerant	6. Release some refrigerant
7. Receiver pressure low. Motor and compressor head hot	7. Restricted condenser tube	7. Clean or replace
8. Motor and compressor head hot. Liquid line warmer than normal	8. Condenser too small	8. Replace

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—normal superheat	Condensing unit overloaded due to: 1. Hot foods 2. Doors left open 3. Loose fitting doors 4. Poor gaskets 5. Poor insulation 6. Condensing unit and coil too small	 1. Instruct owner in proper operation 2. Instruct owner in proper operation 3. Replace 4. Replace 5. Replace 6. Replace

Type System: Any—Temperature Controlled

Suction pressure—very low and does not rise when compressor is stopped Head pressure—low Evaporator—high superheat	An obstruction is preventing the flow of refrigerant	
1. Liquid line cooler than normal	1. Liquid line service valve closed	1. Open
2. Liquid line cooler than normal	2. Strainer or dehydrator completely plugged	2. Clean strainer. Install new dehydrator charge
3. Liquid line cooler than normal	3. Liquid line closed by kink or other obstruction	3. Remove kink or obstruction
4. Liquid line cooler than normal. Test for ice—allow valve to warm up to melt ice. Tap lightly to free ice or dirt	4. Ice, wax or dirt completely closing refrigerant metering valve	4. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
4. Liquid line cooler than normal	5. Refrigerant metering valve closed due to mechanical failure	5. Repair or replace
6. Suction pressure in evaporator is much higher than at compressor	6. Suction line completely stopped by kink, closed service valve or other obstruction	6. Remove kink or obstruction, open valve
Suction pressure—high Head pressure—normal or low Evaporator—warm Suction line—sweating Hissing at metering valve	Metering valve stuck open and so much refrigerant enters low side that it cannot evaporate at a pressure low enough to give a low temperature	Repair or replace

Type System: Thermostatic Expansion

Suction pressure—above normal cut-in point Head pressure—low Evaporator—high superheat	Refrigerant is not circulating at proper rate due to obstruction	

REFRIGERATED SYSTEM TOO WARM IV (Continued)

Observations	Trouble	Remedy
1. Test for ice—allow valve to warm up to melt ice or tap lightly. Test for wax or dirt—flush valve by allowing large amount of liquid to flow through valve in short time. Repeat a number of times	1. Ice, dirt or wax partially closing expansion valve	1. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
2. Try turning expansion valve adjusting nut slightly. If this does no good something else is wrong	2. Expansion valve out of adjustment	2. Adjust
3. Warming feeler bulb raises suction pressure and decreases superheat only slightly	3. Expansion valve power element has partially lost charge	3. Replace expansion valve
4. Warming power element with hands corrects trouble temporarily	4. Expansion valve in too cold a location. Power element is colder than feeler bulb	4. Move expansion valve to warmer location
Suction pressure—above normal cut-in point pressure Head pressure—low Evaporator—sweating or frosting suction line Crankcase—sweating	Expansion valve stuck open	Repair or replace

Type System: High-Side Float—Temperature Control

Suction pressure—very low and rises slightly when compressor is shut down Head pressure—low Evaporator—high superheat	Shortage of refrigerant	Add refrigerant
Suction pressure—very low and does not rise when compressor is shut down Head pressure—very high Evaporator—high superheat	High head pressure has caused float to collapse and sink, closing refrigerant valve	Determine cause of high head pressure and replace float

COMPLAINT: REFRIGERATED SPACE TOO COLD

First Observation: Condensing Unit Operating Too Long

Type System: Any

Observations	Trouble	Remedy
Suction pressure—below normal cut-in point Head pressure—normal Evaporator—too cold—normal superheat	1. Pressure or temperature control set too low 2. Pressure or temperature control defective—will not cut out	1. Adjust 2. Repair or adjust

Type System: Any—Pressure Controlled

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—no superheat; frosted or sweating suction line	1. Dirt, wax or ice on valve seat	1. Clean. Use oil of lower wax content. Install dehydrator

Type System: Thermostatic Expansion—Pressure Controlled

Suction pressure—above normal cut-in point Head pressure—normal	1. Expansion valve out of adjustment	1. Adjust
Evaporator—no superheat; frosted or sweating suction line	2. Expansion valve feeler bulb loosely clamped to suction line	2. Fasten securely

COMPLAINT: CONDENSING UNIT OPERATES TOO LONG

First Observation: Refrigerated Space Cold Enough

Type System: Any

Observations	Trouble	Remedy
Suction pressure—above normal cut-in point Head pressure—low Evaporator—superheat slightly higher than normal	Refrigerant is not circulated at proper rate due to an obstruction, inefficient compressor or shortage of refrigerant	
Also:		
1. Compressor head and liquid line cooler than normal. Lights dim when motor starts	1. Low line voltage	1. Decrease load on line or install larger line
2. Compressor head and liquid line cooler than normal	2. Compressor belt slipping	2. Tighten belt
3. Hissing at refrigerant metering valve. Liquid line cooler than normal	3. Liquid line service valve partially closed or too small	3. Open or replace
4. Hissing at metering valve. Temperature drop across strainer or dehydrator	4. Strainer or dehydrator partially plugged or too small	4. Clean strainer. Install new dehydrator charge. Replace
5. Hissing at metering valve Temperature drop across solenoid valve	5. Liquid line solenoid valve too small	5. Replace
6. Hissing at metering valve. Temperature drop across obstruction	6. Liquid line partially closed by kink or other obstruction	6. Remove obstruction
7. Hissing at metering valve and gradual drop in temperature of liquid line	7. Liquid line too small or too great a lift	7. Replace with larger line. Subcool liquid or install heat interchanger
8. Suction pressure in evaporator is much higher than at compressor	8. Suction line partially stopped by kinks or other obstruction or too small	8. Remove kinks. Replace with larger line

CONDENSING UNIT OPERATES TOO LONG (Continued)

Observations	Trouble	Remedy
9. Hissing at metering valve. Compressor head cooler than normal	9. Slight shortage of refrigerant	9. Add refrigerant
10. Liquid line and compressor head cooler than normal	10. Inefficient compressor (a) Leaky compressor valve (b) Leaky piston rings (c) Leaky gaskets between cylinders	10. Repair or replace compressor
11. Test for ice—allow valve to warm up to melt ice or tap lightly. Test for wax or dirt—flush valve by allowing a large amount of refrigerant to flow through valve in short time. Repeat a number of times	11. Ice, wax, or dirt partially closing metering valve	11. Clean valve. Install dehydrator. Replace oil with oil of lower wax content
12. Same	12. Compressor speed too low	12. Speed up by changing pulley size or using larger motor
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—superheat higher than normal	Condensing unit capacity too small	Install larger condensing unit
Suction pressure—above normal cut-in point Evaporator—superheat slightly higher than normal	Condensing unit has reduced capacity due to high head pressure caused by:	
1. Receiver pressure low. Motor and compressor head hot	1. Partially closed discharge service valve	1. Open valve
2. Motor and compressor head hot. Liquid line warmer than normal	2. Air-cooled condenser (a) Dirty (b) Air temperature too high (c) Poor air circulation	2. (a) Clean (b) Install evaporative or water-cooled condenser (c) Remove obstruction
3. Motor and compressor head hot. Liquid line warmer than normal	3. Water-cooled condenser	3.
(a) No water discharged to sewer	(a) No water	(a) Remove obstruction in water pipe
(b) Discharge water very warm and quantity excessive	(b) Restricted water flow	(b) Increase pressure or pipe size, remove obstruction
(c) Discharge water very warm and quantity excessive	(c) Water temperature too high	(c) Install evaporative condenser
(d) Discharge water cool and quantity excessive	(d) Corroded condenser	(d) Clean or replace

Observations	Trouble	Remedy
4. Motor and compressor head hot. Liquid line warmer than normal	4. Evaporative cooled condenser	4.
(a) Hot dry air leaving condenser	(a) No water	(a) Correct trouble in water line or float valve
(b) Hot dry air leaving condenser	(b) Spray nozzle clogged	(b) Clean
(c) Hot dry air leaving condenser	(c) Pump not operating	(c) Repair
(d) Same as (c)	(d) Coil surface dirty	(d) Clean
(e) No air leaving condenser	(e) Air inlet or outlet obstructed	(e) Remove obstruction
(f) No air leaving condenser	(f) Fan not operating	(f) Repair
5. Motor and compressor head hot. Liquid line warmer than normal	5. Air in system	5. Purge system
6. Motor and compressor head hot. Liquid line warmer than normal and compressor may be jumpy and slug at start	6. Too much refrigerant	6. Release some refrigerant
7. Receiver pressure low. Motor and compressor head hot	7. Restricted condenser tube	7. Clean or replace
8. Motor and compressor head hot. Liquid line warmer than normal	8. Condenser too small	8. Replace
Suction pressure—above normal cut-in point Head pressure—normal Evaporator—normal superheat	Condensing unit overloaded due to:	
	1. Hot foods	1. Instruct owner in proper operation
	2. Doors left open	2. Instruct owner in proper operation
	3. Loose fitting doors	3. Replace
	4. Poor gaskets	4. Replace
	5. Poor insulation	5. Replace
	6. Condensing unit and coil too small	6. Replace

Type System: Thermostatic Expansion

Suction pressure—above normal cut-in point Head pressure—low Evaporator—high superheat	Refrigerant is not circulating at proper rate due to obstruction	
1. Test for ice—allow valve to warm up to melt ice or tap lightly. Test for wax or dirt—flush valve by allowing large amount of liquid to flow through valve in short time. Repeat a number of times	1. Ice, dirt or wax partially closing expansion valve	1. Clean valve. Install dehydrator. Replace oil with oil of lower wax content

CONDENSING UNIT OPERATES TOO LONG (Continued)

Observations	Trouble	Remedy
2. Try turning expansion valve adjusting nut slightly. If this does no good something else is wrong	2. Expansion valve out of adjustment	2. Adjust
3. Warming feeler bulb raises suction pressure and decreases superheat only slightly	3. Expansion valve power element has partially lost charge	3. Replace expansion valve
4. Warming power element with hands corrects trouble temporarily	4. Expansion valve in too cold a location. Power element is colder than feeler bulb	4. Move expansion valve to warmer location

COMPLAINT: WATER BILL TOO HIGH

First Observation: Refrigerated Space Cold Enough

Type System: Any—Head Pressure Controlled Water Valve

Observations	Trouble	Remedy
Water flow—high Head pressure—normal	1. Water pressure too high 2. Water valve out of adjustment 3. Valve stuck open 4. Condenser is dirty resulting in poor heat transfer	1. Install pressure regulating valve 2. Adjust 3. Repair 4. Clean or replace
Water flow—normal Head pressure—normal	1. Water valve leaks when shut off 2. Condensing unit operates too long	1. Repair 2. See section entitled "Condensing Unit Operates Too Long"
Water flow—high Head pressure—high	Excessive water flow caused by high head pressure due to: 1. High water temperature 2. Air in system 3. Too much refrigerant 4. Condenser is dirty resulting in poor heat transfer 5. Condenser too small 6. Partially closed discharge or receiver valve	 1. Install evaporative condenser 2. Purge 3. Remove some refrigerant 4. Clean or replace 5. Replace 6. Open

Type System: Any—Electric Water Valve

Water flow—normal	1. Water valve leaks when shut off 2. Condensing unit operates too long	1. Repair 2. See section entitled "Condensing Unit Operates Too Long"

Observations	Trouble	Remedy
Water flow—high	1. Water pressure too high	1. Install pressure regulating valve
	2. Water valve stuck open	2. Repair
	3. Water valve too large	3. Install smaller valve

COMPLAINT: ELECTRIC BILL TOO HIGH

First Observation: Refrigerated Space Cold Enough
Type System: Any

Observations	Trouble	Remedy
	1. Wrong electric rate	1. Get in touch with electric power company
	2. Machine operates too long	2. See section entitled "Condensing Unit Operates Too Long"

COMPLAINT: NOISE

Type System: Any

Observations	Trouble	Remedy
Compressor is noisy	1. Compressor foundation bolts loose	1. Tighten
	2. Drive belt is noisy	2.
	(a) Squeaks	(a) Clean. See if pulleys are in line. Use soap or belt dressing
	(b) Belt slaps because it is loose or worn	(b) Tighten
	3. Compressor bearings worn	3. Repair
	4. Valves noisy	4. Repair
	5. Squeaky seal	5.
	(a) Oil level too low	(a) Add oil and repair if necessary
	(b) Defective seal	(b) Replace or repair seal
	6. Compressor pumping oil	6. Remove excess oil
	7. Shipping blocks not removed	7. Remove
	8. Excessive refrigerant	8. Purge some refrigerant
	9. Air in system	9. Purge
	10. Compressor bearings worn	10. Repair
Motor is noisy	1. Motor loose on base	1. Repair or replace
	2. Bearings lack oil	2. Add oil
	3. Brushes squeak	3. Replace or fit brushes to commutator
	4. Motor bearings worn	4. Tighten bolts

COMPLAINT: ODORS

Type System: Any

Observations	Trouble	Remedy
Odor in refrigerated space	1. Cabinet dirty or spoiled food	1. Clean and wash with strong solution of baking soda and water
	2. Strong foods left uncovered	2. Instruct owner to cover all strong foods
	3. No trap in drain to sewer	3. Install trap
	4. Cabinet door kept closed when system is shut off	4. Have owner leave door open when system is turned off
	5. Refrigerant leaks	5. Locate and repair
	6. Brine leaks	6. Locate and repair
	7. Coil surface dirty	7. Wash with baking soda solution
Odors outside of cabinet	1. Refrigerant leaks	1. Locate leak and repair
	2. Brine leaks	2. Locate leak and repair
	3. Belt burned due to slipping	3. Tighten belt. Replace if necessary
	4. Motor overheated	4. Locate cause of trouble Install overload relay

PART VI

AIR CONDITIONING

Associate Engineer, Charles N. Deverall, Engineer,
Niagara Blower Company, Buffalo, N. Y.

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39. Small Comfort Cooling Systems and Units—S. Charles Segal, Chief Engineer, Kramer-Trenton Company, Trenton, N. J. 637

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35. AIR CONDITIONING

Part I. Theory of Air Conditioning

I. Properties of Mixtures of Dry Air and Steam ("Moist" Air)

1. The water vapor or steam mixed with dry air in the atmosphere is **superheated** when the relative humidity is less than 100% and saturated when the relative humidity is 100%. The **state** of a mixture of dry air and saturated steam is completely defined if the dry-bulb temperature and total pressure of the mixture are known. The tables of Chapter 10 give the properties of mixtures of dry air and saturated steam for a total pressure of 29.921 in. Hg abs and for temperatures from −160 F to 200 F.

The state of a mixture of dry air and superheated steam is not completely defined by dry-bulb temperature and total pressure. A third property of state is necessary, such as relative humidity, partial pressure of the water vapor, saturation ratio, dew point, or temperature of adiabatic saturation (also called thermodynamic wet-bulb temperature).

2. Such a property in a mixture of dry air and water vapor is the **temperature of adiabatic saturation,** sometimes called the **thermodynamic wet-bulb temperature.** If a mixture of dry air and water vapor flows over a large surface of water, the specific humidity of the mixture will be increased, and the mixture may be saturated. If the saturation process occurs at constant total pressure and with no external heat transfer, the temperature at which the mixture will be saturated depends upon the initial state of the mixture and upon the temperature at which the makeup water is supplied.

If the water is assumed to be supplied at the exit temperature of the saturated mixture, then this temperature, now defined as the temperature of adiabatic saturation, depends only upon the initial state of the mixture and is, therefore, a true property of state. The temperature of adiabatic saturation is also the temperature at which mixtures of air and water vapor may be saturated by contact with recirculated water in adiabatic processes of evaporative cooling.

The state of a mixture of dry air and water vapor is fixed and the various properties of state may be found when the total pressure of the mixture, the dry-bulb temperature, and the temperature of adiabatic saturation are known. There is little difficulty in finding the first two of these quantities, but the **experimental determination** of the temperature of adiabatic saturation is more difficult. The temperature of adiabatic saturation of a mixture of air and water vapor is commonly taken as the temperature indicated by the **wet-bulb thermometer** of a **sling psychrometer.** This procedure is quite accurate but entirely fortuitous, for the same procedure may involve large errors for other mixtures, particularly when the molecular weight of the gas is greatly different from that of the vapor.

To read accurately the temperature of adiabatic saturation of air-water vapor mixtures, the wet-bulb thermometer must be swept by an air stream at a velocity of at least 500 fpm and must be exposed to radiation from surfaces that are at a temperature close to the dry-bulb temperature of the air. A wet-bulb thermometer in relatively stagnant air (air velocity of 20 fpm), when unshielded from radiation from surfaces at the dry-bulb temperature, will read a temperature that is about 15% of the wet-bulb depression **higher** than the temperature of adiabatic saturation; if shielded from radiation, the wet-bulb thermometer in stagnant air will read a temperature that is about 15% **lower** than the temperature of adiabatic saturation.

For extremely high air velocities (5,000 fpm or more), the wet-bulb thermometer, whether shielded or unshielded from surfaces at the dry-bulb temperature, will read closely the temperature of adiabatic saturation.

The **energy balance** for the process of adiabatic saturation is

$$h+\frac{W'-W}{7{,}000}h_{f'}=h' \qquad (1)$$

where h = the enthalpy of the mixture of dry air and water vapor, Btu per lb of dry air
W = the specific humidity of the mixture, grains per lb of dry air
W' = the specific humidity for saturation at the temperature of adiabatic saturation, grains per lb of dry air
$h_{f'}$ = the specific enthalpy of the liquid (or solid) at the temperature of adiabatic saturation, Btu per lb
h' = the enthalpy of the saturated mixture at the temperature of adiabatic saturation, Btu per lb of dry air

The **saturation ratio** of the mixture, μ, also known as **percentage humidity** or **per cent saturation** is the ratio of the specific humidity of the mixture to the specific humidity for saturation at the dry-bulb temperature, or

$$\mu=\frac{W}{W_s} \qquad (2)$$

Therefore,

$$h=h'-\frac{W'-\mu W_s}{7{,}000}h_{f'}$$

Also, the enthalpy of the mixture is given by the following equation

$$h=h_a+\mu(h_s-h_a)=h_a+\mu h_{as} \qquad (3)$$

where h_a = the enthalpy of the dry air in the mixture at the dry-bulb temperature, Btu per lb of dry air
h_s = the enthalpy of the mixture of dry air and saturated steam at the dry-bulb temperature, Btu per lb of dry air

For simplicity in the use of Eq 3, it may be assumed with small error that the enthalpy of dry air in a mixture of dry air and superheated steam is equal to the enthalpy of dry air at the same total pressure and dry-bulb temperature. Actually, because the partial pressure of the dry air in a mixture is less than the total pressure, the enthalpy of the dry air in the mixture slightly exceeds the enthalpy of dry air at a pressure equal to the total pressure.

It may be shown from the preceding equations that the saturation ratio is

$$\mu=\frac{h'-h_a-\dfrac{W'h_{f'}}{7{,}000}}{h_s-h_a-\dfrac{W_sh_{f'}}{7{,}000}} \qquad (4)$$

If the pressure-volume product of superheated steam is the same as the pressure-volume product of saturated steam at the same temperature, it may be shown that the volume of a mixture of dry air and superheated steam per pound of dry air is

$$v=v_a+\mu(v_s-v_a) \qquad (5)$$

where v_a = the specific volume of dry air at the total pressure and temperature of the mixture, cu ft per lb
v_s = the volume of the mixture of dry air and saturated steam at the dry-bulb temperature, cu ft per lb of dry air

3. Examples will now be given to show how the properties of state of a mixture of dry air and superheated steam may be found from the tables of Chapter 10 covering properties of mixtures of dry air and saturated steam:

Find the saturation ratio, specific humidity, relative humidity, dew point, enthalpy and volume of a mixture of dry air and superheated steam when the dry-bulb temperature is 80 F and the wet-bulb temperature is 65 F if: (a) the total pressure is the standard atmospheric pressure of 29.921 in. Hg abs; (b) the total pressure is 29.00 in. Hg abs.

(a) From Table 1 (Chapter 10) at $t'=65$ F, find $W'=92.51$ grains per lb of dry air, $h_{f'}=33.1$ Btu per lb of steam and $h'=30.03$ Btu per lb of dry air; from the same table at $t=80$ F, find $h_a=19.23$ Btu per lb of dry air, $h_s=43.63$ Btu per lb of dry air, and $W_s=155.8$ grains per lb of dry air. The per-

centage humidity (saturation ratio) of this mixture may then be found from Eq 4 as follows:

$$a = \frac{30.03 - 19.23 - \dfrac{92.51(33.1)}{7{,}000}}{43.63 - 19.23 - \dfrac{155.8(33.1)}{7{,}000}} = 0.438$$

From Eq 2, the specific humidity of the mixture is then found to be

$$W = 0.438(155.8)$$
$$= 68.22 \text{ grains per lb of dry air}$$

The relative humidity, ϕ, is the ratio of the partial pressure of the steam in the mixture to the saturation pressure of steam at the dry-bulb temperature. It is permissible to interpolate in Table 1 (Chapter 10) on the basis of this specific humidity to find the partial pressure of the steam in this mixture to be 0.461 in. Hg. The relative humidity is

$$\phi = \frac{0.461}{1.0323} = 0.447$$

By a similar interpolation the dew point is found to be 56.6 F. This is the temperature at which condensation of the water vapor would begin if the mixture were cooled at the constant total pressure of 29.921 in. Hg.

The enthalpy of the mixture may be found from Eq 1 as

$$h = h' - \frac{W' - W}{7{,}000} h_{v'}$$
$$= 30.03 - \frac{24.29(33.1)}{7{,}000}$$
$$= 29.92 \text{ Btu per lb of dry air}$$

The volume of the mixture per lb of dry air is found from Eq 5 as

$$v = 13.602 + 0.438(14.09 - 13.602)$$
$$= 13.816 \text{ cu ft}$$

(b) When the total pressure of the mixture differs from the standard atmosphere, properties of the mixture of dry air and saturated steam at the given total pressure must be found first. At a total pressure of 29.0 in. Hg and a temperature of 80 F, properties of dry air are found as follows:

$$v_a = \frac{29.921}{29.0}(13.606) - 0.0042$$
$$= 14.034 \text{ cu ft per lb}$$
$$h_a = 19.327 - \frac{29.0}{29.921}(0.097)$$
$$= 19.233 \text{ Btu per lb}$$

Properties of a mixture of dry air and saturated steam at a pressure of 29.0 in. Hg and a temperature of 80 F follow:

$$v_s = \frac{13.606(29.921)}{29.0 - 1.0323} - 0.0042$$
$$= 14.552 \text{ cu ft per lb of dry air}$$
$$W_s = \frac{7{,}000\, v_s}{v_g} = \frac{7{,}000(14.552)}{633}$$
$$= 160.9 \text{ grains per lb of dry air}$$
$$h_s = 19.327 - \frac{(29.0 - 1.0323)0.097}{29.921} + \frac{160.9}{7{,}000}(1096.1)$$
$$= 44.43 \text{ Btu per lb of dry air}$$

In a similar manner, properties of a mixture of dry air and saturated steam at a pressure of 29.0 in. Hg and a temperature of 65 F may be shown to be $v_s = 13.942$ cu ft per lb of dry air, $W_s = 95.52$ grains per lb of dry air, and $h_s = 30.50$ Btu per lb of dry air.

The properties of the mixture of dry air and superheated steam follow:

$$\mu = \frac{30.50 - 19.233 - \dfrac{95.52(33.1)}{7{,}000}}{44.43 - 19.233 - \dfrac{160.9(33.1)}{7{,}000}} = 0.443$$

$$W' = 0.443(160.9)$$
$$= 71.28 \text{ grains per lb of dry air}$$
$$v = 14.034 + 0.443(14.552 - 14.034)$$
$$= 14.333 \text{ cu ft per lb of dry air}$$
$$h = 30.50 - \frac{(95.52 - 71.28)}{7{,}000}(33.1)$$
$$= 30.39 \text{ Btu per lb of dry air}$$
$$\phi = 0.443\left(\frac{14.552}{14.333}\right) = 0.450$$

The partial pressure of the steam in the mixture is 0.450 (1.0323) or 0.4645 in. Hg, and the dew point, or saturation temperature of steam at this pressure, is 55.1 F.

II. Processes in Air Conditioning

4. a. Regardless of the nature of the process by which the state of a mixture of air and water vapor may be changed in conditioning the air, two important relations for such processes are the **conservation of weight** and the **conservation of**

energy equations. In drawing up a weight balance for steady flow conditions, the weight of all fluids entering the equipment (whether heater, cooler, etc.) must be equal to the weight of all fluids leaving in the same unit of time. In setting up the energy balance for steady flow of the fluids through the apparatus with no appreciable changes in elevation or velocity, the enthalpy of all fluids entering plus any heat or mechanical energy supplied (minus any heat or mechanical energy removed) may be equated to the enthalpy of all fluids leaving the apparatus.

b. The **fundamental processes** required in air conditioning are heating, cooling, humidifying, dehumidifying, and mixing; in addition, there are the processes of cleaning, supplying, exhausting, and distributing the air. The refrigerating engineer is usually concerned with maintaining the required dry-bulb temperature and relative humidity within an enclosure where there is a gain of heat and a gain of water vapor. The required conditions may be maintained by direct expansion of a refrigerant in coils in the enclosure or by the supply of conditioned air. When the latter method is used, the conditioned air must be supplied at the proper rate and at a lower dry-bulb temperature than the room air in order to balance the gain of heat; the specific humidity of the conditioned air must also be lower than that maintained in the enclosure, in order to balance the gain of water vapor.

Assume that a dry-bulb temperature of t_r and a wet-bulb temperature ot t_r' must be maintained in an enclosure where there is a gain of heat of Q_h Btu per hr from transmission, solar effect, lights, people, processes, air change, etc., and a gain of water vapor of Q_v grains per hr from air change, processes, people, etc. If the conditioned air is supplied at the rate of M_a lb per hr (dry) and in a state a with a dry-bulb temperature of t_a and a specific humidity of W_a, the weight balance for water vapor is

$$M_a(W_r - W_a) = Q_v \tag{6}$$

The energy balance is

$$M_a\left(0.24 + \frac{0.44 W_a}{7{,}000}\right)(t_r - t_a) = Q_h \tag{7}$$

From these relations, the required specific humidity of the conditioned air supplied to the enclosure is

$$W_a = W_r - \frac{(1{,}680 + 0.44 W_r)(t_r - t_a)}{\dfrac{7{,}000 Q_h}{Q_v} + 0.44(t_r - t_a)} \tag{8}$$

For a given state of air in the enclosure and a given gain of heat and water vapor, fixing any one of the other three variables, (1) dry-bulb temperature of conditioned air, (2) specific humidity of conditioned air, or (3) weight rate of supply of conditioned air, will immediately fix the remaining two.

Example. A room is to be maintained at a dry-bulb temperature of 80 F and a wet-bulb temperature of 67 F ($W_r = 78.3$). There is a gain of heat of 30,000 Btu per hr and a gain of water vapor of 105,000 grains per hr. If the dry-bulb temperature of the conditioned air supplied to the room is 65 F, find the required specific humidity and rate of supply of conditioned air.

Solution. From (8) the required specific humidity is

$$W_a = 78.3 - \frac{(1714.5)(15)}{2{,}007}$$

$$= 65.5 \text{ gr per lb of dry air}$$

From (6) $$M_a = \frac{105{,}000}{12.8}$$

$$= 8{,}200 \text{ lb of dry air per hr}$$

The state of the conditioned air that must be supplied to the room may be found on the psychometric chart of Chapter 4 from the known dry-bulb temperature (65 F) and the known specific humidity (65.5). The corresponding wet-bulb temperature is 59 F; also, the volume of the supply air is 13.42 cu ft per lb of dry air, and the conditioned air must therefore be supplied at the rate of 1,830 cfm.

5. a. Some of the **processes** in air conditioning will be explained with reference to the psychrometric chart. **Adiabatic saturation** or **evaporative cooling** may be represented on the psychrometric chart without appreciable error by a process of constant wet-bulb temperature. This process is exemplified by the humidification of air in spray-type equipment if the spray water is neither heated nor cooled. For

example, if the initial dry-bulb temperature of the air is 90 F, initial wet-bulb 63 F, and if the dry-bulb is reduced to 72.5 F by spraying the air with recirculated water, the initial and final properties of the mixture are:

	Initial	*Final*
Dry-bulb temperature, F	90	72.5
Wet-bulb temperature, F	63	63
Relative humidity, %	20	60
Dew point, F	44.5	57.6
Specific humidity, gr per lb	43	71
Enthalpy, Btu per lb	28.35	28.47

b. Cooling with dehumidification may be accomplished by passing air through a spray of water or over a surface maintained at a temperature below the initial dew point of the air. The problem is to select the surface and its temperature such that dehumidification and sensible cooling will be in the right proportion to balance the moisture and sensible heat gains in the conditioned enclosure. A skeleton psychrometric chart is shown in **Fig. 1** in which the specific humidity is plotted as ordinate vs. the dry-bulb temperature as abscissa. State 1 represents the air introduced to the cooling surface, while t_s represents the temperature of that surface. If a point, S, is located on the saturation curve at the temperature t_s, the state of the air leaving the surface (one such point is represented by 2) will be on the straight line from 1 to S (with constant surface temperature).

The temperature at point 2 depends on the rate of flow of air, the area of the cooling surface, the sensible heat transfer coefficient of the surface, the surface temperature, and the dry-bulb temperature of the entering air.

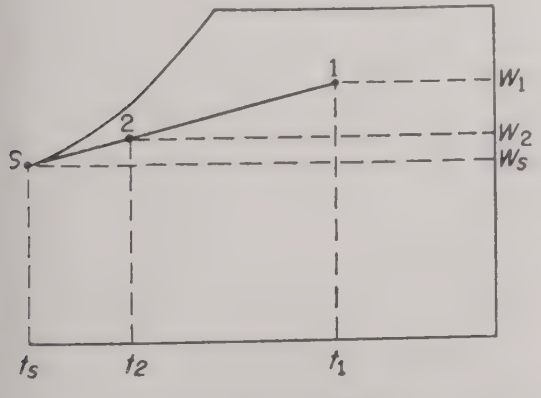

Fig. 1. Cooling and Dehumidification of Air

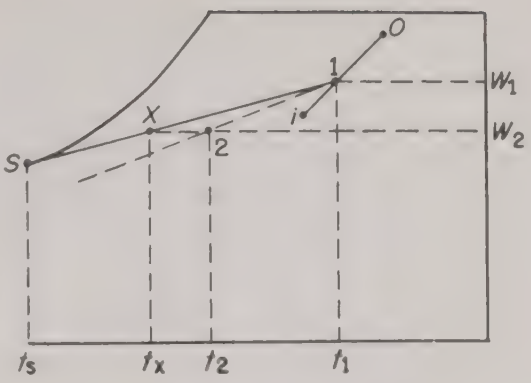

Fig. 2. Use of Reheat to Increase Dehumidification Ratio

The straight line joining the state of the air entering the air conditioner and that leaving it may fail to intersect the saturation curve as shown by the dashed line 1–2 in **Fig. 2.** This indicates that the ratio of latent to sensible heat loads is so great that at least two treatments must be used. With the state of the entering air fixed, the maximum ratio of latent to sensible cooling capacity obtainable with either spray- or surface-type equipment is fixed by the slope of the straight line drawn through this point tangent to the saturation curve. When the ratio of latent to sensible cooling loads is greater than can be handled in a single process, the air may be reheated after cooling. As shown in Fig. 2, the air may be cooled by a surface at t_s to some point X, where the desired specific humidity exists. In this cooling, there is a greater reduction in the dry-bulb temperature than is desired; if the air were admitted to the conditioned space in state X the temperature maintained would be lower than the design value. To avoid this the air must be reheated without change in specific humidity from X to 2.

When reheating is necessary, the refrigeration capacity must be greater than the total heat gain by the amount of heat added during reheating. In this case the temperature of the cooling medium should be as high as is economical in order to get the most refrigerating effect for the expenditure of a given amount of energy. Increasing the temperature of the cooling

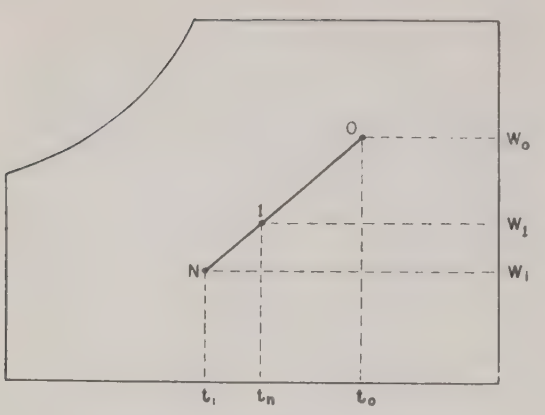

Fig. 3. Mixing of Two Quantities of Air at Different Conditions

medium, however, raises the surface temperature and moves point X to the left, thereby increasing the amount of reheating required, so there is an economical limit to the temperature of the cooling medium used.

c. The **mixing of two quantities of air** in different states may be represented on the psychrometric chart by a straight line joining the points representing the initial states, as shown in **Fig. 3.** Let point O represent the condition of outdoor air. Let point N represent the condition of indoor or recirculated air. The mixture is then represented by state 1 so located that the ratio of the distance from N to 1 to the distance from N to O is equal to the ratio of the supply of outdoor air to the total circulation. Then

$$t_1 - t_N = \frac{M_0}{M}(t_0 - t_N) \qquad (9a)$$

$$W_1 - W_N = \frac{M_0}{M}(W_0 - W_N) \qquad (9b)$$

where M_0 = the weight rate of supply of outside air
M = the weight rate of total circulation (outside plus recirculated air)

d. In **Fig. 4** is shown a summary of what happens to air in **typical summer conditioning.** Air at state 1 is cooled and dehumidified to state 2, where it enters the duct system. In the duct system heat may be transferred to the air, warming it to 3. The air is then introduced to the conditioned space in state 3. Here the air picks up heat and moisture until its state is changed to N. The length of the line from 3 to 4 is proportional to the sensible heat gain of the room (exclusive of the heat gain of outdoor air supplied mechanically and mixed with the return air). The length of the line from 4 to N is proportional to the latent heat gain of the room (exclusive of mechanical outdoor air supply). Air is removed from the room in state N and mixed with outdoor air in state O, thereby increasing its specific humidity and dry-bulb temperature to the conditions represented by state 1. The length of the line from N to 5 is proportional to the sensible heat gain from the supply of outdoor air, and that of the line from 5 to 1 is proportional to the latent heat gain from the supply of outdoor air.

e. Although most of the previous discussion has involved surface cooling, the same principles may be applied when **spray-type** equipment is used. In this case the temperature of the refrigerated spray water would replace the temperature of the cooling surface, t_s, in the analysis of the cooling process.

f. Dehumidification may also be accomplished by using the **absorbing** or **adsorbing properties** of various liquids and solids. Absorbents change physically or chemically during the absorption of water vapor; liquid absorbents include solutions of calcium chloride, lithium chloride, and ethylene glycol, while calcium chloride is also used in the solid form. Adsorbents do not change physically or chemically during the process of dehumidifying air. Solid adsorbents have a porous, sub-microscopic

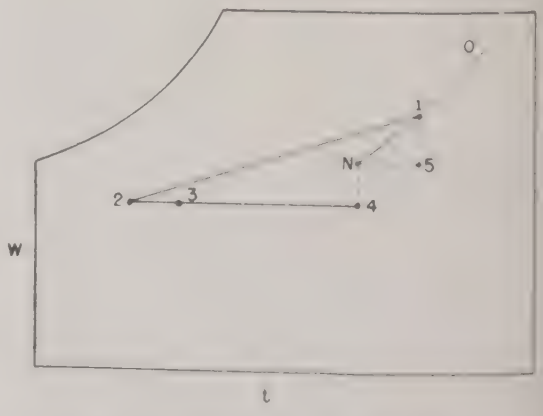

Fig. 4. Summary of Air Passages

structure with many small voids and an extremely large area of adsorbing surface; silica gel, activated alumina, activated charcoal and activated bauxites are examples of solid adsorbents.

Liquid absorbents may be reactivated by heating the solution to drive off water vapor into a stream of activating air. Solid adsorbents are commonly reactivated by contact with a stream of hot (300 F to 400 F) products of combustion or hot air.

In the process of adsorption, the dry-bulb temperature of the air increases appreciably and the wet-bulb temperature of the air increases slightly as the specific humidity of the air decreases. Typical initial and final states of a mixture of air and water vapor resulting from an adsorption process using silica gel are:

	Initial	*Final*
Dry-bulb temperature F	75	128
Wet-bulb temperature, F	70	73.5
Specific humidity, grains per lb of dry air	102.2	38
Enthalpy, Btu per lb of dry air	34.0	36.6

In the above example, the dehumidified, warm air would be cooled before delivery to an air-conditioned enclosure.

Dehumidification by processes of absorption or adsorption is used principally where low humidity must be maintained or where a close control of humidity independent of temperature control is desired.

III. Comfort and Health as Influenced by Air Properties

6. Some of the principal factors that affect human **comfort** insofar as it is influenced by **air environment** are: Dry-bulb temperature, humidity, air motion, air distribution, odors, dust, smoke, bacteria and toxic gases. Other properties of the air that may influence human comfort, but the effects of which are not so clearly established at the present time, are the ozone content and the ionic content. Large changes in air pressure also have effects upon health and comfort that are beyond the scope of this discussion. The simple processes of heating and cooling are used to change the dry-bulb temperature of the air only. Ventilation is the process of changing the air within an enclosure by supplying and distributing fresh air and exhausting used air. Air conditioning, however, is now understood to mean the simultaneous control of at least the first three of the above listed factors.

Human occupancy of an enclosure produces the following **alteration in the properties of the air:** (1) oxygen is used in metabolism; (2) carbon dioxide, a waste product of metabolism, is exhaled; (3) the dry-bulb temperature of the air is increased, because the energy liberated in metabolism is partly lost as heat; (4) the humidity of the air is increased by the evaporation of moisture from the body surface and the respiratory tract; (5) odors are given off, and (6) the number of small ions in a unit volume of the air is decreased.

Old and fallacious beliefs were that human discomfort was a consequence of the increase in the **carbon dioxide** content of the air in an enclosure or of the poisonous effect of products of organic decomposition in respired air. The modern theory is that human discomfort results when the temperature-regulating system of the body is unable to compensate comfortably for differences between the rate of energy release in **metabolism** and the rate at which **heat can be dissipated** from the body by evaporation, radiation, convection, and conduction.

7. One arbitrary index of the combined effects of dry-bulb temperature, humidity, and air motion upon the feeling of warmth or cold is **effective temperature.** The numerical value assigned to the effective temperature is the dry-bulb temperature of saturated still air producing the same feeling of warmth or cold as the air in any given state and motion. Effective temperature, then, is not a temperature at all, but an index that combines the effects of temperature, humidity, and air motion.

The engineering problem involved in providing summer comfort reduces to the maintenance of comfortable and reasonably uniform combinations of dry-bulb temperature, relative humidity (or wet-bulb temperature), and air motion. At the same time, clean outdoor air must be supplied and vitiated air removed in sufficient quantities to keep by dilution the concentration of odors, smoke, and fumes below

objectionable limits. Due to acclimatization or human adaptation to changing air conditions and also to various personal and psychological factors, it is practically impossible to set up invariable standards that will insure comfort for each human occupant in every air-conditioned enclosure. Instead, all that can be done is to prepare standards of recommended practice based upon experience in an attempt to provide comfort for most of the people.

Fig. 1, Chapter 37, is based on A.S.H.V.E. **comfort** data, with lines of constant effective temperature and zones for summer and winter where the majority of people feel comfortable. In the absence of very precise information, it seems desirable to limit the comfort zones to a range of relative humidity from 30 to 70%. This comfort chart applies only to inhabitants of the United States between 20 and 70 years of age living where central heating of the convection type is used during four to eight months of the year. Application of the **summer comfort zone** is further limited to homes, offices, etc., where the exposure of the occupants to the artificial conditions is for periods of 3 hr or more. When the exposure is relatively short, as in a bank or a store, the contrast between indoor and outdoor conditions becomes more evident, and higher indoor temperatures should be carried if the conditioning is to be done for the benefit of the customers. Factors other than the **time of occupancy** which must be used to supplement the information given on the comfort chart are the effects of high or low wall and glass temperatures (affecting radiation to or from the body surface), the state of activity of the occupant (the comfort chart is for people at rest), the amount of clothing worn, the extent of crowding of a room (effect upon radiation from body), and the state of the body at entry to the conditioned enclosure.

8. Table 1 gives suggested indoor conditions that are desirable for certain outdoor dry-bulb temperatures; the conditions apply to people engaged in sedentary or light muscular activity where the period of occupancy of the conditioned space is over 40 minutes. This schedule is not absolute and should be used with judgment. For periods of occupancy shorter than 40 minutes, the effective temperatures in the table should be slightly increased. Due to the effects of adaptation, higher effective temperatures may be more comfortable in southern climates.

Table 1. Preferred Schedule of Indoor Air Conditions Corresponding to Outdoor Summer Dry-Bulb Temperatures

(Period of occupancy over 40 min; individuals engaged in sedentary or light muscular activity)

Outdoor dry-bulb temperature, F	Indoor Air Conditions			
	Effective temperature	Dry bulb, F	Wet bulb, F	Relative humidity, %
100	75	83	66	40
	75	82	67	45
	75	81	68	51
	75	80	70	60
95	74	82	64	36
	74	81	66	44
	74	80	67	51
	74	79	68	57
	74	78	70	68
90	73	81	63	36
	73	80	64	41
	73	79	66	50
	73	78	67	56
84	72	80	61	32
	72	79	63	41
	72	78	64	46
	72	77	66	56
80	71	78	61	36
	71	77	63	45
	71	76	64	52
	71	75	66	61

Constant **air motion** is necessary in an enclosure to maintain uniformity in temperature and humidity. The air distribution may be considered satisfactory if the variation in dry-bulb temperature is 3 F or less throughout the frequented portion of a single room. The maximum air velocity in the zone between the floor and the five-foot level that can be tolerated by people at rest is about 50 fpm.

The cooling effect on the clothed human body of air at a given temperature in-

creases with the square root of its velocity; air at 65 F and a velocity of 100 fpm will produce about the same convection heat loss from the clothed human body as air at a temperature of 40 F and a velocity of 15 fpm.

The total **quantity of air** that must be supplied to a conditioned space is fixed by the heat and water vapor loads and the state of the air supplied. Only a part of this air need be introduced from the outside as fresh air, in most cases. The greater the amount of air that can be **recirculated,** the smaller is the load on the refrigerating equipment. In the absence of any other means of removing odors and tobacco smoke, there is a **minimum quantity** of clean outdoor air that must be supplied in a given case to give adequate ventilation. Standards of suggested practice are given in **Table 2.** In no case, however, should the air change be less than that required by local ordinances.

Table 2. Ventilation Standards for Summer Cooling

Application	Cfm per person	
	Preferred	Minimum
Apartment	20	15
Banking space	10	7½
Barber shop	10	7½
Beauty parlor	10	7½
Broker's board room	30	20
Cocktail bar	40	25
Department store	7½	5
Directors' room	40	30
Funeral parlor	10	5
Hospital room	20	15
Hotel room	20	15
Office, general	15	10
Office, private	20	15
Restaurant	15	12
Shop, retail	10	7½
Theater	7½	5
For general application		
Each person, not smoking	7½	5
Each person, smoking	40	25

9. Extended studies have been made and published by the A.S.H.V.E. on the **heat and moisture losses** from the human body at work and at rest under different atmospheric conditions. The human body loses heat by convection to the surrounding air and by radiation to surrounding solid surfaces whenever the temperature of the body is higher than the temperature of the air or of these surfaces. Heat is also dissipated by evaporation of moisture from the body surfaces whenever the temperature of these surfaces exceeds the dew point of the air. The heat lost by radiation and convection increases the sensible cooling load, and the heat lost by evaporation of moisture increases the latent cooling load. The hourly rate of heat loss and the distribution of the heat loss at a dry-bulb temperature of the surrounding air of 79 F are given in Fig. 3 of Chapter 37 for several different states of activity.

The total hourly loss of heat from the human body at rest is substantially constant for any effective temperature within the limits of the comfort zones. The distribution of the total loss of heat between sensible heat and latent heat depends chiefly upon the dry-bulb temperature of the air and the state of activity. Within the limits of the comfort zones, this distribution is not greatly affected by the humidity of the air, although humidity becomes very important at higher dry-bulb temperatures. For a constant state of activity, the fraction of the total loss that is in the form of sensible heat decreases as the dry-bulb temperature of the air increases. The effect of dry-bulb temperature upon heat and moisture losses from the human body is shown in **Table 3.**

10. a. The **quality** of air, from the standpoint of ventilation, is affected by the amount and nature of the **dust** carried. Dust is sometimes classified as solid particles in the air between 1 and 150 microns in diameter (one micron is one-thousandth of a millimeter). Classified in this manner, dusts settle in accordance with the laws of gravity. Some industrial operations produce dusts that, due either to their abrasive or toxic actions, are injurious to health when inhaled.

The **dust content** of the air may be measured by impingement, scrubbing, electrostatic precipitation, or settling. In the Owens dust counter, a measured volume of air is drawn into the apparatus by a piston, moistened and caused to impinge upon a

Table 3. Heat Losses from Human Body

A. As affected by state of activity; indoor dry-bulb temperature of 80 F (ACRMA Application Engineering Standards, 1947)

Activity	Total loss, Btu per hr	Sensible heat loss, Btu per hr	Equivalent latent heat loss, Btu per hr	Moisture evaporated, grains per hr
Seated at rest	330	180	150	1,000
Seated, very light work	400	195	205	1,370
Moderately active, standing, light work	450	200	250	1,670
Light factory work	750	220	530	3,560
Moderately heavy factory work	1,000	300	700	4,660
Heavy work	1,450	465	985	6,560

B. As affected by room dry-bulb temperature

Indoor dry-bulb temp F	Per person seated at very light work		Per person at light factory work	
	Sensible, Btu per hr	Equivalent latent, Btu per hr	Sensible, Btu per hr	Equivalent latent, Btu per hr
80	195	205	220	530
75	240	160	295	455
70	275	125	365	385

Table 4. Physiological Effects of Gases and Vapors

(Concentration in parts per 10,000 parts of air by volume)

Substance	Rapidly fatal	Maximum allowable for prolonged exposure
Ammonia	50	1
Aniline	—	0.05
Arsine	2.5	—
Benzene	200	1
Carbon dioxide	1000	50
Carbon disulfide	20	0.2
Carbon monoxide	40	1
Chlorine	10	0.01
Gasoline	240	5
Hydrochloric acid	10	0.1
Hydrocyanic acid	30	0.2
Hydrofluoric acid	2	0.03
Hydrogen sulfide	6	0.2
Lead	—	5
Nitrobenzene	—	0.01
Phosgene	0.5	0.01
Phosphine	10	—
Sulfur dioxide	4	0.1
Toluene	200	2
Turpentine	—	1
Xylene	200	2

glass disc. The dust particles collected on the disc are counted under a microscope. One arbitrary rule concerning the quality of the air is that the number of dust particles present in a well-ventilated room should not exceed 25,000 per cu ft.

b. Fumes are usually classified as particles between 0.2 and 1 micron in diameter and may result from distillation or chemical reaction. The physiological effects of certain gases and vapors appear in **Table 4.**

c. Smoke may be classified as particles less than 0.3 micron in diameter, usually resulting from incomplete combustion. Smokes do not settle from the air and may be so fine as to penetrate, in part, most filters. To ventilate a room occupied by tobacco smokers may require a supply of as much as 20 cfm of air per smoker more than is supplied when the occupants are not smoking.

d. The usual procedure in determining the **bacteria** content of the air is to expose culture plates to the air for 2 min, then incubate at 98 F for from 24 to 48 hr and count the colonies of bacteria which have developed under a microscope.

11. Distinct from the effect of the properties of air upon human comfort are the environmental influences upon **health.** Such influences are difficult to measure due to inability to establish a concise and accurate definition of health or disease. The general relationship between extreme conditions of weather and human health is well known. High environmental temperatures and humidities produce a rise in body temperature owing to a decrease in the rate of heat loss by radiation, convection, and evaporation. When the temperature of the environment attains that of the body, the loss of heat by radiation and convection stops.

However, the effectiveness of air conditioning in the treatment of individuals suffering from symptoms of **hay fever** and **pollen asthma** has been demonstrated. Also of interest but of no statistical value, because of the limited number of subjects, are the experiments conducted by Kerr and Lagen, which seem to indicate that the **common cold** cannot be transmitted from one person to another in a properly air conditioned room..

A high relative humidity (about 65%) has been found to be beneficial to the health and growth of premature infants up to the time that the infants reach a weight of about 5 lb.

Airborne contagion may be reduced by dust elimination and droplet-nuclei disinfection. Solutions of triethylene glycol and propylene glycol are germicidal for most airborne bacteria, particularly in dust-free air with relative humidity between 40 and 60%. Additional data, however, are needed for conclusive evidence of the effects of glycol solutions and ultraviolet radiation.

Part II. Air Purification*

Purification of air is one of the major functions of air conditioning. Just as air temperature may be controlled through the use of heating and cooling coils in the system, so the content of air as regards dust, fumes, pollens, odors and other contaminants may be controlled through the use of the various air cleaning devices which are available. These devices are installed in the duct system, preferably ahead of the coils or at the intakes of unit air conditioners. Sufficient area is provided to assure the air velocity at which the particular device operates with the greatest efficiency. Provision is made for the removal of the filtering elements for cleaning or reactivation. In many installations, permanent gages are installed which indicate the resistance through the cleaning device. When the resistance reaches a prescribed point, the elements are taken out and cleaned or thrown away according to the type of equipment used.

In practically all air conditioning systems, a certain quantity of air is discharged from the system and replaced with **fresh air** or new air. The balance of the air is recirculated through the conditioner, is recooled, reheated, humidified or dehumidified according to what function the conditioner may be performing at the time.

In some systems where the contamination of air is very slight, the recirculated air is not cleaned but is mixed with the fresh air which is subjected to a cleaning treatment before it enters the system. In other cases, where contamination is heavy, both fresh and recirculated air are treated. In certain systems, even the air wasted must be cleaned before it may be discharged to the atmosphere because it must always be remembered that what may be "waste air" to one system may become the "fresh air," in part or in whole, of a system installed next door.

Fresh air, so-called, is air taken into the system from out-of-doors to make up for the air discharged from the system. Its degree of freshness may vary widely. It may contain very little contamination or it may well be less "fresh" than the air in the conditioned area and so may require a great deal of cleaning or purification. Air recirculated in the conditioned space may

* This portion of Chapter 35 was prepared by C. N. Deverall.

pick up anything from a little fine dust to heavy, airborne waste from machine processes, smoke, fumes, organic vapors and odors. The amount of purification required in any given system must, therefore, be based on the job the system has to do, the conditions under which it must operate and the content of the air available for makeup.

The selection of **equipment** for the system becomes the problem of the designing engineer, who must evaluate all of the points mentioned and decide what type of equipment or what combination of types of equipment will best suit his needs. An expensive system which might be perfectly justified in one case might be entirely out of line in another, from the standpoint of first cost, operation and maintenance.

Equipment Types

There are a number of methods available for the cleaning and purification of air. All of them have certain merits; most of them have certain deficiencies. Some of the methods will remove heavy particles but will pass the fine particles. Some will efficiently collect the fine particles but would soon become clogged if subjected to the heavy particles. Some will remove fumes, organic vapors or odors but would not be suitable for the removal of solids. Very frequently a combination of two or more of the above-mentioned types will be the answer.

The various types of equipment available will fall into the following general classifications:

Centrifugal collectors
Air washers
Dry-type filters
Viscous filters for manual operation
Viscous filters for automatic operation
Electrostatic precipitators
Adsorption equipment

The **centrifugal collector** will be effective in the removal of the heavy, airborne contaminants found in industrial plants, such as waste from woodworking machines, exhaust from standing, grinding and buffing operations, fly ash, etc., all of which have particles of considerable magnitude and specific weight. These heavy particles are "whirled out" of the air stream as it flows through a centrifugal collector operating on the centrifuge principle. After the air has passed through this device, it is either wasted to the atmosphere or taken through a secondary process composed of equipment of a different type which will remove the finer particles, rendering the air fit for reuse.

Air washers fall into two general types. One is the **spray washer** composed of a watertight casing, banks of spray nozzles which break the water up into very fine droplets and eliminators for the arresting of entrained moisture which would otherwise go through into the air stream. The other type is the **surface** or **scrubber type** which presents a very considerable surface of scrubber plates to the air stream, which surface is sprayed or washed down constantly with spray nozzles from above. Impurities are removed in the spray unit through the action of the falling droplets from the sprays which contact the contaminants and carry them down into the pan. In the scrubber unit, the air passing through the unit is impinged upon the wetted surface, the contaminants captured and carried down through the washing action of the sprays above.

The air-cleaning function in air washers is usually combined with the treatment of air to control its temperature and moisture content. As a matter of fact, the cleaning action is rather secondary to these other two important functions. The control of the spray temperature permits delivery of air at a given dew point which, in turn, controls the moisture content of the air in the system. It does, however, do a pretty good job of cleaning where certain types of impurities are present. In the case of water-soluble, organic vapors a pretty fair job of cleaning can be done.

The scrubber principle is used in certain types of waste-air systems, such as paint-spray booths, etc., where the air is drawn through vertical eliminators which are washed down with scrubber nozzles. In many localities, paint-spray waste would not be permitted to be discharged to the atmosphere without first going through some kind of a cleaning system.

Dry-type filters, so-called, are either of the throwaway type or the semi-permanent type. The throwaway type filter, as its name indicates, is made up of a number of elements which, after they have picked up their quota of airborne impurities, are discarded and replaced with new ones. They are usually made up of cardboard frames filled with filter media of paper, gauze, metallic or glass wool, animal hair, etc. They are necessarily made of such materials that their cost will be low. The semi-permanent types are made up in the same general form but are provided with permanent metal frames and supports for the filter media. When they have performed their function, they are withdrawn from the frame; the filter media is removed, thrown away and replaced by the user. With this type of filter, the permanent sections will be built of durable materials but the filter media must, of course, be of low cost.

Viscous-type filters of the manually-operated type are usually made up in much the same form as the dry-type filter as regards to size, etc. In this type of filter, however, the frame and filter media are of a permanent type and are considerably more expensive. The filter media is usually made of durable wools or fibres, screen wire, metal punchings, etc., contained in a metal frame and treated with a viscous oil which captures and holds the particles picked up through impingement in their passage through the filter. When the filter becomes loaded to the point where further resistance would unduly retard air flow, it is taken out of the bank and cleaned in an oil bath, allowed to drain and replaced in service.

Both the dry-type and manually-operated viscous filters are usually mounted in permanent slides placed in the intake air ducts. Access must be provided for the removal of the filter elements either by pulling them through slides in the wall of the ducts or by entering the duct through an access door and removing them from the face of the frame.

Automatic viscous filters are usually made up of frames somewhat similar to those described for manually-operated viscous filters or of a series of perforated metal plates through which the air must pass. The frames or plates are mounted on an endless chain and are driven in vertical motion through the air stream, then through a cleaning bath or trough in the bottom of the unit. The plates or frames pass through this cleaning bath, are cleansed of their impurities, recoated with the adhesive and drained before they re-enter the air stream. This type of filter is very efficient, is very widely used and is, of course, correspondingly expensive from a first cost standpoint. Where a large amount of air is handled the year around, the comparatively low service cost of this type of filter should be considered.

The **electrostatic precipitator** operates on the principle that airborne particles subjected to the influence of a high-tension electric field will become charged or ionized and will be precipitated when they contact an electrode of opposite polarity. The units are made up of a series of plates coated with a viscous material which captures and holds the precipitated particles.

In one type of unit which is manually operated a periodic shutdown is necessary for the cleaning of the device. The plates are washed down with water sprays and then recoated with the viscous oil. In another type plates are mounted on an endless chain very similar to the arrangement described for the automatic viscous filter, and the plates are carried successively through the air stream, then through a cleaning bath in which the dust particles are removed and the surface recoated for another trip through the air stream.

The electrostatic precipitator has found wide use in the removal of heavier, airborne particles such as fly ash, etc. It is frequently used in conjunction with dry or viscous filters as the first stage, taking out the heavier particles, after which the air passes through the dry or viscous filters for final cleaning.

Adsorption equipment, as the name implies, is used for the adsorption of odors, organic gases and vapors. The air is usually taken through a first stage filter before passing on to this device. There are a number of adsorbents, of which activated charcoal or activated carbon is a good

example. This material has a very high adsorption value, as it will pick up 40% to 50% of its own weight in many of the gases ordinarily found in the air we breathe. By properly proportioning the adsorptive area, the system may be made to operate for long periods of time without servicing. When they become saturated the filter beds are reactivated by the application of heat at about 1,000 F.

Activated carbon is placed in perforated containers or canisters, so arranged that they may be placed in the air stream and easily removed for reactivation. The time cycle of reactivation will depend upon the retentivity of carbon for the particular gases or vapors picked up in this operation. This type of equipment has been found particularly desirable in the removal of odors in rooms used for public assembly, such as restaurants, theatres, meeting halls, etc. Much commendable work has been done in the design of the equipment for ready servicing. Application Data Section 42, published in the September, 1948, issue of *Refrigerating Engineering*, covers this subject in detail.

Design Data

No attempt is made here to present design data. The manufacturers of the various types of equipment have done a great deal of research, each on his own particular equipment. These manufacturers will be glad to supply very complete design data upon request and, as improvements are constantly being made in the equipment, it is well to keep in touch with these manufacturers so that the very latest information may be available.

A certain amount of similarity exists in the respective types and considerable physical standardization has been done in the way of element sizes and general installation arrangements. Filters 16 in. × 20 in., 20 in. × 20 in., 16 in. × 25 in. face area will be found in many of the manufacturers' lines and, indeed, a great many of the filters are interchangeable with other makes of filters for a given frame arrangement. The thickness of these filter elements varies from 1 in. to 4 in. or 5 in. Two 1-in. elements are frequently used, the intake element only being discarded at each cleaning. The second element, being fairly clean, is brought forward to take the place of the discarded section and a new one inserted back of it.

Air Velocity

Air velocities through air cleaning equipment will vary to a considerable degree. In centrifugal equipment, velocities will range from 50 to 2,000 fpm, according to the specific weight and nature of the particles handled. Air washer velocities will average around 500 fpm. Velocities through filters of the dry type seem to average around 300 fpm, while in the viscous filters they will vary from 300 to 500 fpm. Air resistance data are given in terms of clean filters and will vary in the different makes from .06 to .2, the higher resistance as a rule existing in the filters in which higher air velocities are used.

The manufacturer should be consulted as to the arrangement of the equipment and should be fully acquainted with the specified design data as to allowable air resistance, cfm, etc. Local safety codes and underwriters' requirements should be carefully studied before the choice of any filter is made. Good practice would indicate filter media of a fire-resisting type and it will be found that many codes specifically require this feature.

In general, the choice of the equipment either in individual types or in combination should be made upon the basis of the work to be done, the efficiency desired and the conditions under which the equipment must operate.

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36. FANS AND DUCTS

I. Fans

FANS are used in air conditioning and ventilation applications to produce air flow at free delivery or through duct systems. The type and style of fan to be used depends upon the noise limitations, volume of air required, and the static pressure of the given application.

1. The usual **classifications** are the propeller type and centrifugal type. The **propeller fan** is used where air movement against small static pressures is required, and usually affords quieter operation at low pressures than the **centrifugal fan.** This is usually in cases where little or no ductwork is used. The centrifugal fan will operate efficiently against the higher static pressures encountered in duct systems, and with less noise than the propeller fan. Central plant air conditioning systems and ventilating systems using ducts for air distribution generally use centrifugal fans ("blowers") because the static pressures encountered are usually too high to be handled efficiently by a propeller fan.

In the propeller fan, air is discharged in a direction parallel with the shaft. In the case of the centrifugal fan, the blades may be straight, as in a paddle wheel, or curved forward or backward in the direction of air flow. Air is discharged from centrifugal fans in the plane of rotation. Many designs of both types are on the market.

Centrifugal fans with **forward curved blades** differ in performance from those with **backward curved blades.** To yield a specified air output the former have to be run at slower speed, or they may be smaller in diameter. If similar fans are run at the same speed, the output of the one with forward curved blades will be considerably greater than that of the fan with backward curved blades.

There is also a difference in typical characteristic performance as is shown in **Figs. 1, 2** and **3.**

Sometimes it is desirable to adopt suitable **fan speeds** in relation to output, which may work out to better advantage with one style wheel than with the other.

A third style of centrifugal fan utilizes wide **radial blades.** Such fans are often used in very large sizes.

Total pressure of a fan is the difference between the total pressure head (sum of static and velocity pressure heads) of the air leaving, and the total pressure head of air entering the fan. Total pressure is thus a measure of the useful work done on the air.

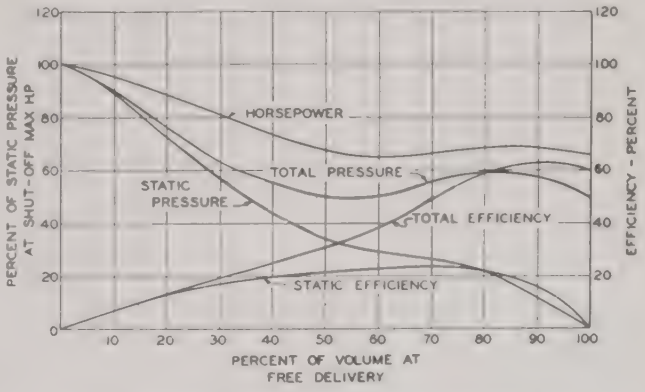

Fig. 1. Typical Characteristics of Propeller Fans

Static pressure is the pressure of air at rest, or it is the pressure exerted equally in all directions independently of any pressure exerted by the air because of its motion.

Static pressure of a fan may be defined arbitrarily as the total pressure of the fan minus the velocity pressure corresponding to the air velocity at the fan outlet. This is most commonly accepted as a definition of static pressure at the present time, but, though useful, it is not a true definition. Since it is an arbitrary statement, it may be subject to modification, and in working with fan data care should be exercised to

ascertain the exact definition used in connection with such data.

Air horsepower is a measure of the work required to move the actual volume of air against a pressure equal to the total fan pressure, thus:

$$\text{Air horsepower} = \frac{\text{cfm} \times (\text{total pressure in in. } H_2O)}{6{,}356}$$

Standard air is air weighing 0.07488 lb per cu ft, which corresponds to air having a barometric pressure of 29.921 in. Hg, a dry-bulb temperature of 68F and 50% relative humidity. This definition is universally accepted and gives a standard basis for all air flow calculations. Laws which will be given later may be used for corrections when air deviates from standard conditions.

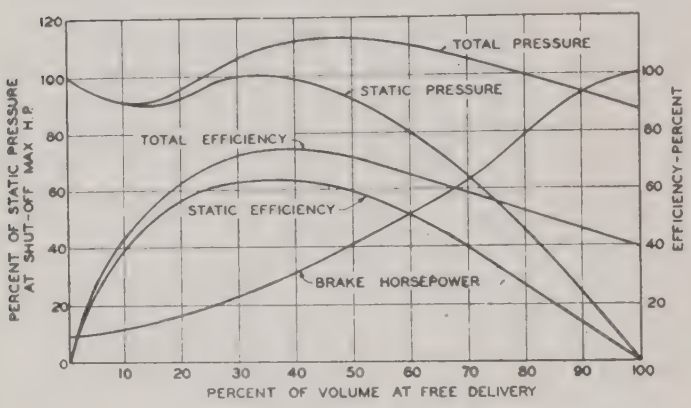

Fig. 2 (Left). Characteristic Curves for Forward Curved Centrifugal Fans

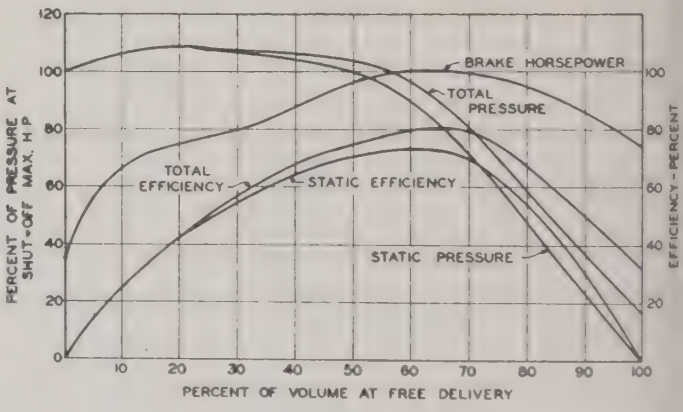

Fig. 3 (Right). Characteristic Curves for Backward Curved Centrifugal Fans

Total efficiency, also called mechanical efficiency, is the ratio of the power required to move the actual volume of air against the total pressure of the fan to the shaft power input (brake horsepower):

$$\text{Total efficiency} = \frac{\text{cfm} \times (\text{total pressure in in. } H_2O)}{6{,}356 \times (\text{hp input to shaft})}$$

Static efficiency is the ratio of the power required to move the actual volume of air against the static pressure of the fan to the shaft input:

$$\text{Static efficiency} = \frac{\text{cfm} \times (\text{static pressure in in. } H_2O)}{6{,}356 \times (\text{hp input to shaft})}$$

Brake horsepower is the input to the fan shaft, but does not always include bearing or belt losses. This point must be watched carefully when selecting the driving motor.

2. Fan characteristic curves, representative of propeller fans in general, are shown in Fig. 1. The curves show that the capacity of the fan drops quickly with an increase of static pressure, and that the horsepower increases because of the reduced efficiency with increased pressure. Hence, propeller fans are limited, for the most part, to the movement of relatively large volumes of air at comparatively low static pressures.

Characteristic curves for a **forward curved** centrifugal fan are shown in Fig. 2. This type of fan is widely used, has a large capacity for the space occupied, and is quiet in operation. The point of maximum efficiency is very close to the point of maximum static pressure. Forward curved fans should be operated close to this point. At lower pressures operation may be accompanied with objectionable noise. Beyond the point of maximum efficiency (to the left of the peak of the curve of static efficiency), forward curved fans may be unstable in operation and subject to wide variations in air flow with comparatively small changes in static pressure. This may be accompanied by excessive noise caused by fluttering, or sudden changes in air flow. The power curve rises continuously from a minimum at shut-off to a maximum at free delivery, showing that from the standpoint of the fan alone the driving motor might be subject to overloading or underloading in case it were subject to fluctuating static pressure.

Fig. 3 shows characteristic curves for a **backward curved** centrifugal fan. Its usual high-speed operation often makes desirable direct-connected motor drive, even with large size fans, and its non-overloading power curve has advantages. The steep pressure curves to the right of the point of maximum efficiency give a relatively flat regulation curve, so that considerable changes in static pressure do not greatly affect the air flow. The maximum power requirement generally occurs at the point of maximum efficiency. If a motor is selected for this point, the possibility of its being overloaded is eliminated, even though the static pressure of the system is considerably smaller than expected.

There are other variations in centrifugal fans between the full forward and full backward curve types, but they all have characteristics varying between those of the above mentioned fans.

3. The actual operation of a fan in a given system depends upon the fan characteristics and the **system characteristics. Fig. 4** shows how a given fan will operate with a typical duct system. The fan will always operate at the point of intersection of the system characteristic with the fan characteristic. When a different operating condition is required, it is necessary to change either the fan speed or the system characteristic. Referring to this figure, suppose a fan is operating at point C, the intersection of system II characteristic with the 265 rpm fan characteristic, where it is delivering 21,400 cfm. If it is desired to reduce the air flow to 19,000 cfm it can be accomplished by changing

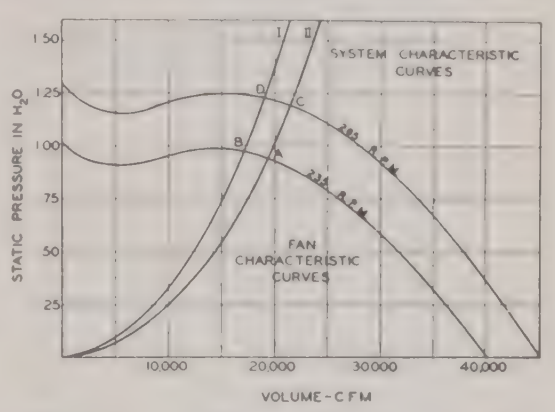

Fig. 4. System Characteristic Curves and Fan Characteristic Curves for a Forward Curved Centrifugal Fan

the fan speed to 235 rpm, in which case the point of operation will be point A. Another method is to throttle the duct system so that the point of operation is D. When the desired result is produced by changing the fan speed, the operating point moves along the system characteristic and remains at the same high efficiency point on the fan characteristic. When the change is made by throttling, the operating point moves along the fan characteristic curve toward the unstable region, with increased static pressure and some decrease in required motor horsepower. However, this decrease in motor horsepower is less than if the change is effected by reducing the fan speed, because the efficiency is decreased.

Changes in Operating Characteristics

4. The following principles may be used to determine the operating characteristics of fans, either centrifugal or propeller, when the conditions of operation change. These laws, although approximate, are sufficiently accurate for most practical purposes.

a. With a given fan in a given duct system and constant air density:
   1. The capacity varies directly as the fan speed.
   2. The pressure (either static or total) varies directly as the square of the fan speed.
   3. The fan horsepower varies directly as the cube of the fan speed.

b. With a constant weight of air circulated by the fan:
   1. The speed, capacity, and pressure vary inversely as the density.
   2. The horsepower varies inversely as the square of the density.

c. To maintain a constant pressure at the fan, the speed, capacity, and horsepower must vary inversely as the square root of the density.

d. With speed constant and capacity constant, the pressure and horsepower vary directly as the density of the air.

5. The first of the above rules is useful in determining the effect of a change of speed on the operating characteristics of a fan. The others are useful in selecting fans and determining their performance when the air density differs appreciably from standard because of change of temperature or elevation above sea level. The following **examples** are illustrative of their use.

I. A certain fan is operating in a given system under the following conditions, with 68F air.

Capacity = 19,000 cfm
Static pressure = 0.93 in. H_2O
Speed = 235 rpm
Bhp = 4.9

What will be the conditions of operation if the speed of the fan is increased to 265 rpm without making any changes in the distribution system? From rule a:

$$\text{Capacity} = 19{,}000 \times \left(\frac{265}{235}\right) = 21{,}400 \text{ cfm}$$

$$\text{Static pressure} = 0.93 \times \left(\frac{265}{235}\right)^2 = 1.18 \text{ in. } H_2O$$

$$\text{Horsepower} = 4.9 \times \left(\frac{265}{235}\right)^3 = 7.0$$

These laws may be applied to the calculation of new fan characteristic curves for different speeds, if the characteristic curve for one speed is known. In the case of Fig. 4, suppose that it is required to determine the location of the static pressure curve for 265 rpm, when the 235 rpm curve is known. The expressions for finding the point D corresponding to the point B are as follows:

$$\text{Static pressure} = 0.96 \times \left(\frac{265}{235}\right)^2 = 1.22$$

$$\text{Cfm} = 16{,}800 \times \left(\frac{265}{235}\right) = 19{,}000$$

Therefore D is definitely located by the above values of static pressure and cfm. In like manner, other points may be located and the complete curve plotted.

II. A certain fan application has the following requirements:

Required cfm = 10,000 at 68F
Static pressure = 0.59 in. H_2O
Mean barometric pressure = 25.4 in. Hg.

The problem is to select the fan and determine its speed and horsepower:

a. The air density at 25.4 in. Hg

$$= 0.07488 \times \left(\frac{25.4}{29.92}\right) = 0.06357$$

b. Use rule b to work backward to determine the cfm and static pressure, with standard air, to be used in selecting the fan from standard rating tables.
   1. The volume of standard air equivalent to the same mass of required air is

$$10{,}000 \times \frac{0.06357}{0.07488} = 8{,}500 \text{ cfm}$$

   2. The static pressure varies inversely as the density.

$$0.59 \times \frac{0.06357}{0.07488} = 0.50$$

c. The following data are obtained from a standard rating table:
   Capacity = 8,500 cfm
   Static pressure = 0.50 in. H_2O
   Speed = 207 rpm
   Brake horsepower = 1.07 hp

d. Use rule b to check the fan performance with 68F air at a barometric pressure of 25.4 in. Hg.

$$\text{Speed} = 207 \times \frac{0.07488}{0.06357} = 244 \text{ rpm}$$

$$\text{Capacity} = 8{,}500 \times \frac{0.07488}{0.06357} = 10{,}000 \text{ cfm}$$

$$\text{Static pressure} = 0.50 \times \frac{0.07488}{0.06357} = 0.59 \text{ in. } H_2O$$

$$\text{Bhp} = 1.07 \times \left(\frac{0.07488}{0.06357}\right)^3 = 1.48 \text{ hp}$$

III. Take the case of a certain fan delivering 3,000 cfm at 230 rpm, with 4.6 hp, at an air temperature of 68F (density=0.07488) and a static pressure of 1.00 in. H_2O. Determine the speed, capacity, and horsepower requirements if the pressure is maintained constant and the air temperature rises to 220 F (density 0.0584). From rule c:

$$\text{Speed} = 230 \sqrt{\frac{.07488}{.0584}} = 261 \text{ rpm}$$

$$\text{Capacity} = 3{,}000 \sqrt{\frac{.07488}{.0584}} = 3400 \text{ cfm}$$

$$\text{Bhp} = 4.6 \sqrt{\frac{.07488}{.0584}} = 5.21 \text{ hp}$$

IV. An air conditioning system located in a city where the mean barometric pressure is 25.4 in. Hg (air density at 68F =0.06357 lb per cu ft) requires that 10,000 cfm of 68F air be circulated. The static pressure under actual operating conditions is calculated as 0.59 in. H_2O. It is required to select a fan, determine its speed and horsepower.

a. Use rule d to work backward to determine the cfm and static pressure, with standard air, to be used in selecting the fan from standard rating tables.

The capacity will remain constant at 10,000 cfm.

The static pressure will vary directly as the density.

$$\text{Static pressure with standard air} = \frac{0.07488}{0.06357} \times 0.59 = 0.695$$

b. The following data are obtained from a standard rating table.

Capacity = 10,000 cfm

Static pressure = 0.695 in. H_2O

Speed = 244 rpm

Brake horsepower = 1.74 hp

c. This fan will have the following characteristics, with 68 F air at 25.4 in. barometric pressure (by rule d):

Speed = 244 rpm

Capacity = 10,000 cfm

$$\text{Static pressure} = .695 \left(\frac{0.06357}{0.07488}\right) = 0.59 \text{ in. } H_2O$$

$$\text{Bhp} = 1.74 \left(\frac{0.06357}{0.07488}\right) = 1.48 \text{ hp}$$

Fan Selection

6. In order to select the proper fan for a given installation it is necessary to know the following:

a. Volume of air required per minute at a definite density
b. Total resistance pressure of the system when the fan is delivering air of this same density
c. Noise limitations
d. Space available
e. Type of load, variable or otherwise
f. Type of motive power available

Fan manufacturers simplify fan selection by supplying tables or curves of fan characteristics, which usually give the following information for each size of fan:

a. Volume of air handled
b. Outlet velocity
c. Fan speed in rpm
d. Tip or peripheral speed of fan rotor
e. Brake horsepower
f. Static pressure

This information is given for air under standard conditions, i.e., 68 F, 50% relative humidity (density=0.07488 lb per cu ft).

Fans should be operated as closely as possible to the point of maximum efficiency. This also usually corresponds to the point of quietest operation.

Table 1 gives representative values of outlet velocities and tip speeds for forward and backward curved centrifugal fans. Tables of this sort are an aid in selecting fans which will operate near their points of maximum efficiency and with a reasonable degree of quietness. Although the numbers

Table 1. Air Velocities and Tip Speeds Used with Multiblade Fans

Press (static) in. H_2O	Forward curved fans		Backward curved fans	
	Outlet vel, fpm	Tip speed, fpm	Outlet vel, fpm	Tip speed, fpm
0.250	1,000–1,100	1,500–1,700	1,000–1,100	3,300–3,700
0.375	1,000–1,100	1,700–1,900	1,000–1,100	3,700–4,100
0.500	1,000–1,200	1,900–2,100	1,100–1,300	4,100–4,500
0.625	1,100–1,300	2,100–2,400	1,100–1,300	4,500–4,700
0.750	1,200–1,400	2,400–2,700	1,100–1,300	4,700–4,900
0.875	1,300–1,600	2,700–2,900	1,100–1,200	4,700–4,900
1.00	1,400–1,800	2,900–3,100	1,000–1,100	4,900–5,100
1.25	1,700–1,900	3,100–3,400	1,000–1,100	5,000–5,200
1.50	1,800–2,100	3,400–3,800	1,000–1,100	5,200–5,600
1.75	1,900–2,300	3,800–4,200	900–1,000	5,600–6,000
2.00	2,200–2,600	4,200–4,500		
2.25	2,400–2,800	4,500–4,700		
2.50	2,600–3,000	4,700–4,900		

here are representative of the general run of fans of these types, values recommended by the manufacturer of the particular fan under consideration should be used.

The following precautions should be observed when applying fans to air conditioning systems. The fan should be selected to deliver the required cfm at a slightly greater static pressure than the calculated value of the system, in order to allow for small errors in calculation and to assure the delivery of the required amount of air.

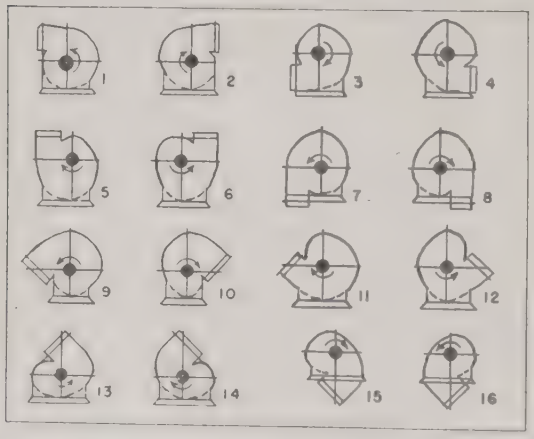

Fig. 5. Standard Designation of Fans

1. Counter-Clockwise Top Horizontal. 2. Clockwise Top Horizontal. 3. Clockwise Bottom Horizontal. 4. Counter-Clockwise Bottom Horizontal.

5. Clockwise Up Blast. 6. Counter-Clockwise Up Blast. 7. Counter-Clockwise Down Blast. 8. Clockwise Down Blast.

9. Counter-Clockwise Top Angular Down. 10. Clockwise Top Angular Down. 11. Clockwise Bottom Angular Up. 12. Counter-Clockwise Bottom Angular Up.

13. Counter-Clockwise Top Angular Up. 14. Clockwise Top Angular Up. 15. Clockwise Bottom Angular Down. 16. Counter-Clockwise Bottom Angular Down.

Also, the motor should have a nameplate rating somewhat in excess of the brake horsepower as obtained from the fan catalog data, for the following reasons:

a. To provide for losses in the bearings of the fan, and belt drives, when used.
b. To provide a margin for increasing the fan speed in case it is found, after installation, that the fan will not deliver the required output.
c. To provide a margin of reserve in the case of a forward curved fan to prevent overloading the motor in case anything happens to decrease the resistance pressure of the system, or in case the calculations were too high.

The mechanical arrangement of a fan is important, and is influenced by the space available and the arrangement of the system. The National Association of Fan Manufacturers[1] has adopted a method of designating fans as follows (Fig. 5): Facing the driving side of the fan, blower, or blast wheel, if the proper direction of rotation is counter-clockwise, the designation is **counter-clockwise.** (The driving side of a single inlet fan is considered to be the side opposite the inlet, regardless of the actual location of the drive.) This method of designation applies to all centrifugal fans, single or double width, and single or double inlet.

The discharge of a fan will be determined by the direction of the line of air discharge and its relation to the fan shaft, as:

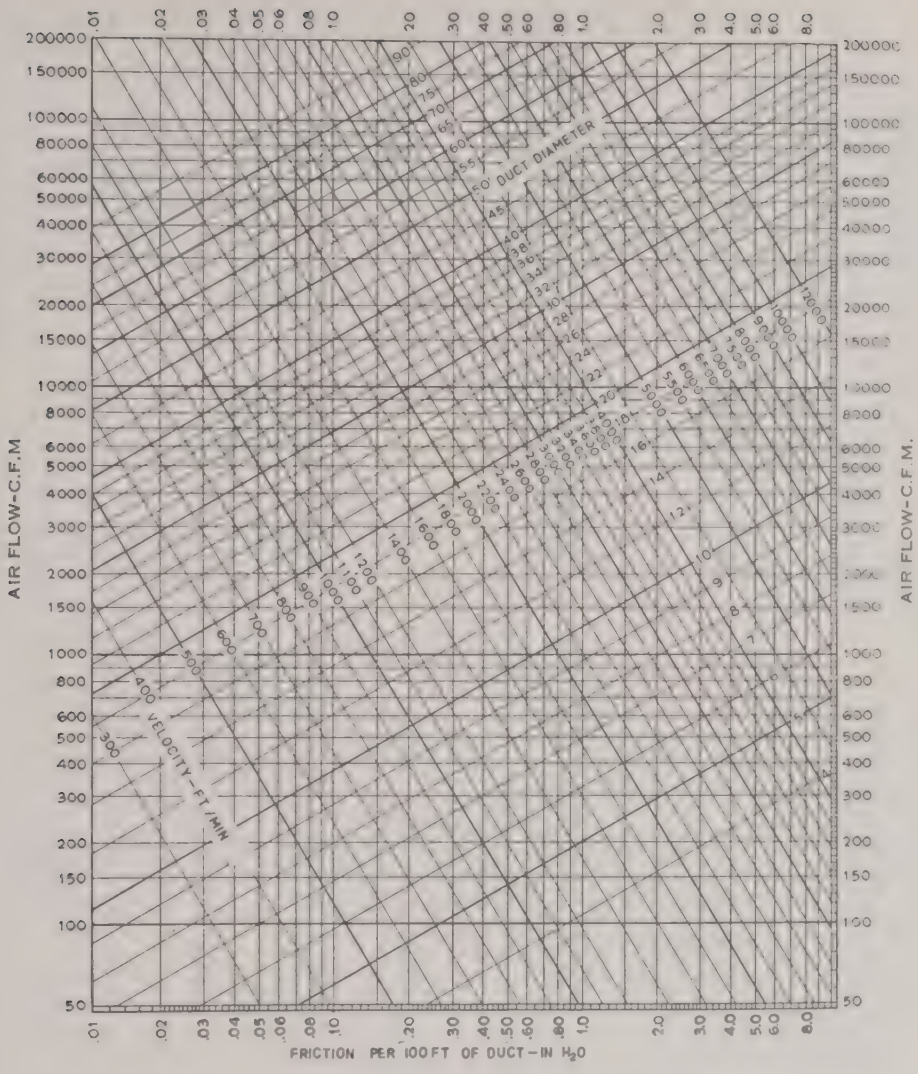

Fig. 6. Friction Losses in Round Ducts per 100 ft

Bottom horizontal: If the line of air discharge is horizontal and below the shaft.

Top horizontal: If the line of air discharge is horizontal and above the shaft.

Up blast: If the line of discharge is vertically upward.

Down blast: If the line of discharge is vertically downward.

In addition to the designations as to discharge, fans are supplied with different arrangements, and single or double width, single or doublet inlet. A single inlet fan takes in air from only one side; a double inlet fan from both sides. Double inlet fans are usually approximately twice as wide as single inlet fans.

II. Design of Duct Systems

This section discusses the design of a duct system from the standpoint of determining the correct sizes of ducts to carry the right amount of air to each supply opening or from each return opening, and of determining the pressure drop in the duct system. Hence, a considerable portion of this section is devoted to the pressure drops in various parts of a duct system. A knowledge of what causes pressure loss is of value not only in determining the **total pressure drop** of a given system, but also in avoiding arrangements which cause excessive pressure loss, and in balancing the pressure losses in the various

branches to proportion the air flow properly.

7. The **pressure drop** in a duct system is made up of two components:

a. Frictional resistance between the air and the duct walls

b. Dynamic losses occurring at pipe entrances, elbows, take-offs, grilles and other points where the velocity or direction of flow changes.

with a resistance not to exceed 0.08 in. per 100 ft. From the chart it is found that a 20-in. round duct at 900 fpm may be used.

Rectangular duct dimensions equivalent to round ducts are given by **Fig. 7.** A rectangular and round duct are equivalent if they have the same frictional resistance at the same air flow. As an example, rectangular ducts 10×40-in. and 20×20-

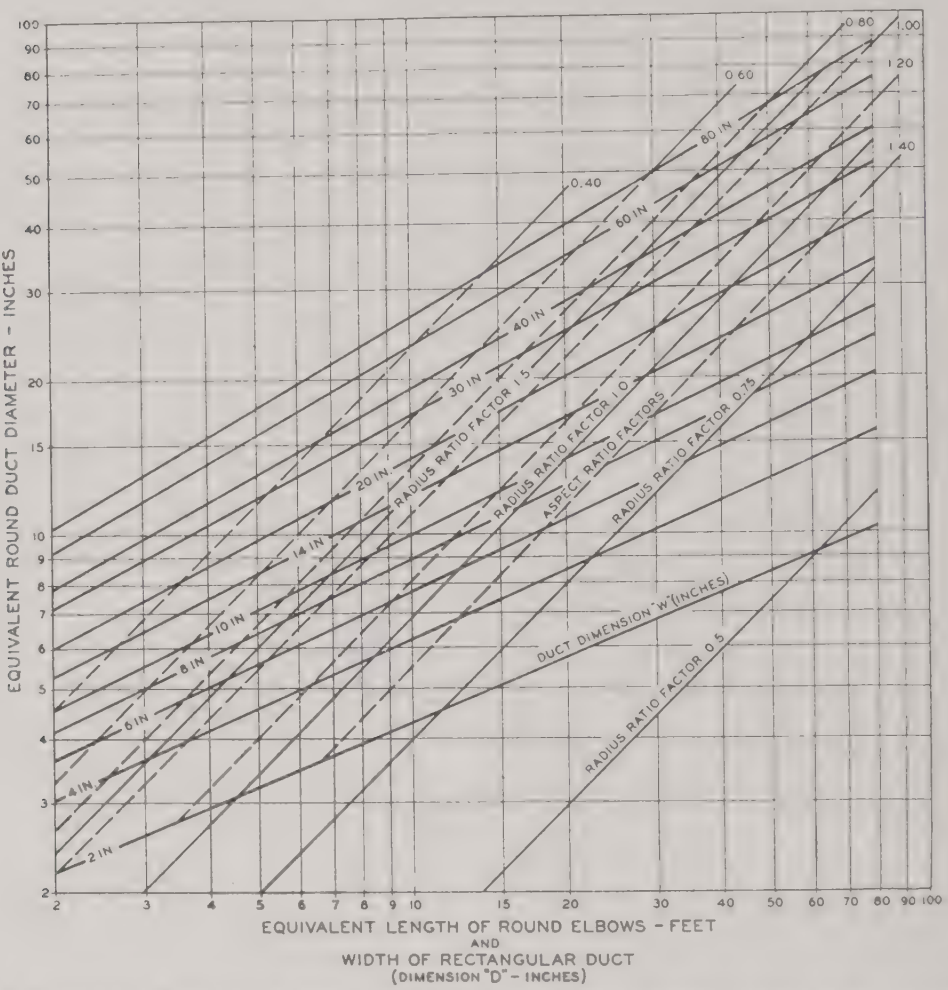

Fig. 7. Equivalent Round and Rectangular Ducts Having Same Friction Loss at Same Air Flow

Friction losses may be expressed by the usual hydraulic equation for fluid flow. For practical engineering purposes this has been converted to the form of **Fig. 6.** This chart gives friction loss in inches of water per 100 ft of straight duct having a circular cross section vs. cfm air velocity, and duct diameter. It is based on standard air.

As an example, it is desired to convey 2,000 cfm of standard air through a duct

in. have the same friction loss as a 20-in. round duct.

It should be noted that for the same quantity of air the velocity in a rectangular duct is somewhat lower than that in an equivalent round duct.

Radius ratio losses may be read directly from the equivalent duct chart, Fig. 7, as follows. To find the length of pipe equivalent to a 12-in. round elbow with a 1.5

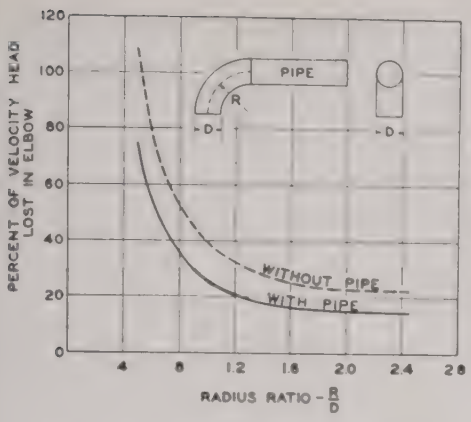

Fig. 8. Resistance of Round Elbows

radius ratio, find the point of intersection of the horizontal diameter line 12-in., and the sloping curve marked 1.5 radius ratio. The answer is given on the bottom scale.

The pressure loss through a given rectangular elbow varies according to its shape. Accordingly the aspect ratio, W/D, curves also appear on Fig. 7. For example, take an elbow where D (as on **Fig. 9**) is 12 in. and w is 24 in. The point of intersection on the chart will lie on the 0.8 aspect ratio factor curve represented by a dashed sloping line. From this point follow the horizontal (equivalent diameter)line to where it intersects the 1.5 radius ratio curve, represented by a solid sloping line. Then we may read on the bottom scale the equivalent length of an 18-in. round elbow with a 1.5 radius ratio, or about 14 ft. To get the equivalent of the rectangular duct one must multiply this by the aspect ratio factor of 0.8.

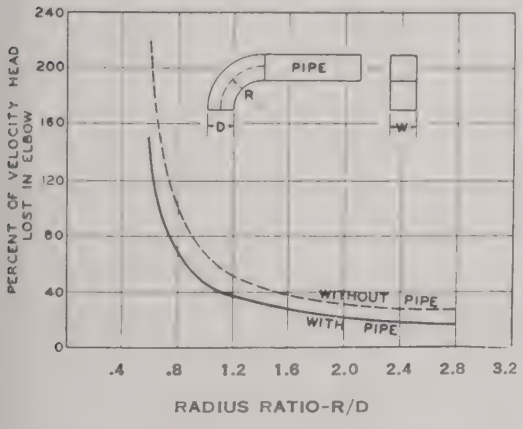

Fig. 9. Resistance of Square Elbows with Round Corners

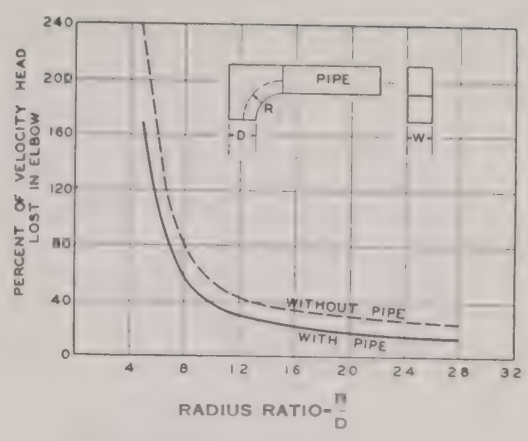

Fig. 10. Resistance of Square Elbows with Square Corners

8. Dynamic losses are usually expressed as a certain number of velocity heads. Velocity head is given by Equation (1)

$$H_v = \frac{v^2}{2g} \tag{1}$$

where

H_v = velocity head, ft of air
v = velocity, fps
g = 32.2 (acceleration of gravity)

Pressures in air conditioning systems are commonly measured in in. H_2O, and velocities in fpm. Equation (1) thus becomes:

$$h_v = \frac{(V/60)^2}{2g} \frac{12w}{62.4}$$

$$h_v = \left(\frac{V}{1097}\right)^2 w \tag{2}$$

where

h_v = velocity head, in. H_2O
V = velocity, fpm
w = density of air, lb per cu ft

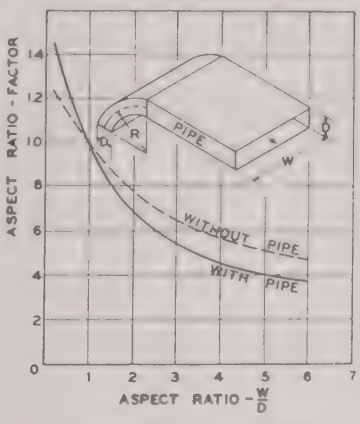

Fig. 11. Effect of Aspect Ratio on Resistance of Elbows

For standard air this reduces to

$$h_v = \left(\frac{V}{4005}\right)^2 \qquad (3)$$

9. Pressure losses in **90° elbows** may be determined from **Figs. 8** and **11.** Figs. 8, 9, and 10 give the per cent velocity head lost vs. the ratio of the radius of curvature of the center line of the elbow to the depth of the duct. The solid curves are for elbows

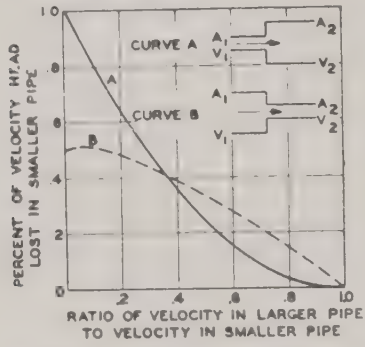

Fig. 12. Effect of Changes of Section

followed by a section of straight pipe several diameters in length. The dotted curves are for elbows discharging directly.

Pressure losses **in round elbows** are given in Fig. 8. The solid curve is taken directly from Bussey[2], while the dotted curve is calculated from the other by means of the data for pipe regain given by Wirt[3].

Pressure losses **in square elbows** are given in Figs. 9 and 10. These are taken from Wirt[3], who has given most satisfactory compilation of data on losses in rectangular elbows.

It will be noted that the pressure loss decreases as the radius ratio increases, but that a ratio in excess of 1.5 affords relatively little additional saving. It is also to be noted that an elbow discharging directly has a greater loss than one followed by a section of straight pipe. In the latter case there is a regain of static pressure in the pipe as the crowded portion of the air stream in the outer portion of the elbow diffuses in the pipe beyond the elbow. The amount of the pipe regain varies from 30 to 40%.

A comparison of Figs. 9 and 10 shows that a rectangular elbow with a square outer corner is about 10% better than one with a rounded corner. Correction factors, by which to multiply the losses given in Figs. 9 and 10, when the duct is of rectangular cross section with various aspect ratios, are given in Fig. 11. **Aspect ratio** is the width of the elbow measured along the axis of the turn divided by the depth of the elbow measured along the radius of the turn. Fig. 11 illustrates an elbow with an aspect ratio greater than 1. It may be seen that the loss in a rectangular elbow decreases as the aspect ratio increases. This is just opposite to the variation in frictional resistance of straight pipe, where the loss increases if one dimension is much greater than the other.

From these curves it may be deduced that right angle elbows may be used, with a very low pressure loss, if fitted with properly designed turning vanes, which really break up the elbow into a number of parallel elbows with advantageous radius and aspect ratios.

10. Pressure losses in changes of **cross sectional area** of the duct are given in **Fig. 12.** In Fig. 12 the change of section is abrupt. Curves *A* and *B* both give the loss in terms of the velocity head in the smaller pipe. In **Fig. 13** the change of section is gradual. The angle of divergence factor is applied to the per cent velocity head given by curve *A* of Fig. 12. For ex-

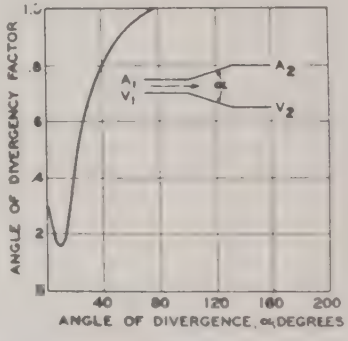

Fig. 13. Angle of Divergence, Degrees

ample, if the ratio A_1/A_2 is 0.5 (or $V_2/V_1 = 0.5$) the per cent velocity head lost if the change of section is abrupt is 25. However, if the change of section is gradual, with an angle of divergence, α, of 13°, the angle of divergence factor is 0.2. Hence the actual loss is

$$h_f = 0.2 \times 0.25\, h_v = 0.05\, h_v$$

A few special cases are given in **Fig. 15.** Fig. 15a is an elbow of less than 90°. Losses depend on the elbow angle, α. Multiply the

losses for 90° elbows given in Figs. 8, 9, 10 and 11 by the elbow angle factor from **Fig. 14.**

Fig. 15b relates to the lost head when a duct discharges directly into a large space and Fig. 15c to the loss when the opening from a large space to a duct is flush with the wall of the space. Fig. 15d refers to the case where the duct projects somewhat into the space from which air is exhausted.

11. The simplest way to keep losses due to **duct turns in square elbows** at a minimum is to use large radius ratios and large aspect ratios. In case space limitations make this impracticable **duct turns with vanes** will reduce excessive loss, see **Fig. 16.** To determine the characteristics of vanes this formula may be used

$$N = \left(\frac{6D}{w} - 1\right)$$

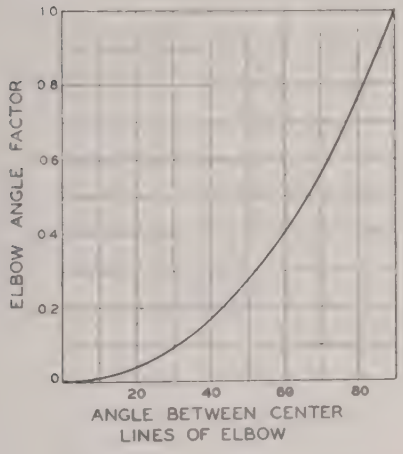

Fig. 14. Elbow Angle Factor

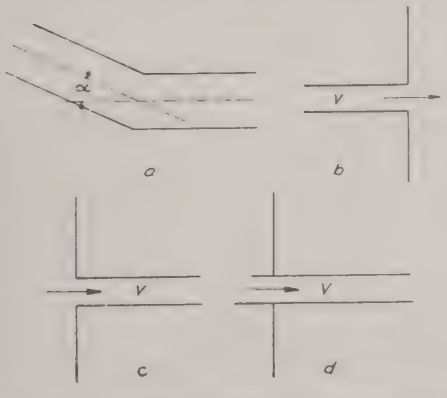

Fig. 15. Special Cases of Pipe Ends

a—Multiply values from Figs. 8, 9, 10 and 11 by angle factors from Fig. 13.
b—$h_f = 1.0h_v$
c—$h = 0.48h_v$
d—$h = 1.0h_v$

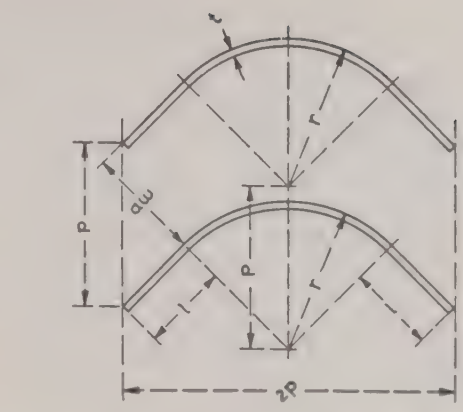

Fig. 16. Use of Duct Vanes

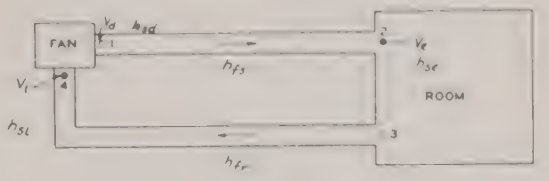

Fig. 17

where

D = depth of duct, in.
w = width of duct, in.
N = number of blades (using nearest whole number)

Again the axial width, Aw, has this relation

$$Aw = D/(N+1)$$

where

p = 1.14 Aw
r = 1.28 Aw
l = 0.75 Aw (=lip on blade)
t = Aw/16 (=thickness)

Blades should be as thin as possible without endangering loss of their shape.

Calculation of Fan Pressure

12. One of the objects of calculating the resistance of the duct system is to determine the pressure against which the fan must deliver the required amount of air. **Fig. 17** illustrates a fan exhausting air from and returning it to a room. The symbols used in Fig. 17 have the following significance:

V_d = Discharge velocity at the fan
h_{sd} = Static pressure at fan discharge

h_{fs} = Total resistance of the supply system, including the duct and heating and cooling surfaces (in blow-through systems)
V_o = Discharge velocity of air from the supply grilles
h_{so} = Static pressure in the room
h_{fr} = Total resistance of the return system, including the duct, filters and heating and cooling surfaces (in draw-through systems)
V_i = Air velocity approaching the fan inlet
h_{si} = Static pressure at fan inlet

Writing Bernoulli's equation between points 1 and 2,

$$\frac{V_d^2}{2g}+h_{sd}=\frac{V_o^2}{2g}+h_{so}+h_{fs} \tag{4}$$

and between points 3 and 4,

$$h_{so}=\frac{V_i^2}{2g}+h_{si}+h_{fr} \tag{5}$$

Adding Equations (4) and (5), and transposing.

$$\left(\frac{V_d^2}{2g}+h_{sd}\right)-\left(\frac{V_i^2}{2g}+h_{si}\right)=\left(h_{fs}+h_{fr}+\frac{V_o^2}{2g}\right) \tag{6}$$

From the discussion on fans, it can be seen that the left side of the above equation is the total pressure of the fan. Hence

$$H_t=h_f+\frac{V_o^2}{2g} \tag{7}$$

where

H_t = Total pressure of the fan
h_f = Total resistance of supply and return ducts, filters and heating and cooling elements
V_o = Discharge velocity of air from the supply grilles

In the foregoing it is stated that the most generally used expression of fan static pressure is:

$$H_s=H_t-\frac{V_d^2}{2g} \tag{8}$$

where

H_s = fan static pressure

Substituting (8) and (7)

$$H_s=h_f+\frac{V_o^2}{2g}-\frac{V_d^2}{2g} \tag{9}$$

The result of this analysis is that, strictly speaking, the fan should be selected to have a static pressure equal to the sum of all the frictional and dynamic losses of the supply and return system plus the velocity head corresponding to the supply grille discharge velocity, but that the system should be credited with the velocity head corresponding to the fan outlet velocity. However, it is common practice to reserve the fan outlet velocity pressure as a factor of safety. This seems to be reasonable, especially since fan catalogs do not at present show the effect of the arrangement of the inlet connection to the fan on the fan pressure. It has been shown that this arrangement may have a very considerable influence on the fan pressure (4). Furthermore, transitions are frequently used between the fan outlet and the main trunk of the duct system which do not fully preserve the velocity pressure at the fan discharge. Therefore, it is common practice to select the fan on the basis of Equation (10). That is, the fan static pressure is calculated as that required to overcome the total resistance of the system plus the supply grille discharge velocity head.

$$H_s=h_f+\frac{V_o^2}{2g}. \tag{10}$$

Design Procedure

13. Any duct system consists of one or more trunks leading from or to a plenum chamber, fan or other common point. Regardless of which path the air follows, it finally issues from a grille into a room at atmospheric pressure. The pressure drop must, of necessity, be the same along each path, and hence the volumes of air flowing in each trunk or branch will automatically distribute themselves to fulfill this condition. It is the object of the ideal design to have this automatic distribution in accordance with the desired flow in each branch, and with all volume control dampers open.

The following logical procedure may be followed in designing a duct system:

a. Sketch a line diagram showing each

trunk and branch duct, and the location of each supply opening (make a separate similar sketch for the return-air system). Mark on the diagram the location of each elbow and change of section that will be required. This sketch should be based on a careful consideration of the building construction and of the most satisfactory arrangement of supply and return openings from the standpoint of air distribution. Care must also be exercised to keep the system as simple as possible.

b. Determine the air quantity from each supply opening on the basis of the **heat loss or heat gain** of the zones served by that opening.

c. Mark on each supply and return opening the **air velocity** to be employed, after having determined these from considerations of air distribution.

d. Mark on each section of the system the **amount of air** flowing in that section.

e. Determine the **sizes** of all main and branch ducts by first assigning a tentative velocity for each section and then calculating the duct size from the air volume and velocity. Trunk velocities should be reduced as the distance from the fan increases, and short branches near the fan should have higher velocities than long branches far away.

f. Calculate the **pressure drop** through several of the more important paths. If these differ from each other by more than approximately 20%, increase the velocities in the paths with the lower resistances or decrease the velocities in the paths with the higher resistances. Finally recalculate the pressure drops.

g. Estimate whether all remaining paths have approximately the same pressure drop. If in doubt, calculate the pressure drops.

h. Repeat the above procedure for the return air system.

i. The total duct resistance is the sum of the resistances of the supply and return systems.

Another method that is frequently employed, after completing steps a to d above, is to determine the sizes of all main and branch ducts on the basis that the friction loss per foot will be the same in all ducts. Unless the system balance is to be entirely dependent on damper settings, this method offers the disadvantage that the resistances of the shorter branches are automatically lower than those of the longer branches. However, it may be used as a starting point for step e in the method outlined above, especially when the pressure available at the fan is specified.

14. The following **example,** based on the simple duct system for a small cooling job in **Fig. 18,** illustrates the application of the data and procedure described above. This example is based on the premise that the duct system must be designed to have a pressure drop not to exceed 0.30 in. H_2O.

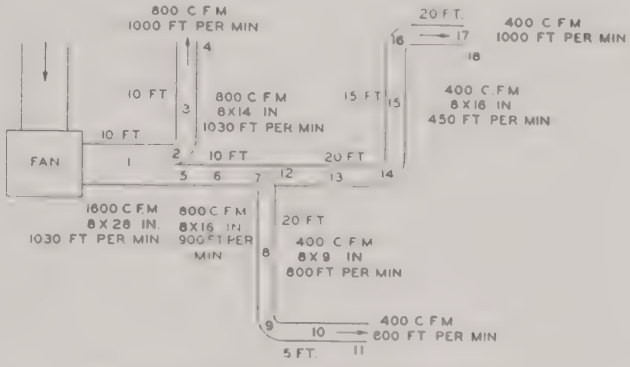

Fig. 18. Example of Pressure against Fan

The quantity and velocity of air discharged from each grille is determined to meet the requirements for cooling and air distribution.

The pressure drops through grilles are determined from catalog data, and the discharge velocity head is calculated from Equation (3). The sum of these two may be termed the total discharge loss. These may be tabulated as follows:

Grille No.	*4*	*11*	*18*
Grille pressure drop	0.050	0.032	0.050
Discharge velocity head	0.062	0.040	0.062
Total discharge loss	0.112	0.072	0.112

As a first approximation, assume that one-third of the duct losses occur in the return system. This leaves 0.20 in. for the

Table 2. Duct Design Example

Duct section	Round diam, in.	Rectangular dimensions, in.	Velocity, fpm	Friction per 100 ft, in. H_2O	Total friction, in. H_2O	Velocity head, in. H_2O
1	16	8 ×28	1030	0.14	0.014	0.066
3	11	8 ×14	1030	0.26	0.026	0.066
6	12	8 ×16	900	0.17	0.017	0.051
8, 10	9	8 × 9	800	0.20	0.050	0.040
13, 15, 17	12	8 ×16	450	0.046	0.025	0.013

supply system, Fig. 18. Design the longest path (to grille No. 18) to have a pressure loss not to exceed 0.24 in., and then balance the other two paths for the same loss. The total discharge loss is 0.112, leaving 0.088 in. for friction, elbows, etc. Assume that one-third of this is dynamic loss. This leaves 0.058 in. for friction alone. The total length of the run is 75 ft. Hence the allowable friction loss is 0.077 in. per 100 ft if all sections were to have equal friction per ft. However, in order to help balance the pressure drops in the paths to grilles 4 and 11 it is advisable to increase the velocities slightly in sections 1 and 6, and to decrease the velocity in section 13. Assume that the losses in sections 1, 6 and 13, 15, 17 are each one-third of the allowable 0.058. Determine the allowable friction loss per 100 ft and from Fig. 6 select the correct size of round duct.

Section No.	*1*	*6*	*13*
Allowable friction	0.019	0.019	0.019
Allowable friction per 100 ft	0.19	0.19	0.035
Cfm	1600	800	400
Round duct diameter, in.	16	12	12
Actual friction per 100 ft	0.14	0.17	0.046
Velocity in round duct, fpm	1120	1020	510

Table 3

Section No.	Dynamic loss, in. H_2O	Nature of loss
2	0.021	Elbow
5	0	Velocity reduction
7	0.013	Elbow
9	0.010	Elbow
12	0.013	Velocity reduction
14	0.004	Elbow
16	0.004	Elbow

Section 3 must have a friction equal to twice that of section 1, or about 0.36 in. per 100 ft. However, in Fig. 6 it is found that this corresponds to a velocity of 1,400 fpm. This would introduce a considerable dynamic loss in section 2. Hence, compromise with an 11-in. round duct, having a velocity of 1,200 fpm and a friction loss of 0.26 in. per 100 ft.

The allowable loss for sections 8 and 10 is 0.019 plus the difference between the discharge losses at 18 and 11, making a total of 0.059, or 0.24 in. per 100 ft. Use a 9-in. round duct, having a velocity of 910 fpm and a loss of 0.20 in. per 100 ft.

Note that in the above preliminary layout it is not necessary to go to the extreme of using fractional diameters in the round duct sizes.

Now select from Fig. 7 the dimensions of rectangular ducts corresponding to the round sizes calculated above. Calculate the velocity and velocity head in each duct section. It is assumed here that a duct having a depth of 8 in. is convenient. See **Table 2.**

Dynamic losses will now be calculated at the various sections. All elbows will be assumed to have a radius ratio of 1.5. From Fig. 9, the per cent of velocity head lost in a square elbow is 26.

The take-off at section 2 is considered an elbow with aspect ratio 8/14=0.57. From Fig. 11 the aspect ratio factor is 1.2. Hence the loss at 2 is

$$1.2 \times 0.26 \times 0.066 = 0.021 \text{ in.}$$

At section 5 there is a velocity reduction from 1,030 to 900 fpm. Since the velocity ratio is 0.87 it can be seen from Fig. 12, curve A, that the loss is negligible.

In the take-off at 7 there are losses from the elbow effect and from the velocity de-

Table 4. Summary of Example

Grille No. 4		Grille No. 11		Grille No. 18	
Duct section	Friction	Duct section	Friction	Duct section	Friction
1	0.014	1	0.014	1	0.014
2	0.021	5	0	5	0
3	0.026	6	0.017	6	0.017
4	0.112	7	0.013	12	0.013
		8, 10	0.050	13, 15, 17	0.025
		9	0.010	14	0.004
		11	0.072	16	0.004
				18	0.112
Total	0.173		0.176		0.189

The dynamic losses at sections 9, 12, 14 and 16 may be calculated in a similar manner. These are summarized as in Table 3. The total pressure drop in each path may now be calculated as shown in Table 4.

Of course these calculations are not perfectly accurate. Adjustable splitter dampers may be installed at sections 2 and 7. If this is not convenient, volume dampers may be installed in each branch.

References

1. Standard Test Code, National Association of Fan Manufacturers.

Table 5. Typical Single Inlet Centrifugal Blower Wheels (Torrington)

Dimensions		No. of Blades	Sp	Cfm	Hp	Sp	Cfm	Hp
Diam	Width			1,725 rpm			3,450 rpm	
4¼	2½	30	0	220	0.035	0	440	0.28
			0.2	190	0.030	0.8	380	0.24
			0.4	150	0.025	1.6	300	0.20
				1,140 rpm			1,725 rpm	
5½	3 7/16	32	0	230	0.021	0	350	0.075
			0.1	195	0.018	0.3	280	0.056
			0.2	155	0.015	0.6	156	0.030
6 7/16	3½	32	0	360	0.050	0	550	0.167
			0.2	285	0.042	0.5	430	0.130
			0.4	180	0.037	1.0	220	0.075
7 11/16	4 13/16	36	0	750	0.160	0	1,160	0.62
			0.2	670	0.140	0.5	990	0.52
			0.4	560	0.120	1.0	840	0.44
9 11/16	5 7/16	36	0	1,160	0.35	0	1,750	1.20
			0.3	1,050	0.30	1.0	1,490	1.00
			0.6	900	0.26	2.0	1,060	0.65
				860 rpm			1,140 rpm	
11	5½	42	0	1,225	0.033	0	1,620	0.075
			0.2	1,140	0.029	0.6	1,420	0.06
			0.4	1,030	0.026	1.2	1,060	0.043

crease. However, the latter is again negligible (velocity ratio = 0.89). The aspect ratio of this elbow is 0.89, and from Fig. 11 the aspect ratio factor may be taken as 1.0. Assume the loss in this elbow to be 26% of the velocity head in section 6.

2. Frank L. Bussey, Trans. A.S.H.V.E., p. 366, 1913.

3. Loring Wirt, "New Data for the Design of Elbows in Duct Systems," *G. E. Review*, p. 286, June 1927.

4. A. A. Berestneff, "Fan Selection and Nomenclature," *Refrig. Eng.*, March 1936.

37. AIR CONDITIONING CALCULATIONS

AIR conditioning is defined as the simultaneous control of temperature, humidity, air motion and air purity in any given space. This definition gives rise to the terms "winter air conditioning," "summer air conditioning," "complete air conditioning," when considering air conditioning for comfort. Industrial air conditioning is not considered in this chapter.

Winter air conditioning includes heating, humidification, air distribution and air cleaning, when outside temperatures are below the inside or room temperature. It is the well known science of heating and ventilating, involving the calculation of heat loss from conduction through building construction, heating requirements for ventilating air, and heating requirements for humidification.

Summer air conditioning includes cooling, dehumidifying, air distribution and air cleaning, when outside temperatures are above the inside or room temperature. It involves the calculation of heat gain from conduction through building construction, direct sunlight heat gain through glass and indirect sunlight heat gain through opaque building construction, heat gain from lights and motors (including fan motor) within the conditioned space; as well as heat and moisture gain from people, from outside air required for ventilation, and from appliances and other heat and moisture generating equipment within the conditioned space.

Complete air conditioning includes both heating and cooling, and both humidifying and dehumidifying, together with air distribution and air cleaning. It therefore involves all the calculations for both winter and summer air conditioning.

Design Conditions

1. The first step in the approach of an air conditioning problem is to determine the proper design conditions. Outside conditions are based on U.S. Weather Bureau statistics, and inside conditions upon the type and duration of occupancy, and also upon the outside temperature.

Inside conditions for cooling. The determining factor for inside design conditions is comfort measured by the chart shown in **Fig. 1** in terms of **effective temperature,** which is an index of conditions producing the same degree of comfort (see Chap. 35). Design conditions have also been recommended for various cities in the United States as in Table 2, Chap. 11. Inside conditions are given in terms of effective temperature, separately for occupancies under 40 minutes and for occupancies over 40 minutes. Inside temperature and humidity conditions corresponding to the recommended effective temperature can be determined by reference to Fig. 1. For inside conditions for cities not listed in Table 2, Chap. 11, reference should be made to the comfort chart of Fig. 1. An effective temperature condition within the comfort zone should be chosen for cooling such that the differential between the inside and outside dry-bulb temperature is as low as practicable. Temperature differentials of from 10 to 20 deg are most desirable, and differentials of 25 to 30 deg should be avoided if possible. However, inside conditions higher than 80 to 82 F dry bulb are seldom very comfortable in practice, as the very low humidities required for comfort at high dry-bulb temperatures are difficult to maintain. Therefore, in very warm climates, a differential to 25 to 30 deg may be necessary at extreme outside conditions. Normally, humidities are kept within the range of 35 to 65%.

Inside conditions for heating. For heating, the recommended practice for inside design conditions for human comfort are 70 F dry-bulb temperature and 35% relative humidity with 30 F outside temperature. In warmer climates where the outside temperature seldom gets down to freezing, it is often desirable to maintain inside temperatures from 72 to 75 F.

Outside conditions. The tables of Chap-

ter 11 give recommended outside design conditions for both heating and cooling seasons. For cities not listed in this table, reference can be made to local Weather Bureau reports for maximum and average maximum conditions of dry-bulb and wet-bulb temperatures for the warmest month for cooling,[1] and to the minimum and average minimum dry-bulb temperatures for the coldest month for heating.[2] The **outside design conditions** should be somewhere between the maximum and average maximum for cooling, and between the minimum and average minimum for heating. It is seldom good engineering to design for the extreme maximum or absolute minimum outside conditions.

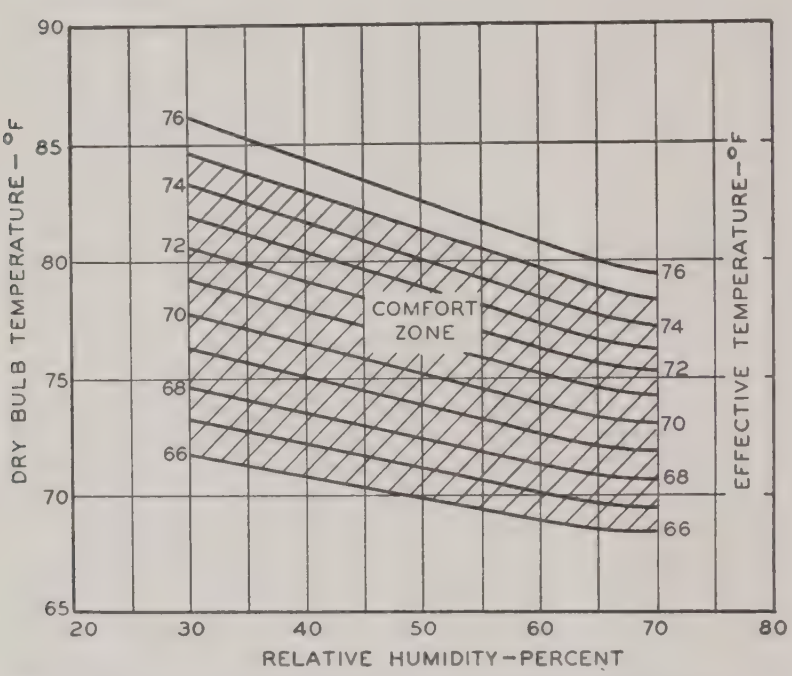

Fig. 1. Summer Comfort Zone Section of Comfort Chart for Air Motion 15-25 fpm Using Basic Effective Temperature

Often wet-bulb temperatures are not obtainable from Weather Bureau data, but relative humidities are usually given. Wet-bulb temperatures may be obtained by taking simultaneous conditions of dry bulb and relative humidity and referring to the psychrometric chart as in **Fig. 2.**

In the absence of more specific data for a required locality not listed in tables of Chap. 11, compare its Weather Bureau statistics with similar statistics for listed cities to determine which is nearest climatically. Then the design conditions (both inside and outside) may be selected from this table for a city having a similar climate to the required locality. However, the comparison may require using one city's heating design conditions and another's cooling design conditions to approximate closely the climate of the required locality.

Sensible Heat Load

2. After selection of the proper design conditions, the first step in the air conditioning calculations is to determine the sensible head load. **Heat leakage calculations**[3] for summer air conditioning are similar to heat loss calculations for heating except that the flow of heat is reversed. In the heat gain calculations, the phenomenon of **radiant heat** gain from the sun plays a prominent part. Furthermore, heavy building construction, particularly masonry walls, has a considerable heat absorbing capacity which interposes a lag in the heat flow which often must be considered for reasonably accurate results in heat gain calculations.

It is often desirable in air conditioning computations to estimate the temperature in **attic spaces** between the roof and the furred ceiling below. This can readily be done as the heat transfer from the outside to the attic space must equal the heat transfer from the attic to the conditioned space below. The following formula may be used, solving for X, the attic temperature:

$$SU(T-X) = su(X-t)$$

Where

S = roof area in sq ft
s = ceiling area in sq ft
U = roof transmission factor, Btu per sq ft hr, deg F

Table 1. Direct Sunlight Gain on Glass[4,5]

(Btu per hr sq ft net glass area)*

Latitude	20°			30°			40°		
	July	Aug.	Sept.	July	Aug.	Sept.	July	Aug.	Sept.
Northeast	148	126	91	144	118	94	138	109	58
East	178	181	179	179	181	174	179	178	165
Southeast	86	120	149	104	137	165	132	157	176
South	—	14	60	19	58	107	65	104	148
Southwest	86	120	149	104	137	165	132	157	176
West	178	181	179	179	181	174	179	178	165
Northwest	148	126	91	144	118	94	138	109	58
Flat skylight	263	258	242	256	244	218	242	220	185

* Values given represent the maximum for each exposure.
For light-colored shades or inside venetian blinds, use 60 % of unshaded values if shade is at least ½ drawn. For outside ventilated awnings and venetian blinds arranged to cut off all direct sunlight use 15 Btu per sq ft per hr for all exposures instead of values in table; for ordinary awnings not ventilated, use 25 Btu per sq ft per hr. When using sash area normally having about 85 % glass area, multiply above values by 0.85. Use other appropriate correction for different percentage of glass.

u = ceiling transmission factor, Btu per sq ft hr, deg F
T = outside roof temperature, deg F
t = inside conditioned space temperature, deg F
X = attic temperature, deg F

To consider carefully all of the complexities of variable heat flow through building construction would make the calculations unduly long and tedious. It is, therefore, common practice in commercial work to make a number of approximations which simplify the problem and in the total give reasonably accurate results. For shaded outside walls or roofs, the full temperature difference between the inside and the outside design conditions is used in the heat gain calculations, except in the case of extremely heavy masonry walls in climates where the nights are considerably cooler than the days.

Sun effect. For sunlit walls it is safe to use an equivalent outside temperature of 120 F in the United States, and for sunlit roofs and equivalent outside temperature of 140 F. For sunlit glass[4,5] add radiant heat gain in accordance with values given in **Table 1.** This radiant heat gain on sunlit glass is in addition to the transmission gain through the glass figured in the usual manner from the transmission factor, glass area, and the temperature differential. When calculating total cooling load, radiant heat gain from the sun need only be figured on the roof and on the exposure having the greatest radiant heat load.

When roofs or walls are painted with aluminum or have highly reflective, light-colored surfaces, the radiant heat gain through such walls or roofs may be reduced 50%. In other words, the surface temperature for roofs may be reduced to 120 F and walls to 110 F, when using an outside design condition of 100 F.

Heat lag. The matter of lag through heavy building construction is much harder to account for in the heat gain calculations. In many cases, the peak of wall transmission may not coincide with peak load for the cooling system due to this time lag in the transmission of heat. Ordinary frame construction such as used in residential work has but little capacity effect and full temperature differences should be used. Masonry wall time lag varies from 3 hr on a 6-in. concrete wall to 10 hr on a 22-in. thick, solid brick and tile, composition wall. Thus the heat from the sun effect on an east exposure may just be getting through the wall when the sun has moved around and is starting to shine on the west exposure.

The effect of this lag is usually averaged by figuring sun effect only on one exposure, the one having the greatest area, and the roof. For example, if the south wall has the greatest sunlit area of glass and wall surface, the sun effect should be figured on the south exposure and not on the east and

west. When extremely heavy masonry west walls are encountered, the radiant heat may not get through these walls until after the building occupants go home for the evening. In such cases, no radiant heat gain need be figured for the west wall, but should be figured for either east or south exposures, if they are sunlit.

per hr. Part of this is sensible heat, and the remainder is latent heat which adds moisture or humidity to the air. The total heat from people increases enormously as their activity increases, this increase being largely due to the greatly increased latent heat produced. **Fig. 3** gives both sensible and latent heat quantities produced by

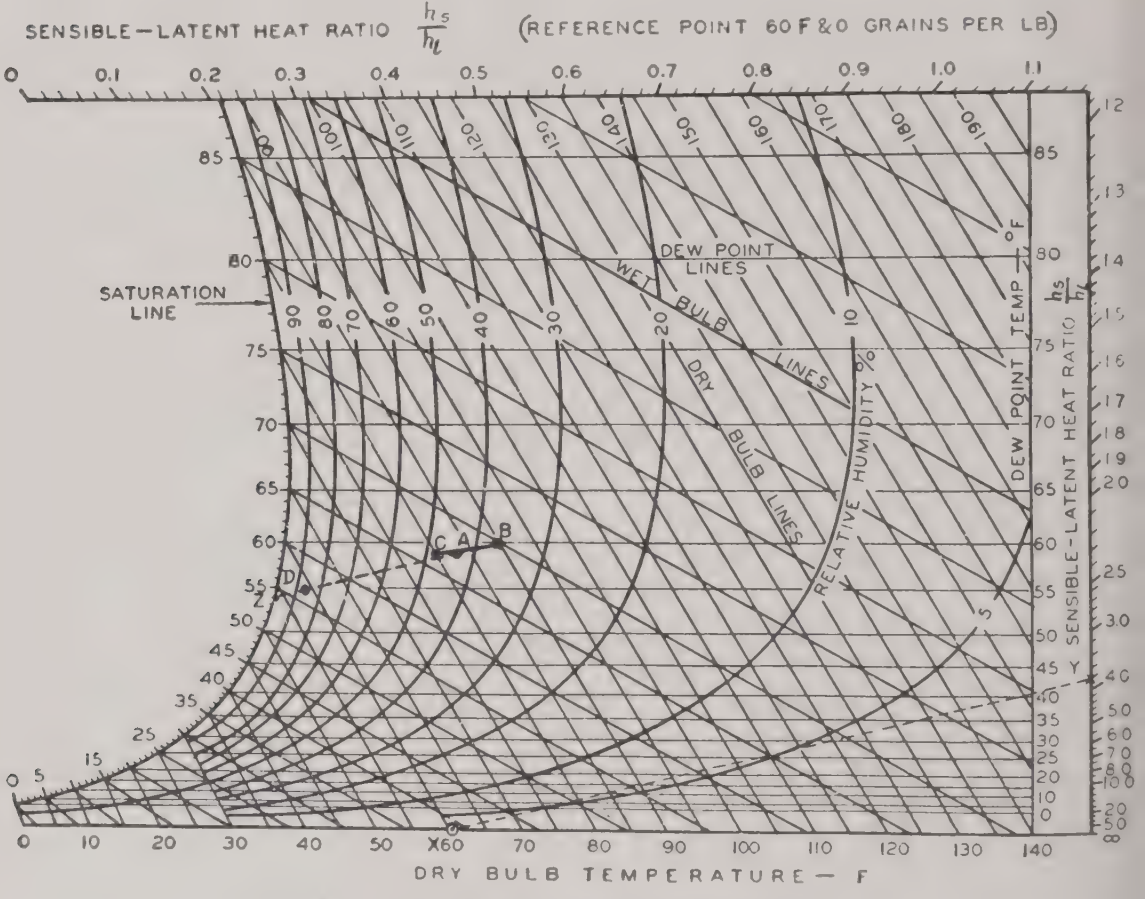

Fig. 2. Psychrometric Chart (See Example II)

Internal heat. In addition to the transmission and radiant heat gain through walls, floors, ceilings and glass areas, there must be added to the sensible heat load, the heat gain from any objects in the conditioned space that give off heat,[6] and the heat gain from the outside air required for ventilation or to offset infiltration. Under this heading come people, lights, motors (including heat equivalent of brake horsepower of fan motor used for air circulation), and any heat evolved in processing.

People. The total heat evolved by people at rest amounts to approximately 400 Btu

people at rest, and at various conditions of activity, for different room temperatures.

Food. For restaurants an allowance of about 30 Btu per person should be added, to account for the sensible heat gain from the food. However, where the heat generating equipment such as steam tables, coffee urns, etc., is in the conditioned room and is accounted for in the heat gain calculations (as in the case of a cafeteria), this allowance of 30 Btu per person is not included in addition.

Electricity. The heat generated by lights, motors, and electrical equipment

within the conditioned spaces must be included in the sensible heat load. Lighting or electrical equipment load in watts multiplied by the factor 3.4 will give the heat equivalent in Btu per hr. Often the electrical equipment will not be operating continuously and the proper time factor should be applied in such cases. **Table 2** gives approximate values for heat gain from motors located within the conditioned space. Usually motors are not fully loaded besides not operating continuously. Therefore, the proper load and operating-time factors should be applied to reduce the heat gain accordingly. An average overall factor for a considerable number of motors operating in the conditioned spaces is 50%.

Table 2. Heat Gain from Motors

Nameplate rating, hp	Heat gain in Btu per hr per hp	
	Connected load in same room	Connected load outside of room
$\frac{1}{8}$ to $\frac{1}{2}$	4,250	1,700
$\frac{1}{2}$ to 3	3,700	1,150
3 to 20	2,950	400

The heat gain from the fan motor used to circulate the conditioned air must also be included in the sensible heat calculations. Only the heat equivalent of the fan brake horsepower need be added since the motor heat losses will not be absorbed in the air stream. It is common practice, however, to use the nameplate horsepower (rather than the actual brake horsepower) for an additional factor of safety, multiplying the horsepower by 2,545 to obtain the heat equivalent in Btu per hr.

Outside air. To the sensible heat gains, calculated as explained above, must be added the sensible heat gain from infiltration and from outside air introduced, to obtain the total sensible heat gain. Infiltration may be figured by the Code of Application Engineering Standards 4 of the ACRMA.[4] The latter gives factors for infiltration through swinging and revolving doors, for infiltration through cracks around windows and doors, and for ventilation standards (outside air requirements) for various applications.

However, for simplicity, an arbitrary method of offsetting infiltration when cooling, by figuring a minimum of a one hour air change of outside air in the conditioned space, will give reasonable approximate results. Thus, when the outside air introduced is greater than the equivalent of a one-hour air change, no infiltration need be figured. If the actual outside air requirement is less than a one-hour air change, an additional amount should be added for infiltration so that the total will be equivalent to a one-hour air change. This minimum outside air quantity is in addition to any exhaust that may be taken from the conditioned space.

Table 3. Heat Gain from Restaurant Equipment*[6]

	Sensible heat in Btu per hr	Latent heat in Btu per hr
Coffee urns (steam) per gal capacity	800	800
Coffee urns (electric) per gal capacity	700	500
Coffee urns (gas burning) per gal capacity	800	800
Steam tables (per sq ft top surface)	400	800
Steam tables (electric) per sq ft top surface	200	350
Steam tables (gas burning) per sq ft top surface	850	430
Gas toasters (640 slices per hr)	12,000	5,000
Gas toasters (360 slices per hr)	7,700	3,300
Gas toasters (small size)	5,750	830

* When equipment is hooded with positive exhaust above values may be reduced 50%.

Latent Heat Load

3. The second step in the cooling load calculations is to determine the latent heat load. This load comes from the moisture pickup or humidity rise in the conditioned space, and from the excess moisture in the outside air if the outside dew point is higher than the inside design dew point. The moisture pickup in the conditioned space comes from people and any special processes that give off moisture.[6] **Fig. 3** gives the latent heat quantity produced by people at rest and at various conditions of activity.

In dealing with cafeterias and restau-

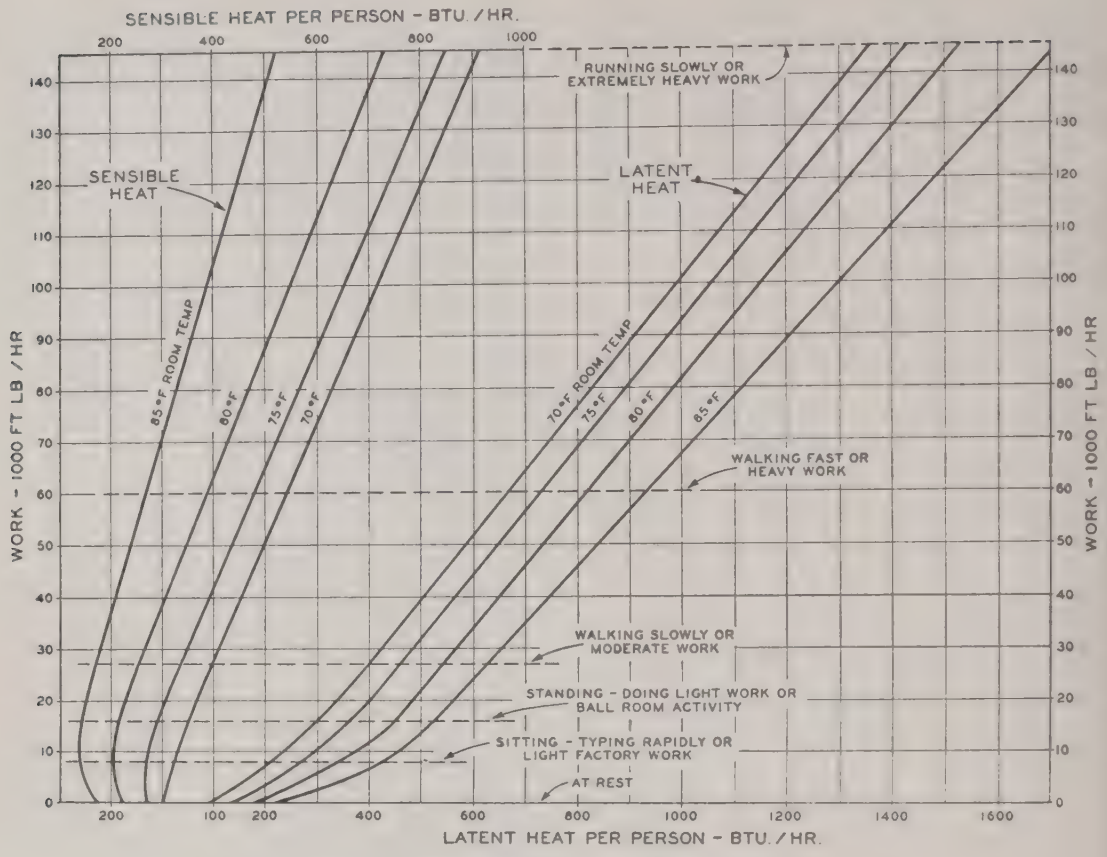

Fig. 3. Heat Gain from People, Btu per hr

rants we have many sources of moisture load to consider. **Table 3** gives both sensible and latent heat loads for coffee urns, steam tables, and other restaurant equipment. If such equipment is adequately hooded with a positive exhaust, both the sensible and latent heat loads may be reduced 50%. If the steam tables, coffee urns, etc., are in the conditioned space, no additional load is ordinarily figured for the food. Otherwise a factor of 30 Btu per person should be added to account for the latent heat from the food.

The heat equivalent of moisture removed is 1,060 Btu per lb and there are 7,000 grains of moisture per lb. Thus to change grains of moisture removed to latent heat load in Btu divide by 7,000 and multiply by 1,060.

Air Quantities

4. Outside air. The outside air quantity must be determined before the total sensible and total latent heat loads can be calculated. As previously explained, the minimum outside air quantity for use in determining heat loads is equivalent to a one hour change of air in the conditioned space. For ventilation requirements[4] it is good practice to figure 10 cfm of outside air per person—25 to 40 cfm if they are smokers. In crowded places such as theaters, where the concentration of people is very great, an absolute minimum of 5 to $7\frac{1}{2}$ cfm of outside air is often figured for peak load conditions.

Supply air. The total air quantity circulated is a function of the internal sensible heat load and the temperature difference between the supply air and the conditioned room. The supply air temperature for cooling usually varies from 10 to 30 F below the room temperature. With the advance in air distributing methods, and in grille and plaque design, temperature differences of 20 to 30 F are being increasingly used without causing uncomfortable drafts.

The immediate method of calculating the supply air quantity is to divide the

total internal sensible heat load (outside air and fan motor loads not included) by the heat taken up per pound of air circulated. Thus, the sensible load in Btu per min, divided by the temperature difference (between the supply air and the room) and the specific heat of air (0.24), will give the pounds of air circulated per minute. Multiplying this value by the specific volume will give cubic feet of air per minute.

Return air. The return air quantity is, of course, merely the difference between the supply and the outside air quantities. Many engineers prefer to use pounds of air per minute in all the calculations, rather than cubic feet per minute, to avoid confusion due to the different specific volumes at various points in the air cycle. If the calculations are carried through using cubic feet per minute it simplifies the computations, and it is common practice to figure all air quantities in cubic feet per minute based on the specific volume at room conditions.

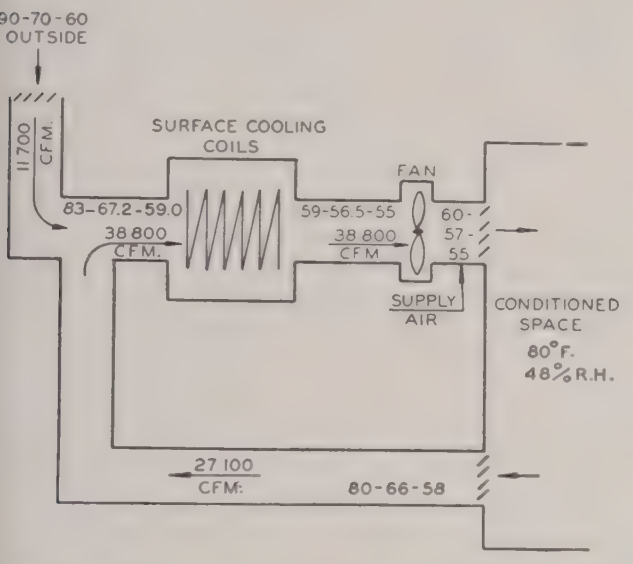

Fig. 4. Surface Cooling Cycle, Showing Dry-Bulb, Wet-Bulb and Dew-Point Temperatures at Various Points (Example II)

Load Calculations

5. The most important step in the refrigeration load calculations is the preparation of the "Btu sheet," the heat load estimate. **Example I** is a typical calculation sheet for a medium size department store. At the top, the various air properties corresponding to the outside and inside design conditions should first be set down. The outside dry and wet-bulb design conditions are obtained from Table 2, Chap. 11, or other weather data available as previously described. The inside design conditions are determined by reference to Figs. 1 and 2, and by considering the phenomenon of effective temperature. The various other air properties necessary for the air conditioning calculations can then be determined from the psychrometric chart, and tables of air properties (see Chapters 4 and 10).

In Example I heat lag is accounted for by figuring all exposures shaded excepting the roof and the south wall. On shaded exposures the full temperature difference is used in the calculations although additional lag may be present. No additional factor of safety need be added, when calculating in this manner, except in unusual cases as the factors used are sufficiently safe to cover reasonable duct and apparatus losses.

The "dry tons" (sensible load) and "moisture tons" (latent load) should be calculated separately, as indicated in Example I. Having calculated these quantities, the next step is to determine the sensible-latent heat ratio h_s/h_l, which is merely the total dry tons divided by the total moisture tons. The total refrigeration load is the sum of the dry tons and the moisture tons.

Equipment Selection[7]

6. **Surface cooling.**[8] Before the cooling equipment is selected to meet the calculated load requirements, the supply air, outside air and return air quantities must be decided upon in accordance with methods already described. In selecting surface cooling coils, the governing factors are the entering air dry-bulb and wet-bulb temperatures (which determine the relative humidity), and the evaporator temperature. Having arrived at the air quantities, it is merely a matter of arithmetic to determine the entering air conditions by ref-

Example I. Human Comfort Air Conditioning Calculations

	Winter db	Summer db	Summer wb	E.T.	rh	dp	gr/lb	Btu/lb	cu ft/lb
Outside	35	90	70	—	37%	60	77.29	34.04	14.12
Inside	70	80	66	73.5	48%	58	71.86	30.79	13.84

Department store, Los Angeles, California. Type of room—First floor sales area with lunch counter
Room volume (Length 169½ ft × width 103½ ft × height 15¼ ft = 272,000 cu ft

Sensible Load (Dry Tons)	Cooling				Heating		
Heat Leakage	Net sq ft of surface	Heat transfer factor U	Temp diff F	Btu per hr	Heat transfer factor U	Temp diff F	Btu per hr
Walls or portions of walls exposed to outside							
North	—	—	—	—	—	—	—
South-shaded	—	—	—	—	—	—	—
South-unshaded	950	0.30	40	11,400	0.30	35	9,975
East	1,300	0.30	10	3,900	0.30	35	13,650
West-shaded	570	0.30	10	1,710	0.30	35	5,985
West-unshaded	—	—	—	—	—	—	—
Walls or portions of walls directly in contact with ground	—	—	—	—	—	—	—
Glass in outside walls							
North	—	—	—	—	—	—	—
South-shaded (entrance doors)	240	1.13	10	2,712	1.13	35	9,500
South-unshaded (show windows and backing)	1,435	0.45	40	25,830	0.45	35	22,600
East	305	1.13	10	3,447	1.13	35	12,100
West-shaded (entrance doors)	120	1.13	10	1,356	1.13	35	4,750
West-shaded (show windows and backing)	915	0.45	10	4,118	0.45	35	14,400
Interior walls or portions of walls next to unconditioned areas							
North	2,625	0.25	5	3,281	0.25	0	—
Ceilings or portions of ceilings							
Under flat roof	17,550	0.10	60	105,300	0.10	35	61,425
Under attic	—	—	—	—	—	—	—
Under fin. rooms	—	—	—	—	—	—	—
Skylights	—	—	—	—	—	—	—
Floor or portions of floor							
Over fin. rooms	—	—	—	—	—	—	—
Over basement	17,550	0.35	5	30,713	0.35	25	153,565
Over ground	—	—	—	—	—	—	—
Sub-total				193,767			307,950

Internal Heat	Unit	No. of units	Cooling Factor	Cooling	Heating
Outside air — As infiltration and/or air through conditioner	cfm	11,700	$\frac{(0.24)(60)(10°)}{13.84}$	121,660	$\frac{(0.24)(60)(35°)}{13.47}$ = 437,580
People (normal no. of people)	person	1,170	220	257,400	Add to heating duty for latent load:
Lights (watts on sunny day)	watts	87,750	3.4	298,350	
Other sources of heat					Heat of vaporization for humidifying — lb water per hr × 1,060 = None × 1,060 = 0
1. Coffee urn—steam (hooded)	gal	3	(50%) (800)	1,200	
2. Steam tables electric (hooded)	sq ft	4	(50%) (200)	400	
3. Motors—supply fan	hp	15	2,545	38,175	
4. Motors—three ⅓ hp = 1 hp total	watts	1,250	3.4	4,250	
5. Electrical apparatus (hooded)	watts	27,650	(50%) (3.4)	47,000	
6. Gas apparatus (hooded)	small toaster	2	(50%) (5,750)	5,750	
			Total sensible cooling load = 967,952		Gross heating load 745,530
			Total sensible load, tons $= \frac{967,952}{12,000} = 80.7$		Minus lights 298,350
					Net heating load 447,180

Example I. Human Comfort Air Conditioning Calculations. (Continued)

Latent load (Moisture tons)	Cooling				Cooling load summation
Internal Heat	Unit	No. of Units	Factor	Btu per hr	
Outside air — As infiltration and or air through conditioner	cfm	11,700	$\frac{(1,060)\ (60)\ (5.43\ Gr)}{(13.84)\ (7,000)}$	41,650	Total sensible heat load, tons 80.7
People	person	1,170	180	210,600	Total latent heat load, tons 21.2
Other sources of moisture					Total load, tons 101.9
1. Coffee urns steam (hooded)	gal	3	(50%) (800)	1,200	Sensible to latent heat ratio $= \frac{h_s}{h_l}$
2. Steam tables—electric (hooded)	sq ft	4	(50%) (350)	1,700	
3. Gas apparatus (hooded)	small toaster	2	(50%) (830)	830	$= \frac{80.7}{21.2} = 3.8$
	Total latent heat load, Btu per hr			254,980	
	Total tons latent heat load $= \frac{254,980}{12,000} = 21.2$				

erence to the tables of air properties. Surface cooling coil tonnage ratings are found from manufacturers' tables at various entering air conditions, evaporator temperatures and sensible-latent heat ratios, h_s/h_l. It is usually necessary to refer to the psychrometric chart, Fig. 2, at some time during the process of selection. It is also often necessary to vary the air quantities (particularly the supply air quantity), in order to obtain an economical coil selection. **Example II** illustrates typical calculations for the selection of surface cooling coils for the cooling load calculated in Example I. (Conclusions based on a coil manufacturer's data.)

Air washer cooling.[9] When using an air washer for dehumidifying, the governing factors in the selection are the entering air wet-bulb temperature, and the required leaving air dew-point temperature. The supply, outside and return air quantities are determined in the same general manner as for surface cooling, but in addition the dehumidified air quantity must be calculated. Only sufficient air is dehumidified, by going through the air washer, to balance the total sensible load. The dehumidifier leaving air dew point is determined from the latent load requirements.

For most comfort cooling applications, the dehumidified air quantity will be the supply air quantity, as it is unusual for the dehumidifier leaving air temperature to be more than 30 deg below the required room temperature. Where the supply air quantity, or "room diffusion" (difference between room and supply air temperature), is definitely fixed by specifications, or in cases of unusual air distribution problems requiring a low room diffusion, it may be necessary to bypass a portion of the supply air around the air washer. Most air washers are provided with a bypass having automatic dampers for control, but the continuous bypassing of a minimum amount (as in **Example III**) is not ordinarily required. However, Example III has been figured with the same air quantities as Example II for purposes of comparison between surface cooling and dehumidifier calculations.

As in the case of surface cooling, the psychrometric chart, and the sensible-latent heat ratio, h_s/h_l, are used to determine the dehumidifier leaving air temperature (or dew point in a saturating type washer, as is generally used for comfort air conditioning). Example III illustrates the calculations for the selection of an air washer for the cooling load calculated in Example 1. The dehumidifier leaving air temperature, as obtained by using the sensible-latent heat ratio on the psychrometric chart, should always be checked arithmetically to see that it is sufficiently low to balance the

calculated latent heat load. Using moisture values from the tables at the entering and leaving air dew points, the actual moisture removal and latent heat equivalent thereof may be calculated as indicated in Example III. If the moisture removal is not enough to balance the latent heat load, a lower leaving air dew point will be required and the dehumidified air quantity adjusted accordingly. It is also necessary, when using an air washer, to increase the total refrigeration lead by the heat equivalent of the horsepower of the pump used to circulate the spray water.

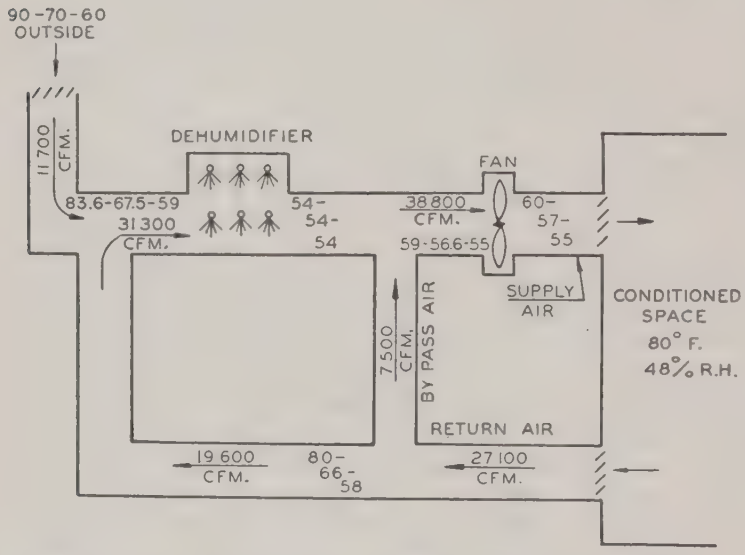

Fig. 5. Dehumidifier Cycle, Showing Dry-Bulb, Wet-Bulb and Dew-Point Temperatures at Various Points (Example III)

Conclusion

7. Cooling load calculations are very similar to heating calculations, but are considerably more complicated in that the air conditioning engineer must be thoroughly familiar with the following special circumstances which affect the design[10]:

1. Inside design conditions for comfort
2. Outside design conditions
3. Sunlight heat gain and heat lag
4. Ventilation requirements
5. Air properties and the psychrometric chart

On a large building[11] where the solar radiant heat gain forms a large percentage of the cooling load, the air distributing system should be divided into sections (called zones) so that the cooling may be varied in accordance with the movement of the sun effect across the building. Preferably, each exposure should be zoned separately, but north and east exposures may be put on one zone, and south and west exposures on another. The radiant heat load should be figured for each zone to determine the maximum air requirements for the zone. However, when calculating the total refrigeration load for the building, only the radiant load on the roof and one exposure need be included as the maximum solar heat gain does not occur simultaneously on any two exposures.

On all fan systems there will be some loss of air in the distributing ducts. For this reason fans should be selected for about 5% greater capacity than the calculations actually show, depending, of course, upon the workmanship with which the duct systems are installed. For air conditioning work great care should be taken to insure tightness of the duct work, as excessive loss of refrigerated air is very expensive.

The field for air conditioning is still in a state of constant expansion, and much research work is in progress. World War II gave us many new applications for air conditioning, advanced engineering data and knowledge, and consequent improvements and refinements in methods of calculating air conditioning problems and in selecting equipment. New and interesting data can be found in current trade literature and technical journals. Such information is very helpful to the air conditioning engineer in keeping up to date on the changes and improvements in air conditioning apparatus, and on new and simplified methods for approaching the air conditioning problem.

Example II—Surface Cooling Calculations

Outside air quantity
$= 1{,}170$ people $\times$ 10 cfm $= 11{,}700$ cfm

$$= \frac{11{,}700}{13.84} = 845 \text{ lb per min}$$

Supply air quantity:
Internal sensible load
= total sensible − (fan motor and outside air heat gain)
$= 967{,}952 - (38{,}175 + 121{,}660)$
$= 967{,}952 - 159{,}835 = 808{,}117$ Btu per hr
Supply air 20° below room temperature
$= 80° - 20° = 60$ F
Supply air quantity

$$= \frac{(808{,}117)(13.84)}{(0.24)(20°)(60)} = 38{,}800 \text{ cfm}$$

$$= \frac{38{,}800}{13.84} = 2{,}800 \text{ lb per min}$$

Return air quantity

$$= 38{,}800 - 11{,}700 = 27{,}100 \text{ cfm}$$

$$= \frac{27{,}100}{13.84} = 1{,}955 \text{ lb per min}$$

Entering air temp, db

$$= \frac{(1{,}955)(80) + (845)(90)}{2{,}800} = 83°$$

Entering air, total heat

$$= \frac{(1{,}955)(30.79) + (845)(34.04)}{2{,}800} = 31.8$$

$\therefore$ Entering air wb $= 67.2°$
Entering air rh $= 44\%$ (from Fig. 2)

Locate the above entering air conditions on the psychrometric chart, determining point A. This point should fall on the straight line between point B, the outside air condition of 90 db and 70 wb and the room condition of 80 db and 48% rh at C. Using the sensible-latent heat ratio $h_s/h_l = 3.8$, construct line XY on the psychrometric chart from the reference point of 60° and 0 grains to 3.8 on the h_s/h_l scale. Draw another line ZA parallel to the first but going through the entering air condition of 83 db and 44% rh. The point where this second line intersects the saturation line at Z, determines the coil surface temperature (or the dew point if a washer is used), in this case 54°.

Leaving air db = Entering air db −

$$\frac{\text{Total sensible load in Btu per hr}}{(0.24)(\text{lb air per min})(60)}$$

$$= 83° - \frac{(967{,}952)}{(0.24)(2{,}800)(60)}$$

$$= 83° - 24° = 59°$$

The required leaving air condition will fall on the line ZA on the psychrometric chart. Therefore, refer to point D on Fig. 2 at 59 F leaving db temperature and find 56.5 F wb and 55.0 F dp as the air conditions leaving the surface cooling coils. One may now select direct expansion cooling coil from catalog. (Aerofin bulletin D. E. 48–1, pp. 12–27 incl.)

Assume 500 fpm face velocity over coil

$$\text{Required face area} = \frac{38{,}800}{500} = 77.6 \text{ sq ft}$$

Use 6 coils − 29-in. casing − 6 ft 0-in. tubes
= (6 coils) 12.4 sq ft ea. = 74.4 sq ft face area

$$\therefore \text{ actual face velocity} = \frac{38{,}800}{72} = 540 \text{ fpm}$$

t_c = coil temperature = 54°
t_r = refrigerant temperature
h_c = total heat corresponding to t_c
= 22.59 Btu per lb
h_a = total heat at entering wb, or 31.8 Btu per lb
Total Btu per hr $= (L)(\text{cfm})(h_a - h_c)$;
$(12{,}000)(101.9) = L(38{,}800)(31.8 - 22.59)$

$$L = \frac{(101.9)(12{,}000)}{(38{,}800(9.21)} = 3.42$$

4-row coil, $L = 3.47$ (from manufacturers' tables)
$\therefore$ Use 4-row coil, 8 fins per inch $M = 1{,}600$ (from manufacturers' tables)
Total Btu per hr $= (M)(\text{face area})(t_c - t_r)$
$(101.9 \text{ tons})(12{,}000) = (1{,}600)(72)(54 - t_r)$
$101.9 = 519 - 9.6t_r$

$$t_c = \frac{417.1}{9.6} = 43.5\text{F}$$

Unless the compressor is close to the coils there will be an appreciable suction line pressure drop, resulting in a lower refrigerant temperature at the compressor than at the cooling coils. In this case assume suction line of such size and length that the

line loss is equivalent to a 1 degree drop in suction temperature between coil and refrigerating unit.

Select Freon-12 refrigerating unit for 101.9 tons refrigeration at 42.5° suction temperature.

Summary of equipment selection:
Condensing system—101.9 tons refrigeration at 42.5F suction temp
Evaporator—6 No. 84 Aerofin coils—29-in. casing—6-ft, 0-in. tubes at 43.5 F suction temp, total face area 74.4 sq ft
Supply fan—No. 10 S.W.S.I. = 41,000 cfm at 2,100 fpm outlet velocity.

Example III—Dehumidifier Calculations

Outside air = 11,700 cfm = 845 lb per min
Supply air = 38,800 cfm = 2,800 lb per min
Return air = 27,100 cfm = 1,955 lb per min
Supply air dew point = 54° (See Example II)

It is impossible to obtain complete saturation in an air washer when dehumidifying. However, for practical purposes it is sufficiently accurate to assume 100% saturation for the dehumidifier calculations, in which case the dehumidifier leaving db = wb = dp = 54F.

Dehumidified air quantity

$$= \frac{(\text{Internal sensible} + \text{fan motor heat})}{(60)(0.24)(80° - 54°)}$$

$$= \frac{(808{,}117 + 38{,}175)}{(60)(0.24)(26°)}$$

$$= \frac{846{,}292}{(60)(0.24)(26)} = 2{,}260 \text{ lb per min}$$

$$= (2{,}260)(13.84) = 31{,}300 \text{ cfm}$$

Bypass air = 38,800 − 31,300 = 7,500 cfm
All of the outside air should be dehumidified
Return air through dehumidifier = 2,260 − 845 = 1,415 per min
Entering air to dehumidifier:

$$\text{db} = \frac{(1{,}415)(80) + (845)(90)}{2{,}260} = 83.6\text{F}$$

$$\text{Total heat} = \frac{(1{,}415)(30.79) + (845)(34.04)}{2{,}260}$$

= 32.04 Btu per lb
wb = 67.5F (from Table 1, Chap. 10)
dp = 59F = 74.54 gr per lb from chart of Chap. 10

	gr per lb
Moisture content at 59F entering dp	= 74.54
Moisture content at 54F leaving dp	= 62.01
Moisture removed in dehumidifier	= 12.53

Total moisture removed

$$= \frac{(60)(2{,}260 \text{ lb})(12.53 \text{ gr})}{7{,}000 \text{ gr per lb}} = 243 \text{ lb per hr}$$

Latent heat equivalent
= (243 lb)(1,060) = 257,580 Btu per hr
Calculated latent load
(Example I) = 254,980 Btu per hr

	Btu per lb
Total heat at 70F outside wb	= 34.04
Total heat at 54F dehumidifier wb	= 22.59
Heat removed from outside air	= 11.45
Total heat at 66F room wb	= 30.79
Total heat at 54F dehumidifier wb	= 22.59
Heat removed from return air	= 8.20

	Btu per min	tons
Total heat removed from outside air = (845)(11.45) =	9,675	
Total heat removed from return air = (1,415)(8.20) =	11,603	
Cooling load (sub-total)	21,278	= 106.4
Allow 10 hp for spray pump = 10 × 42.5	= 425	= 2.1
Total cooling load, tons	= 21,703	= 108.5

Select dehumidifier for 500 to 750 fpm face velocity

$$\frac{31{,}300 \text{ cfm dehumidified}}{600 \text{ fpm face velocity}} = 52.2 \text{ sq ft face area}$$

Allow 7° rise in spray water
Spray water required

$$= \frac{21{,}703 \text{ Btu per min}}{7 \times 8.33} = 372 \text{ gpm}$$

$$\frac{372 \text{ gpm}}{52.2 \text{ sq ft}} = 7.1 \text{ gpm per sq ft face area}$$

Using 2 spray banks

$$= \frac{7.1}{2} = 3.55 \text{ gpm per sq ft per spray bank}$$

54F dp − 2° difference = 52F leaving water
52F − 7° range = 45F entering water.

Difference between leaving air and leaving water depends upon dehumidifier type and design values given are typical.

Thus one selects refrigerating unit and water cooler for 108.5 tons refrigeration when cooling 372 gpm spray water from 52F to 45F, using 2-bank air washer for 31,300 cfm.

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38. AIR SYSTEMS

1. The performance requirements of a distribution system are as follows:

a. To introduce to the conditioned space a quantity of air at a temperature and humidity which will maintain the desired indoor temperature and humidity.

b. To distribute the air so that:
The temperature throughout the occupied zone is essentially uniform.
The temperature difference between floor and breathing line is minimized.
Motion of air will be adequate but not so great as to produce a sensation of draftiness.

c. To introduce the air as quietly as possible, so that noise from the air conditioning system will not be objectionable.

In addition the distribution system should be designed so as to blend with architectural treatment of conditioned space, be economical to install, and result in minimum owning and operating expense. **Fig. 1** shows typical situations.

2. Total air changes is defined as the number of complete changes per hour of the air in the room, giving the rate of circulation equal to the cfm discharged through the supply openings. This factor must not be confused with air changes equivalent to the rate of supply of outside fresh air. The total air changes of a given space

$$N = 60\,Q/V \qquad (1)$$

Where

N = Total air changes per hour
Q = cfm of air, referred to the specific volume at design conditions, equivalent to the total air discharged from the supply openings
V = Volume of conditioned space, cu ft

$$Q = NV/60 \qquad (2)$$

Uniformity of motion and temperature distribution are particularly dependent upon the relation of total air circulation to the volume of the conditioned space. With a small number of supply openings and a low total of air changes, there may be a

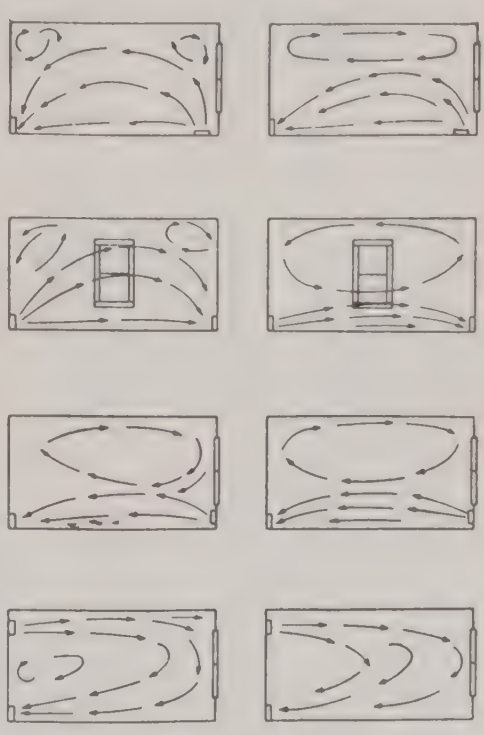


Fig. 1. Vertical Sections of Small Rooms Showing Types of Circulations

tendency toward dead air pockets. On the other hand a high number of changes may mean drafts.

Table 1 may be used in estimating the **quantity** of air supply, or in determining whether a given rate of supply will give satisfactory air distribution. Judgment must be exercised in using this table, as it sets up wide limits.

If the number of air changes is approximately halfway between the upper and lower limits given in Table 1, no particular difficulties should be encountered. Rates

of air changes within the lower portions of the ranges given are more suitable if the heat loss or heat gain per unit volume is low than if the heat loss or gain per unit volume is high.

3. Air changes in excess of 12 per hour generally are suitable only when the heat gain per unit volume is very high. In such cases, every possible effort should be made to reduce the **internal sensible heat gain** by insulation, awnings, use of mechanical ventilation, Venetian blinds, etc. When it is not possible to do this, there should be a compromise between the number of air changes and diffusion temperature difference. The **temperature difference** is the temperature difference between the air temperature at supply opening and design indoor temperature. This diffusion temperature difference should be increased if possible. Relocating or increasing the number or decreasing the size of outlets may permit this.

When the number of air changes is very high, the air should be directed toward unoccupied spaces, and induced secondary circulation should be avoided. With 12 or more changes per hour the air in the room is pretty certain to be well stirred up, and more effort should be directed toward avoiding drafts than toward supplying the conditioned air to the exact sources of heat gain. In all types of applications with very **high ceilings** the number of air changes should be based on the volume from the **floor level** to approximately six feet above the level of the supply openings, neglecting the remaining volume. Unless the number of air changes is very high this upper portion of the space will remain relatively undisturbed.

4. The **temperature** and **quantity** of air introduced to the conditioned space are dependent on each other, with heat loss or heat gain as the relating factor. The formula for sensible heat is

$$h_s = \frac{60Q}{v} \times 0.24 \times \text{dtd} \qquad (3)$$

From this we get the following formulae for air quantity required to accomplish a given heating or cooling load.

$$Q = \frac{h_s}{1.06 \text{ dtd}} \text{ for cooling} \qquad (4)$$

$$Q = \frac{h_s}{1.08 \text{ dtd}} \text{ for heating} \qquad (5)$$

where

Q = Total air supply, cfm referred to design indoor conditions

dtd = Diffusion temperature difference, deg F

h_s = Internal sensible heat gain for cooling, and internal sensible heat loss for heating, Btu per hr

v = specific volume of air at design outdoor conditions, cu ft per lb

Internal heat loss or heat gain refers to

Table 1. Approximate Rates of Air Supply for Heating and Cooling

Type of application	Location of supply openings	Approximate equivalent air changes per hour	
		Heating	Cooling
Residences	In baseboard near floor, discharging horizontally	3–6	—
	In floor, discharging vertically upward	3–6	6–9
	High in wall, discharging horizontally	5–9	6–9
Offices, stores, and restaurants	Generally high in wall, discharging horizontally	5–8	6–12
Theaters, assembly halls, dance halls, etc.	Generally high in wall, discharging horizontally	5–10	6–12

the heat loss or heat gain exclusive of that of ventilation air which is mixed with recirculated air before entering the air conditioner. In the above formulae the constants 1.06 and 1.08 are determined so that Q is the cfm based on the specific volume of air at design indoor conditions. Values corresponding to 80 F, 50% relative humidity for summer and 70 F, 35% relative humidity for winter, are used.

5. In order to obtain the maximum diffusion temperature difference in the above formulae for cooling, refer to **Table 2.** It

Table 2. Approximate Maximum Diffusion Temperature Differences Permissible for Various Heights of Supply from Floor (for Cooling)

Height of supply above floor, ft	Maximum diffusion temperature difference, deg F	
	Relatively few large volume discharges	Relatively many small volume discharges
10	15	20
12	17	22
14	19	24
16	21	26
18	23	28
20	25	30

is evident that greater diffusion temperature differences are allowable when many discharges are used than with few. Again, as the height of the discharge increases, the allowable difference increases.

If the temperature difference between room air and supply air is increased, the air stream bends down more steeply and tends to enter the occupied zone a shorter distance from the point of supply. Hence, for any given height of grille there is a **minimum height of supply** above the floor for a given discharge velocity and type of grille. If the grille is located at lower level, air will enter the occupied zone at an undesirably high velocity and possibly at too low a temperature.

The diffusion temperature difference, on heating, should be kept as low as possible. However, if the discharge temperature is less than about 100 F, great care should be taken to prevent this air from blowing directly over an occupant. Usually 100 F is as low as warm air should be discharged into a room. Air at lower temperatures seems cold to the average person if the velocity is greater than 40 to 50 fpm.

To summarize, in designing air distribution systems it is necessary to consider both diffusion temperature difference and total air changes in relation to the type and location of supply openings. The total air changes must be adequate to give good distribution and at the same time the temperature difference must be low enough so the air stream will not enter the occupied zone before its temperature has been raised essentially to that in the room or before its velocity has been reduced to approximately 50 fpm.

Gravity Flow of Air

6. When air is heated it tends to rise and when cooled it tends to fall. Supply and return **openings** should be located with due consideration for the natural gravity flow characteristics of warm and cool air. For **heating,** this would indicate that warm air supply openings should be located near the floor blowing horizontally, or near the ceiling blowing slightly downward. Similarly for **cooling,** the air should be introduced horizontally through openings well above the floor level, or vertically upward through openings near or in the floor.

If a given cooling load is to be handled with a relatively small quantity of air, the diffusion temperature difference will be large, and care must be exercised to prevent cold air from falling too quickly to a level where it can produce a sensation of draftiness. The cold air must be warmed by mixing action, which may be promoted by:

a. Locating the supply openings so that the supply air is in contact with room air for a longer time before reaching the occupied zone. This may be accomplished by high outlets, high discharge velocities, or directing the air slightly upward.
b. Breaking up the supply air into a number of relatively small jets so that mixing takes place rapidly. For example, numerous small-volume, well-distributed outlets or long, nar-

row, slot-type outlets with directional flow.

c. Use of special grilles such as aspirating, directional flow, etc.

7. Air may be **projected** to a desired point by giving it adequate velocity and proper direction at point of supply, but this method of control of air motion is not applicable in the case of **return openings.** Tests have shown that even with a face velocity of 1,000 fpm the air velocity is not more than 50 at distances more than 3 ft from a return grille. Thus returns must be located so that air will flow to them freely by gravity.

Returns for heating should be located so that air does not travel a great distance nor encounter a number of bends. Cold air comes from the point where heat loss occurs, such as windows and outside walls. In cooling the location of returns is less vital. In theory they should be located well above the floor but in practice gradients in cooling are not so great as in heating.

The **number of returns** is variable. Relatively few are sometimes adequate in both heating and cooling systems, but in cases of highly concentrated loads or bad exposure, many well-distributed returns are often required.

8. As a jet of air issues from an opening it encounters **resistance** to flow, partially by friction between the air in the sides of the jet and the air in the room. The amount of room air thus **entrained** increases with the perimeter and velocity of the jet. At relatively high initial velocities (800 fpm or more) the amount of entrained room air may be great enough to produce a very appreciable secondary or induced circulation. This entrainment of room air may be increased, at a given velocity, by using grilles specially designed to increase turbulence in the air jet **(aspirating grilles).** On the other hand, the ratio of perimeter to area may be kept small, to reduce the amount of entrained room air and thus to maintain the jet velocity.

The secondary circulation induced by the entrainment of room air is often a great aid in maintaining uniform air motion and temperature distribution, with a minimum number of supply openings and with a minimum quantity of air. The entrained room air also tempers the supply air, reducing the tendency for cool air to fall to the floor and room air to rise to ceiling.

Grille Characteristics

9. Air flow characteristics of different types and arrangements of supply openings and grilles must be considered in conjunction with the principles discussed above. The important characteristics to be considered may be enumerated as follows:

a. **Throw** vs. proportions and arrangement of the grille, and discharge velocity. This determines the distance conditioned air may be projected from the point of supply.

b. Horizontal **spread** of the air vs. proportions of the grille. This affects the number of openings required to cover a given area.

c. Air **velocity** at the lower edge of the air stream vs. proportions and arrangements of the grille, diffusion temperature difference, discharge velocity, horizontal distance from grille and vertical distance from grille level. Cool air discharged from a grille drops, causing the air stream issuing from a supply opening to describe a path which bends downward. Furthermore, the velocity of the air at the boundaries of the air stream is lower than the average velocity. It is important that the air velocity should not be in excess of 40 to 50 fpm where the air stream enters the occupied zone.

d. Vertical **fall** or rise of air from the supply opening to where the jet temperature is essentially equal to that in the room vs. proportions and arrangement of the grille, diffusion temperature difference and discharge velocity. This is useful in determining whether a given arrangement will spill cold air into the occupied zone in summer, or cause stratification of warm air at the ceiling in heating; hence it is practicable in designing a system to avoid these two undesirable effects.

e. **Quantity** of **entrained** room air as a

percentage of the initial jet volume vs. proportions and arrangement of the grille, and discharging velocity.

Data concerning the above characteristics are available in varying degrees of completeness from grille manufacturers.

10. The following conclusions on air distribution are based upon experimental work conducted by one well known grille manufacturer:

a. The throw from a straight flow grille varies with the square root of the daylight area of the grille and with the face velocity within ordinary grille proportions.
b. The aspect ratio of a grille has no appreciable effect on the distance of air throw.
c. Fanning out the air stream shortens the throw, the amount depending on the degree of deflation.
d. The drop, for a given throw, of an air stream below room temperature, varies about inversely as the face velocity and directly as the temperature differential.

11. Tables 3 and 4 may be used as guides in estimating discharge velocities for bar-type or lattice-type grille without deflectors, and for plain nozzles. These tables are approximate, and intended only for use in the general preliminary layout. The performance of numerous special types of grilles may differ appreciably from these tables.

When cooling with floor openings discharging vertically, the velocity depends upon the ceiling height. A velocity of at least 500 fpm should be used for average ceiling heights of 7 to 8 ft. For higher ceilings the velocity should be increased somewhat. When heating with floor openings discharging vertically, the velocity should be kept low, not over 400 fpm if possible. Velocities of approximately 500 fpm should be used only when the system is to be used for both heating and cooling, with the same air quantity the year 'round.

Table 3. Approximate Velocities for Various Diffusion Temperature Differences and Lengths of Throw

Throw from point of supply, ft	Approx velocity over face area, fpm			
	Diffusion temperature difference, F			
	15	20	25	30
15	400	525	650	775
25	500	625	750	875
35	600	725	850	975
50	750	875	1000	1125
75	1000	1125	1250	1375
100	1250	1375	1500	1625

Table 4. Approximate Supply and Return Air Velocities

Type of opening	Average velocity through free area of grille, fpm	
	Heating	Cooling
Supply—Low in wall, discharging horizontally	300–400	—
Supply—In floor, discharging horizontally	300–400	500–800
Supply—High in wall, discharging horizontally	600–1,500*	300–1500*
Return	300–500	300–500

* Note: Higher velocities are sometimes permissible with some types of supply grilles, and when a long throw of the air is required.

Return velocities given in Table 4 represent average practice for residential applications, where it is usually necessary to do everything possible to keep the resistance of the duct system low. However, in commercial applications higher velocities (up to 1,000 fpm) may be used.

Certain precautions must be observed in installing high-velocity supply systems. There should be no obstructions such as columns or chandeliers in the direct path of the air jet. If the ceiling has cross beams perpendicular to the direction of the air jet, the supply openings should be located at least 2 ft below the lowest point of the ceiling.

Location of Openings in Relation to Exposure

12. A considerable percentage of the conditioned air should be supplied to that portion of the space where heat gain or heat loss occurs. For heating the best way to accomplish this is to discharge the warm air across the exposed walls. In cooling the problem is more one of a fairly uniform

distribution of supply air throughout the conditioned space. The following summary, coupled with experience and judgment, may be used as a guide in locating supply and return openings. In general, supply openings are to be located to minimize the possibility of blowing air directly on occupants; return openings near unusual sources of odor concentration should be avoided; exhaust vents to outdoors are advisable where there are sources of exceptional odor concentrations; supply risers should be placed in inside partition walls where possible to avoid excessive heat loss in duct and low discharge temperature.

With respect to supply openings in cooling systems, the following requirements apply:

a. **High in wall blowing horizontally:** This arrangement has proved most satisfactory where air is discharged at velocities of 500 fpm or more. Care should be exercised in using outlets near ceiling to prevent streaking. Supply outlets should be about 7 ft from floor and have air directed horizontally.
b. **Ceiling outlets** with diffusers or a deflecting pan to direct the air horizontally are very satisfactory. In places where ceilings are very low, ceiling outlets can be used to advantage to avoid drafts.
c. **Floor openings along outside wall blowing upward** give reasonably satisfactory results, with discharge velocity of 500 fpm or more, especially in residences, as a compromise arrangement for summer and winter operation. The air should be deflected at a slight angle from the vertical, to prevent the air from mushrooming, i.e., falling directly down on itself.
d. **Baseboard supply** is unsatisfactory for cooling, if air is directed horizontally. There are baseboard registers now made that are adjustable, such that for cooling operation the face damper is closed and the top of the grille pulled out so that the air is discharged vertically.

13. In large commercial jobs it is almost always necessary to have some form of **zone control** in order to maintain even temperatures in the different conditioned spaces, because of different and changing load conditions for both heating and cooling. Some of the most common methods of zoning are as follows:

a. Separate conditioner, fan, and duct system for each zone.
b. Central conditioner and fan with separate branch duct systems for each zone, controlled by automatic dampers in each zone.
c. Central conditioner with blow-through fan, where zones are divided at the coils in the conditioner with separate duct systems in each zone. This type of system usually has automatic control of the individual coils in each zone (see p. 650).
d. Central conditioner and fan for entire system, with booster coils in each zone duct system automatically controlled from each zone.
e. Central conditioner and fan for entire system with booster fans for each zone automatically controlling the air flow to each zone. Sometimes this type of zone control uses booster coils in each zone.

Air Filters

14. **Air cleaning** is the removal of dust and dirt from the air, which may be accomplished in several ways:

a. By passing the air over a surface coated with an adhesive material which literally "catches" dust particles carried by the air.
b. By passing the air through a finely woven fabric, which "screens" dust particles from the air.
c. By changing the direction of the air abruptly, thus "throwing" the dust particles out of the air stream.
d. By passing the air through a water spray. The water droplets collect on dust particles and cause them to drop down out of the air stream.
e. By passing the air between charged electrodes, which induce electrical charges on the dust particles and cause them to be precipitated from the air stream.

Each one of the above methods may be

used singly, or in combination, in various applications.

Viscous air filters employ the first method listed above. In the unit the filtering material is arranged in units of such size as to be convenient for installing and servicing. Each unit is a pad of filter material (such as spunglass wool, steel wool or animal hair), and the necessary supporting frame, arranged so that it can be removed easily for cleaning or replacing. Unit filters are made of hemp or paper fiber to be thrown away after using.

The **automatic type** of viscous filter uses the same principle of filtering as the unit type, but the filter is kept clean automatically. One method of accomplishing this is to support the filter material on an endless belt which is moved slowly, or intermittently, through a bath of the fluid.

Dry air filters make use of the second method of air cleaning listed above. Such filters are always of the unit type, and may be either cleanable or throw-away, more frequently the latter.

Scrubbers are more effective as filters than air washers. They make use of the third method of air cleaning listed above. The zig-zag path of flow throws the air against the baffle plates. Some of the dust particles are thus collected on the wetted surfaces and flushed to the collecting basin. The scrubber has the further advantage over the air washer in that the temperature and humidity of the air are not greatly altered.

15. Air washers and humidifiers may be classified in two distinct types; (a) the spray type and (b) the surface type. Dehumidifiers may be classified in three types; (1) spray, (2) surface, (3) chemical dehydration types.

Air washers are usually of the spray type, and if properly designed and adapted may serve as a humidifier, dehumidifier, or both. The spray-type washers commonly used for air conditioning and ventilation are made in standard designs and sizes to meet the requirements of a given duty. The classification of design is usually expressed in the number of banks of spray nozzles, of which there are never more than two in a standard unit.

The two standard designs are **one-bank** and **two-bank** washers. The banks of nozzles referred to are spray nozzles, which spray the water in fine particles and distribute the spray over the entire cross section of the washer chamber. All washers are usually equipped with one extra row of flooding nozzles which spray a sheet of water down the entering side of the leaving eliminator plates to wash down any dirt and dust which may have accumulated and to help remove some of the entrained moisture being carried through the washer by the air.

Tables 5 and **6** give typical specifications for one-bank and two-bank washers, devices arranged with flooding nozzles to wash the eliminator plates. A one-bank washer is satisfactory for humidifying and washing the air, but is not adequate for cooling or dehumidifying.

The quantity of water sprayed in a standard one-bank washer varies from 4 to 5 gal per 1,000 cu ft of air based on an air velocity of 500 fpm through the washer chamber. The flooding nozzles spray 3 to 4 gal of water per min per ft of washer width.

The **saturation efficiency,** or the reduction of the entering wet-bulb depression, of a one-bank washer ranges from 60 to 70%, depending upon the fineness of the spray, distance between the spray and eliminators, and velocity of air through the washer chamber.

The spray nozzles in a two-bank washer spray the same quantity of water as those in a one-bank washer, but as there are twice as many, the quantity of water sprayed would range from 8 to 10 gal per 1,000 cu ft of air based on an air velocity of 500 fpm through the washer chamber. The saturation efficiency of a two-bank washer ranges from 85 to 95%.

The **resistance** to air flow of either the one-bank or two-bank washer ranges from 0.2 to 0.4 in. of water, depending upon the spacing and construction of the eliminators, type of entering baffles, direction of sprays (either with or against the direction of air flow), and the connections to and from the washer which may vary the air velocity across the face of the washer and thus cause eddy currents in the spray chamber.

Details of **construction** of air washers are practically the same for all makes of stand-

Table 5. Typical Specifications of One-Bank Washers

Number	Air capacity, cfm	Chamber area, sq ft	Dimensions, in. W	H	L	Washing surface, sq ft	Number of nozzles Spray	Flooding	Water flow, gpm Spray	Flooding	Pump size, in.	Motor hp
1	4,600	9.2	33	61½	86⅞	84.5	12	9	22	9	1½	1½
2	9,400	18.8	64½	61½	86⅞	174	24	20	43	20	1½	2
3	14,200	28.3	95½	61½	86⅞	263	36	30	65	30	1½	3
4	21,900	43.7	95½	61½	86⅞	413	60	30	108	30	1½	3
5	29,300	58.6	127½	61½	86⅞	553	80	40	144	40	3	5
6	36,700	73.4	158½	61½	86⅞	693	100	51	180	51	1½	5
7	44,100	88.2	190	61½	86⅞	852	120	61	216	61	1½	5

Table 6. Typical Specifications of Two-Bank Washers

Nominal size, ft	Air capacity at 500 fpm, cfm	Area of spray chamber, sq ft	Dimensions, ft-in. W	H	Elimination surface, sq ft	Spray nozzles, no.	gpm	Flooding nozzles, no.	gpm
5 ×4	10,000	20	5-3	5-1	244	80	9	18	98
5 ×6	15,000	30	5-3	7-1	366	120	9	18	138
5 ×8	20,000	40	5-3	9-1	488	160	9	18	178
5 ×9	22,500	45	5-3	10-1	549	180	9	18	198
6 ×5	15,000	30	6-3	6-1	366	120	11	22	142
6 ×7	21,000	42	6-3	8-1	512	168	11	22	190
6 ×9	27,000	54	6-3	10-1	658	216	11	22	238

ard air washers, the only difference being in the type and capacity of nozzles, entering baffles, and detail of design of the leaving eliminators.

The **casing** is constructed of either 18-gage galvanized steel or 30-oz copper, rigidly braced with galvanized iron or brass angles to prevent sagging and "breathing" of the casing. All angle bracing and door frames are riveted with rivets soldered. All seams are soldered, and riveted with rivets soldered. The **tank** is constructed of 14- to 16-gage galvanized steel or 36- to 40-oz copper. The tank is also rigidly braced with iron or brass angles for stiffening. All angles and seams are riveted and soldered except in larger sizes where tanks are welded.

Air washer **nozzles** differ in design and performance with each manufacturer. Capacities of the nozzles range from .9 to 2.0 gpm, each with pressure ranging from 17 to 30 psi. They are uniformly spaced both vertically and horizontally over the cross section of the washer to give a uniform distribution and density of spray.

39. SMALL COMFORT COOLING SYSTEMS AND UNITS

THERE are four different methods of comfort cooling: (1) Mechanical refrigeration; (2) cold water; (3) air circulation and ventilation; and (4) evaporative cooling. Each method will be discussed from the viewpoint of its advantages and limitations.

Mechanical Refrigeration

1. The most widely used of the four methods is mechanical refrigeration. It is the only one which provides positive means of control to obtain comfort cooling. There are a number of variations in the construction and installation of this type of equipment, which may be classified as follows:

a. A self-contained unit within a room.
b. A self-contained unit within a room, using ducts and grilles for better air distribution.
c. Cooling section within a room (with or without ducts), connected to a remote compressor.
d. A unit installed outside of the conditioned area, using ducts and grilles.

2. At the present time there is a growing preference for **self-contained units** covering small comfort cooling systems. This type of unit has a definite place in small stores, restaurants, homes, apartments, single rooms, etc. The reasons for its increasing popularity can be listed as follows:

a. Made on production basis; low initial cost.
b. Low installation cost; completely assembled at the factory, thus reducing installation to a minimum.
c. Attractive finish permits mounting within the conditioned area.
d. Complete range of sizes available.
e. Isolated rooms may be treated.
f. Units are virtually portable.

A self-contained unit consists of the following parts contained in a single cabinet: compressor, condenser, receiver, direct expansion coil, fan, filter and controls. The unit can cool, dehumidify, filter and circulate air. In many cases ventilating, heating and humidifying are also available.

Self-contained units are constructed with both air and water-cooled **condensers.** The air-cooled condenser requires a large quantity of air that must be picked up out-

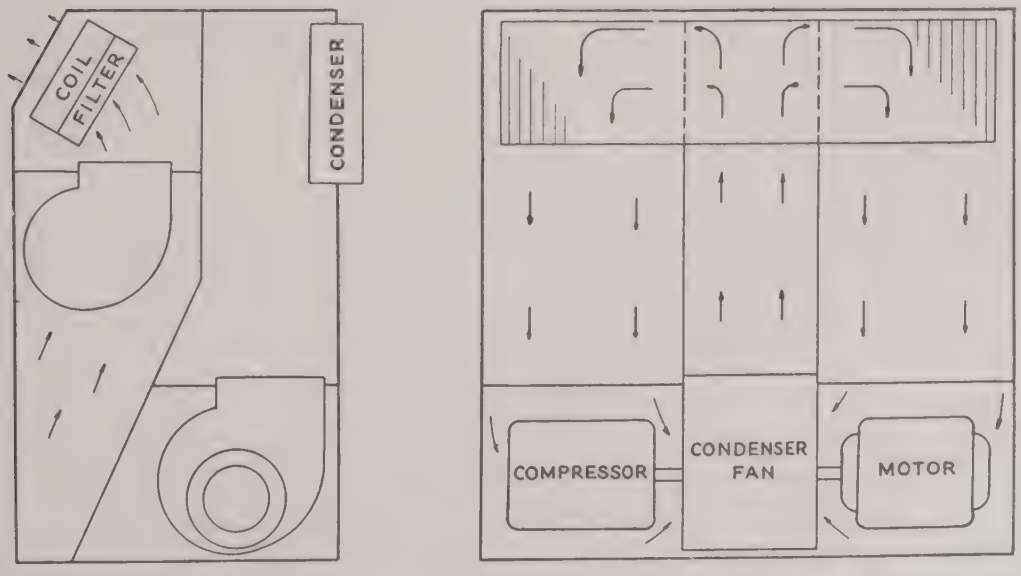


Fig. 1. Floor Type of Self-Contained Unit (Air-Cooled Condenser)

side of the conditioned area, circulated through the high side, and discharged out of the area. It is necessary that this type of unit be installed at an outlet such as a window. Due to this limitation, the air-cooled, self-contained unit has been limited to the $\frac{1}{3}$, $\frac{1}{2}$ and $\frac{3}{4}$-hp sizes. These units are generally referred to as room coolers.

3. There are two general types of **room coolers,** floor and window models. In the floor model **(Fig. 1)** all the mechanism is inside the room and the condenser air circulates through a short duct at the top rear of the unit. This type of unit will take some floor space but requires approximately 6 in. of window space for condenser air. The window model is a complete unit placed in a window **(Fig. 2).** The low side (evaporating coil, circulating fan and filter) is usually on the inside of the room, while the high side (compressor, condenser and condenser fan) usually extends outside. Many designs have attempted to minimize the height of the window type, since a high unit cuts out a large portion of the window, reducing its lighting effect.

Since the room cooler is designed for free air delivery, ducts should not be used. Dampers are usually provided for fresh air intake or exhaust discharge. The condensate from the cooling coil is disposed of by evaporation, using the condenser air; this eliminates the need of a drain. Upon installation a unit of this type requires only an electrical outlet. The normal control is by a manual switch with no provisions for thermostatic control with the unit. Most units are wired to operate the ventilating fan with or without a compressor.

Room coolers can be designed to incorporate **heating** by using the **reverse cycle** principle. Such a unit can provide heat for a single room in the spring or fall when the main heating plant is shut down. Since the refrigeration system is fundamentally a heat pump, removing the heat from the inside and dissipating it to the outside, by reversing the cycle (that is, making the cooling coil the condenser and the con-

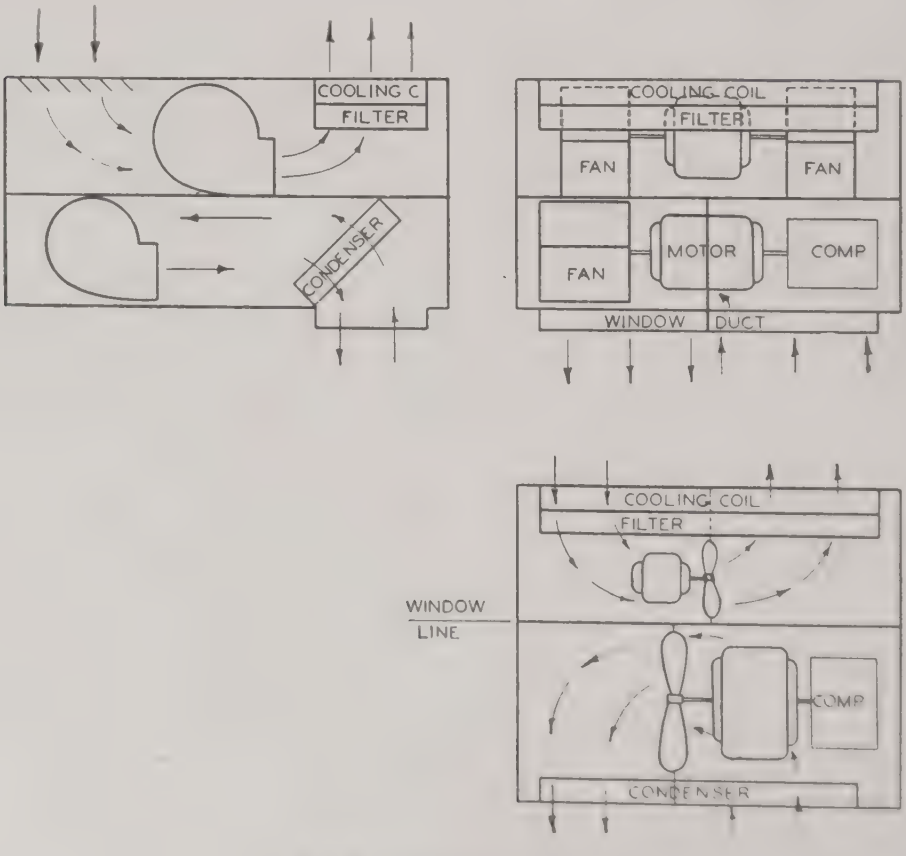


Fig. 2. Self-Contained Units.

denser doing the work of the evaporator) heat can be pumped from the outside air into the room. The heat thus dissipated would be two to four times the electrical energy input. The reverse cycle is limited in usefulness to an outside temperature of approximately 45 F.

4. Self-contained units using water-cooled condensers are usually constructed in sizes of 1 hp or larger, and are commonly referred to as **store coolers.** These units come in a range of sizes as follows: 1, 2, 3, 5, 7½, 10, and 15-hp. The most common are the 3 and 5-hp. It is difficult to distribute satisfactorily large volumes of conditioned air from one outlet; thus store coolers larger than 5 hp are seldom used without ductwork.

The larger self-contained unit **(Fig. 3)** is a complete air conditioning unit designed for installation within a conditioned area. It provides summer air conditioning, that is, cooling, dehumidifying, filtering, ventilation and air circulation. Winter air conditioning, heating and humidifying, can be provided if desired.

The complete high and low side are contained within the store cooler. All electrical work is completed and ready for operation. The only connections required are electrical and water service and sewage. The air intake and discharge grilles are arranged so that they can be mounted on the front or back of the unit as desired for each individual job. The discharge grille is provided with adjustable louvers.

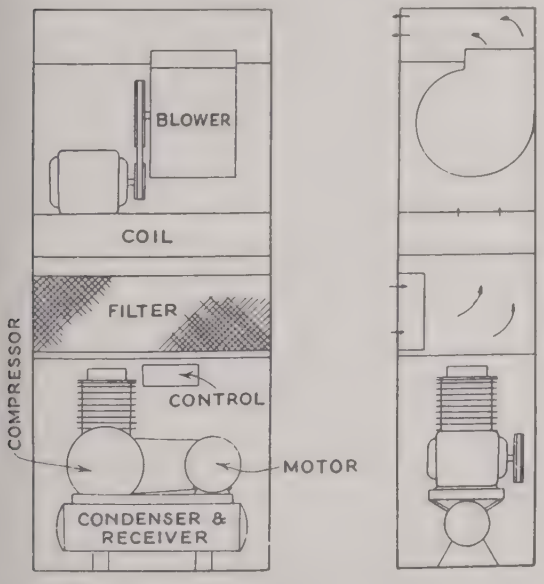


Fig. 3. Complete Air Conditioning Unit Designed for Stores and Offices

When **ducts** are used the increased static pressure can be compensated by increasing the fan speed by means of an adjustable drive pulley on the fan motor.

A **control** switch is provided with "off," "ventilating" and "cooling" positions. When the switch is on the "ventilating" position, the fans provide circulation of air but the compressor is not in operation. At the "cooling" position, the fan and compressor are in operation. The unit may be provided with a thermostat in the air stream and an adjustable control knob mounted on the outside of the unit. If a remote-type thermostat is required, it can be incorporated.

Means of removing motor heat, and the heat of compression, must be provided. This is usually accomplished by passing the condenser water through a small finned cooling coil placed within the compressor compartment. The water passes through the cooling coil before it enters the condenser. Since the larger self-contained units are usually placed outside the conditioned area, the enclosure around the condensing unit is screened or perforated to allow circulation of air.

A Standard (ASRE) Code for testing self-contained units (page 680) has been adopted. Most manufacturers have rated their units under the code conditions. For very rough estimating, the value of one ton per hp is sometimes used, but for exacting comparative values reference should be made to the manufacturer's code ratings.

5. In a **semi-remote installation** the low side is placed within the conditioned area, while the compressor is placed in a remote position. The range of sizes in this type of equipment is usually 1 to 10 tons. It is necessary to run refrigeration leads and a condensate drain from the unit. The suction line and drain must be insulated to prevent dripping on a humid day.

This setup has an advantage over the self-contained unit; since the compressor is remote, the noise level will be lower. These units are usually of the ceiling type which requires less valuable space than the floor unit. An evaporative condenser can be used if desired. A semi-remote unit is

seldom used with ductwork, but if a pressure-type fan is used, ductwork can be applied.

6. In a **remote installation (Fig. 4)** both the high and low side are located outside the conditioned area, with ducts and grilles distributing and circulating the conditioned air. The compressor may be mounted in any convenient place, and an evaporative condenser may be used. This type of equipment usually ranges in capacity from 3 to 25 tons, as indicated in **Table 1**. Units having a capacity up to 50 tons can be obtained. The self-contained unit discussed previously may be used for remote installations.

Although remote installations may prove more expensive, they have definite advantages. Since all the air conditioning equipment is outside the conditioned area, the noise level is at a minimum. Better air distribution can be obtained with ducts and grilles and the ducts can be blended into an architectural design.

It is difficult to determine the exact dividing line between the use of self-contained units and remote installations. The following factors will influence the final decision: Noise, cooling load, size and shape of the floor area, ventilation requirements, decoration problems, initial equipment cost, installation and operation costs.

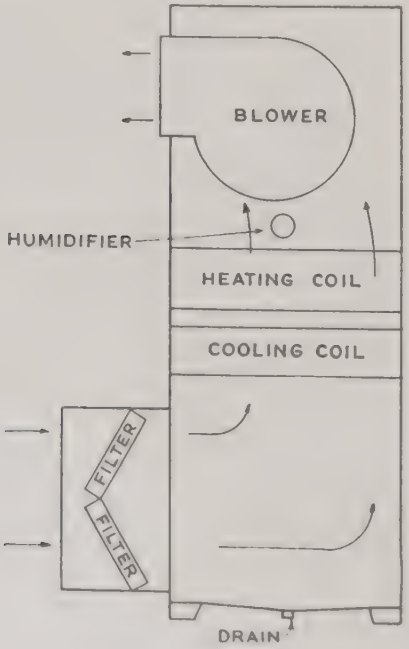
Cold Water Units

7. The use of cold water for a small air conditioning installation is not common, since it requires a large quantity of water. The water temperature must be below 60 F to be effective. To find the quantity of water required, the following formula can be used: gpm $\times$ temperature rise of water $\times$ 500 = Btu per hr.

As an example, let us assume a load of 5 tons (60,000 Btu per hr) with 50 F water and a 6 deg rise in water temperature. The gpm required will be $60{,}000/(6 \times 500) = 20$. The capacity of a cooling coil depends upon the coil temperature. Therefore the size of the coil for a given gpm and capacity will depend upon the amount and temperature of the water.

The **cooling unit** using cold water as a cooling means is similar to the low side of a mechanical refrigeration system. It consists of the following **elements**: Fan and motor, filter and water coil, encased within a metal shell. Three water connections are required—water feed, water return, and condensate drain. In units above 15 tons in capacity, a water spray system is sometimes used, but for smaller systems a water coil is used. The unit is made up as a ceiling or floor type, the ceiling unit being more popular. It may be mounted within the conditioned area or in a remote position.

8. In all air conditioning jobs **fresh air** is desired, and in most instances it is a necessity. The amount varies with the type and size of installation. For values, see page 618 covering load calculations. The



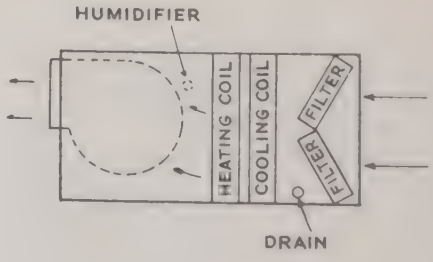


Fig. 4. Floor and Ceiling Low-Side Units for Remote Installations

Table 1. Typical Ratings of Low-Side Units
(Refrigerant at 40 F, entering air at 80 F db and 67 F wb)

Unit number	Air volume, cfm	Coil face area, sq ft	Number of stacks	Capacity, Btu/hr	S.H.R.[1]	Leaving temp, F	
						db[2]	wb[3]
1	1,200	3	2	27,200	.68	66	60.0
			3	37,300	.66	61	57.1
			4	46,000	.65	57	54.5
			5	53,000	.64	54	52.3
			6	59,000	.63	52	50.4
2	2,000	5	2	45,300	.68	66	60.0
			3	62,400	.66	61	57.1
			4	76,600	.65	57	54.5
			5	88,500	.64	54	52.3
			6	98.300	.63	52	50.4
3	3,000	7½	2	68,000	.68	66	60.0
			3	93,400	.66	61	57.1
			4	115,000	.65	57	54.5
			5	132,500	.64	54	52.3
			6	147,500	.63	52	50.4
4	4,000	10	2	90,500	.68	66	60.0
			3	124,300	.66	61	57.1
			4	153,000	.65	57	54.5
			5	177,000	.64	54	52.3
			6	196,500	.63	52	50.4
5	6,000	15	2	136,000	.68	65	60.0
			3	187,000	.66	61	57.1
			4	230,000	.65	57	54.5
			5	265,000	.64	54	52.3
			6	295,000	.63	52	50.4

[1] S.H.R. = Sensible heat ratio = $\frac{\text{Sensible heat}}{\text{Total heat}}$.

1 ton = 12,000 Btu per hr.

[2] db = Dry-bulb temperature, deg F.
[3] wb = Wet-bulb temperature, deg F.

inlet must be so located as to insure the delivery of fresh, clean air, and should be away from the ground to minimize dirt or dust. This intake must be protected by louvers and a wire bird screen and joined to the unit by means of a canvas connection.

9. When ducts are used, care must be taken to keep the **noise level** below an objectionable point. The movement of air through a duct produces some noise, which varies as the velocity of the air. If the sound caused by the air in the duct is below the sound level of the conditioned area, no air noises will be heard. Where there is already some noise, as in restaurants and busy shops, the duct velocity should be designed for 1,000 to 1,100 fpm. Where the noise must be at a minimum, however, the maximum allowable is 650 to 750 fpm. If additional noise reduction is desired, the ducts may be lined with acoustical material.

If the duct is connected directly to the unit, making a metal-to-metal contact, any **vibrations** set up within the unit will be transmitted to the duct and be objectionable at a distant point. To prevent such transmission of vibration, a canvas collar should be connected close to the unit, with one or two inches of slack. The canvas should not be painted. Complete data covering ducts and their application can be found in Chap. 15.

10. In order to permit easy selection by untrained sales personnel, many manufacturers have developed forms for rapid **calculations** of **heat loads.** These methods, based on average conditions, will in general give satisfactory results, but if there are

any unusual conditions the results are misleading. For accurate load calculations see Chap. 37.

An **undersize** unit should not be used with the idea of getting a little cooling effect. The result may be less comfortable

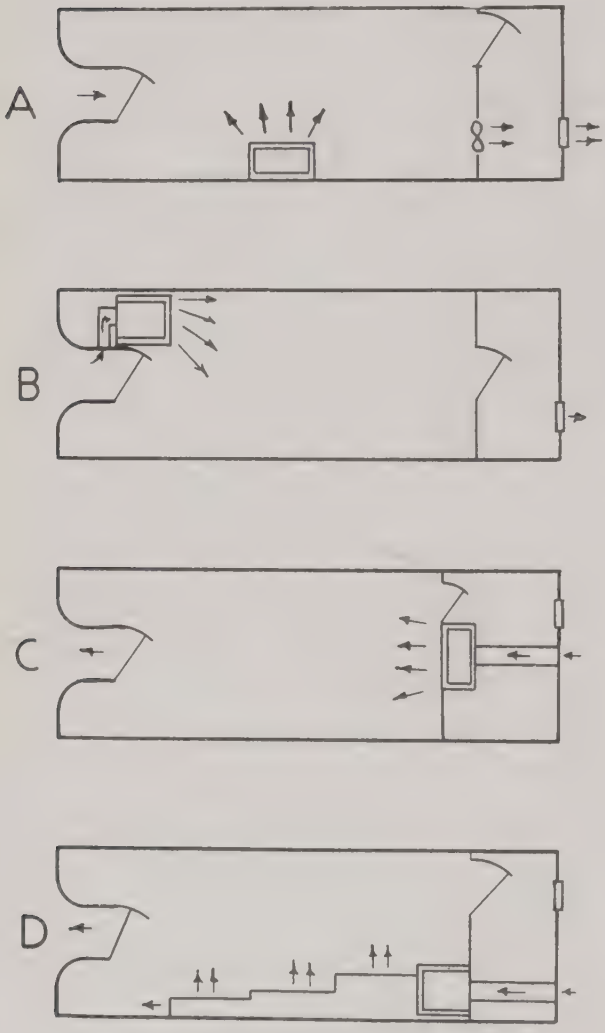


Fig. 5. Typical Air Introduction Methods

than without a unit with the windows open to the breeze. It is not practicable to cool an adjacent room through an open door.

Fig. 5 illustrates a number of methods of cooling a given area. Method A has no ducts, fresh air being obtained by means of an exhaust fan. A self-contained or a semi-remote unit may be used. Method B is the same, except that there is a positive means of securing fresh air.

Method C is a remote installation requiring a minimum of ducts; it uses a single outlet and a single return. A self-contained unit or a built-up unit consisting of a separate low and high sides can likewise be used. Method D is similar except that ducts and grilles are used, resulting in even air distribution, with the minimum of draft. If the size of the room warrants it, a return air duct can be used.

Air Circulation and Ventilation

11. The attic exhaust fan (Fig. 7) has proved popular as an inexpensive means of obtaining cooling effect in a home. While it cannot be called a substitute for complete air conditioning, its principle is sound.

The maximum temperature during the day usually occurs between noon and 4 p.m. During this time a considerable amount of heat is stored up in the construction material of the building. If the air in the attic is not circulated, temperatures as high as 130 F are common. After sunset the outdoor temperature begins to drop, with the lowest temperature occurging between midnight and 6 a.m. The difference between the maximum and minimum temperatures during the day varies from 15 to 20 deg F. The heat that is stored in the building materials is slowly dissipated during the night.

The attic fan should be started after sundown to draw cool air from the outside and circulate it through the house. The indoor temperature is lowered quickly and the movement of air also has a cooling effect on human bodies within.

There can be no set method of adjusting and controlling the air flow as each house differs in construction. It is advisable in those rooms to be cooled to open the windows most distant from the fan. It may be necessary to close or partly close other windows.

Care must be exercised in selecting the proper fan size. To obtain best results a fan should be selected to give one air change of the entire area in two to four minutes. This quantity of air will be sufficient to cause a gentle breeze through the area.

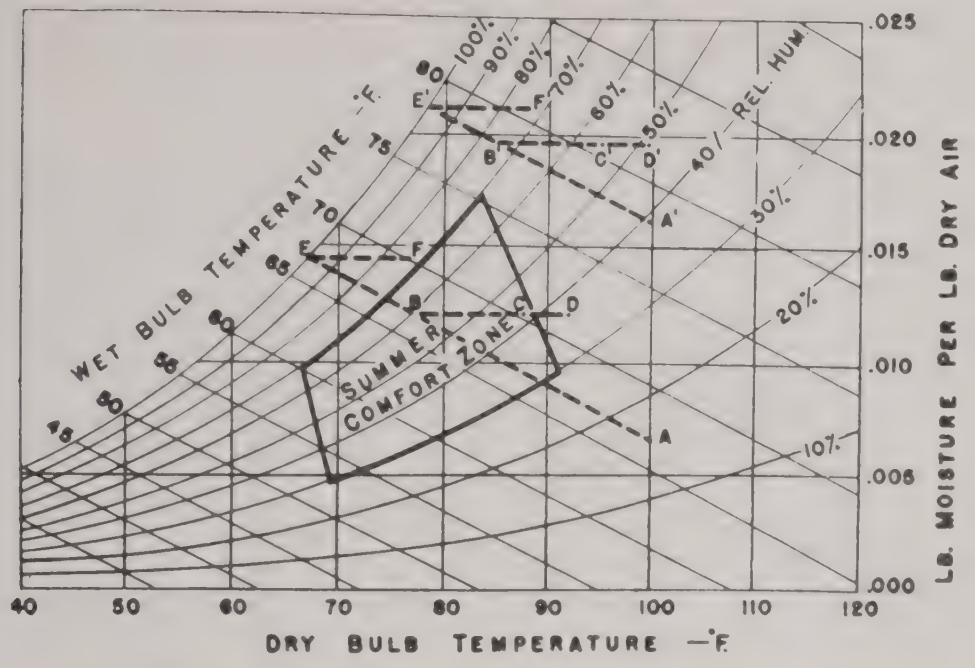


Fig. 6. Outline of Psychrometric Chart with Desert Comfort Zone

Evaporative Cooling

12. In areas where the air is very hot and dry it is possible to reduce the dry-bulb temperature by passing air over a wetted surface. The evaporation of the water requires heat; as the air touches the moisture, the heat necessary to evaporate the water is taken from the air so that the temperature of the air is lowered. However, the sensible heat is converted into latent heat. Thus total heat and the wet-bulb temperature of the air remain unchanged.

If the outside air is very dry, a considerable reduction in dry-bulb temperature will be obtained and at the same time the moisture content may not rise high enough to be objectionable. By studying the psychrometric chart (**Fig. 6**), this principle may be illustrated. Point A may be taken as outside air conditions. Point B represents a desirable air condition leaving the cooler. Point C might be a suitable air condition for discharge from the house. Point D represents an unsatisfactory condition in the last room being cooled. The maximum air cooling from conditions at A with this system is shown by point E. The line E to F then represents air travel through the house and is not satisfactory for human occupancy. Another outside air condition possible is shown by A′. The points B′, C′, D′, E′, and F′ represent the corresponding conditions that might be expected from the operation of such a cooler. No condition reached from A′ would be satisfactory.

If the air leaving an evaporative cooler is recirculated, the wet-bulb temperature would continue to increase and would result in unsatisfactory conditions. There-

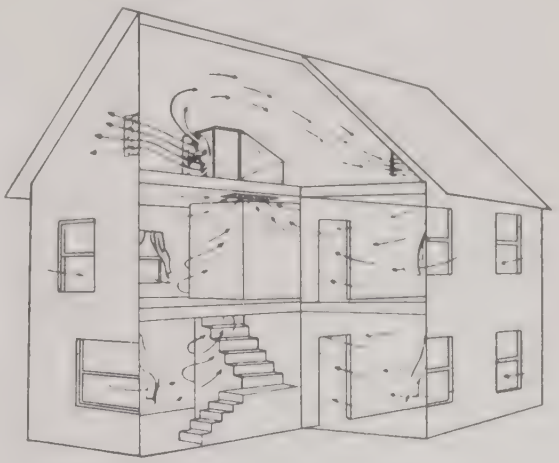


Fig. 7. Air Circulation in House with Attic Fan

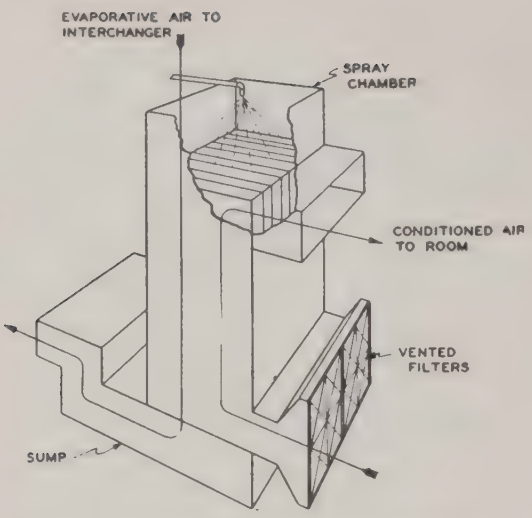


Fig. 8. Evaporative Cooler Unit, Showing Possible Air Movement

fore, fresh outside air should be used with no recirculation. This type of unit cannot be used in damp climates because the moisture content of the air leaving the unit is very high, thus raising the humidity above the comfort level.

Indirect evaporative coolers have been used to give a more positive cooling effect. This is accomplished by passing one air stream by a wetted surface to lower the dry-bulb temperature **(Fig. 8).** This air in turn is used to cool a separate air stream, with which it does not come in contact; thus the moisture content of the conditioned area remains unchanged, while the dry-bulb temperature is reduced. Since there must be a considerable difference between dry bulb and wet bulb, the indirect evaporative cooler is limited to dry climates.

Evaporative cooling is definitely limited in usefulness and cannot be called a substitute for complete air conditioning.

40. CONTROL OF AIR CONDITIONS

THIS chapter deals mainly with systems of control, but also touches upon the important related mechanical characteristics of mechanical equipment and of individual control devices. The material is taken from control problems which have been developed and solved in the field.

Load Characteristics versus Comfort Conditions

1. **Air conditioning load** includes heat gain from outside, sun heat through windows and wall surfaces, ventilation air, and various forms of internal sensible heat, such as light, power equipment, office machines and people. In addition, humidity load is introduced by ventilating air, by infiltration and by people or other internal sources. Rather wide ratios of sensible load h_s to humidity load h_L are experienced in the normal variation of these factors. In general, the h_s/h_L ratio increases with the severity of the air conditioning load, particularly so where outside weather conditions have a large influence (see Chap. 11).

Both sensible cooling and dehumidifying are accomplished at the same time by the usual **surface-type** air conditioners. Their ratio depends on psychrometric conditions which impose certain limitations. Comfort systems are usually designed to maintain temperature below a certain maximum at peak load. Comfort requirements dictate that temperature must not drop below a certain minimum. In controlling humidity as well as temperature, the control system must operate within these limits of dry-bulb temperature. For example, in an attempt to reduce relative humidity, it is not permissible to operate the equipment to cool below the limit selected, even though some air conditioning equipment will actually cause relative humidity to rise while the air conditioner operates at low dry-bulb temperature.

Adjustment of Conditioning Equipment to Load

2. Reduction of the **air velocity** over the coil by changing the amount of air circulated or by bypassing some air around the coil automatically, as the cooling load decreases, reduces the sensible capacity without a corresponding decrease in moisture removal. Usually a satisfactory humidity may thus be reached before the dry-bulb temperature falls to the low limit. In using the former method, distribution of air within the space must be properly maintained at the lowered velocities.

Sensible heat may be added anywhere in the system, except in the return air ducts, to provide reheat and offset the low operation required to dehumidify. **Reheat** may be by steam, electricity, hot water or condensation of the refrigerant and is normally controlled by return air temperature, on comfort installations, or from rise in humidity in cases where a higher temperature is either allowable or required.

The equipment capacity at **light load** is extremely important where a schedule of inside temperatures against outside temperatures is set up by controls. Practically, it is impossible for equipment to follow rapid outdoor temperature changes, and under certain conditions more nominal refrigeration capacity is required to maintain the inside temperature equal to the outside at, say, 75 F, than to maintain a 15 deg differential at 90 F outside.

Compressor capacity may be controlled on individual units by the use of unloading or bypass valves. Two to three steps of capacity are often employed for a single compressor. Likewise, where several compressors are employed, they may be started and stopped in any order desired to give the required steps of capacity. In many cases, multiple compressor installations may also employ bypass valves on the individual compressors. In controlling such systems, care must be taken to provide sufficiently wide steps of change in the refrigerated medium to prevent short-cycling from one step of capacity to another.

For example, in a system like **Fig. 1,** if a given step of capacity reduction is called into play at water temperature of, say 40 F, while the capacity will be restored at

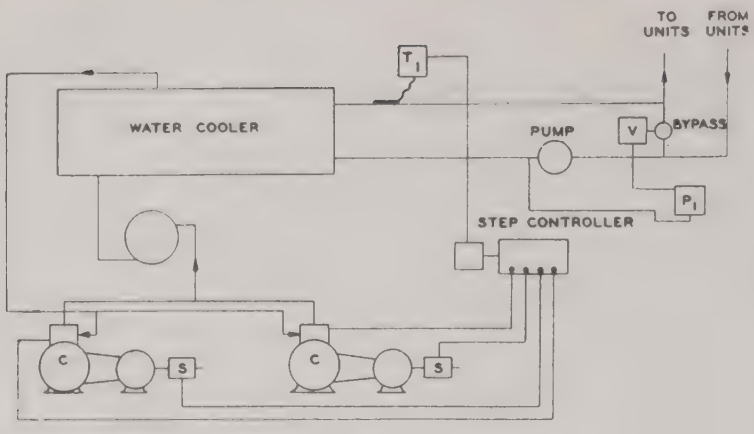


Fig. 1. Operation of Step Control of Capacity

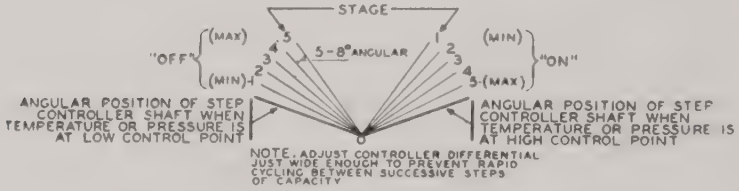


Fig. 1a

water temperature of 40.5 F, it is important that the reduction of refrigeration capacity shall not immediately result in a water temperature rise of 0.5 F. When step-controllers operate multispeed compressors to adjust capacity similar problems exist, but proper analysis results in an extremely flexible and satisfactory installation.

The proper positions for making and breaking of switches which operate the various capacity steps for application to this type are shown in **Fig. 1a.** Five steps of capacity are shown, but the same principles apply for any number. Note that the "on" and "off" points are grouped at the end of the step controller travel and that the differential of the controllers has been adjusted just wide enough so that short-cycling from one step to another does not occur.

If pneumatic controls are used for step operation, a similar sequence is obtained by proper adjustment of the pressure controls for each step of capacity.

Application Principles

3. In applying controls, attention to a few basic principles is important for best results. A control instrument can react only to the air immediately surrounding it. Where loads are dissimilar in various parts of the building **separate zones** should be set up. This may be caused either from unequal exposure or unequal internal loads. For example, see **Fig. 2.** In this L-shaped building, zones 1 and 2 are never exposed to the sun, while zones 3 and 4 are exposed at different periods of the day. However, on the top floors of the building, zone 1 may have sun for a period early in the morning. This would call for at least three zones in the lower floors and four zones in the upper floors of the building. Corner rooms such as 5 may often be served partly from each adjoining zone.

In direct-expansion systems, the compressor should not be allowed to operate unless the fan of the conditioning unit is in operation. Likewise, in a water cooler, the compressor should not operate unless the circulating pump of the water cooler is operating. Proper wiring between fan or pump starter and compressor starter will provide this feature.

In addition, it is often desirable to prevent outside air from entering the conditioner, particularly in the in-between seasons when freeze-up might occur. Where

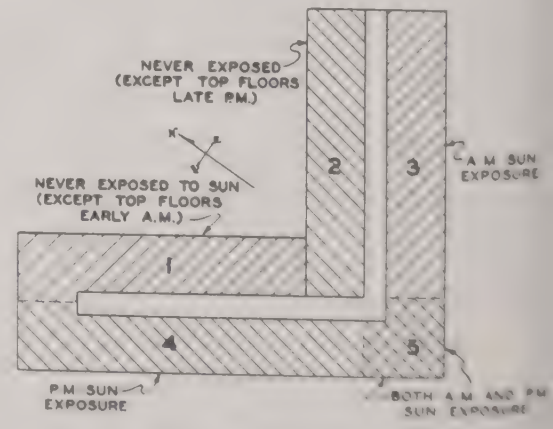


Fig. 2. Zoning of Buildings

manual control is not satisfactory, this may be accomplished automatically with an electric damper motor which closes when the fan is off, or by a pneumatic control system using a solenoid valve in the air line to the damper motor. In either case, de-energizing the fan closes the outdoor air damper.

Sizes for **modulating valves** controlling steam or hot or cold water to coils must be chosen for the actual load, and not simply by reference to the size of existing connections. An oversize modulating valve cannot provide proper control. In modulating or floating control systems, particularly those in which several steps of compressor capacity are employed, enough change in temperature of the controlled medium must be allowed within the control range to prevent short-cycling or hunting.

Positive positioning of dampers or valves on either electrically or pneumatically operated controls is important for best operation of any system. It is now available with pneumatic systems as well as with electrically operated devices.

Where automatic changeover from **summer to winter** is to be provided under automatic control, it is important to have a dead spot between the heating and cooling functions. That is, there should be some period of inoperation in both heating and cooling between the point at which the controls will call for heating on one end of the scale or cooling on the other. Otherwise, serious cycling from one function to the other will occur.

Where refrigeration storage is employed, the proper time cycle is often established by an automatic timer. Sufficient time should be allowed in the morning to pick up the conditioning load well before any heavy load increase in the conditioned spaces.

Typical Control Systems

4. The control systems here shown are taken from actual installations and illustrate some common control methods for both large and small jobs. All figures are schematic, control functions being obtained by either electric or pneumatic devices. A single line connecting controls represents as many wires or pipes as the apparatus requires. The source of power is not shown, nor are safety controls such as high and low-pressure cutouts for the compressors.

Some of the systems shown in which controls automatically affect the compressor capacity are generally arranged on the larger installations for manual starting and stopping of the compressors, the only automatic function being capacity adjustment or emergency shutdown by the safety controls.

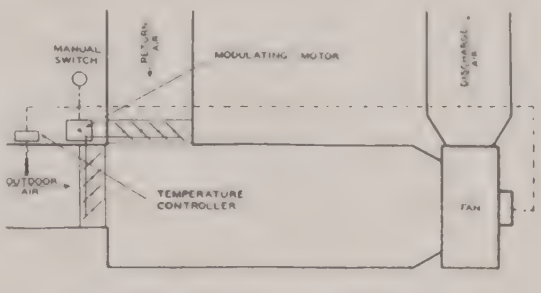


Fig. 3. Relative Amounts of Outdoor and Return Air

It is impossible to illustrate all combinations of apparatus and control, but the diagrams shown illustrate the common problems involved, and may serve as the basis for building through combinations with other apparatus to provide any of the common control sequences. For simplicity, control of the relative amounts of **outdoor** and **return air** are not shown on the separate figures, except when outdoor air control is tied in with other control functions. However, the system of **Fig. 3** may be used on most air conditioning systems or units. Referring to this figure, the modulating motor operates both the outdoor air and return-air dampers. The linkage is so arranged that as one damper opens, the other closes. The motor is interconnected with the fan starter so that whenever the fan is not running, the outdoor air dampers will automatically close.

When the fan runs, the outdoor temperature controller positions the motor. The settings of this temperature control are so arranged that at, say, 65 F outdoor air temperature, the maximum amount of outdoor air is introduced, and as the temperature rises above this value or falls below it, the portion of outdoor air is reduced. Proper settings of the thermostat result in

a maximum of economy through the introduction of greater than the minimum amount of outdoor air whenever it can assist in cooling the space.

The minimum amount of outdoor air required for the number of people in conditioned areas can be adjusted by a manual switch so that this minimum will always

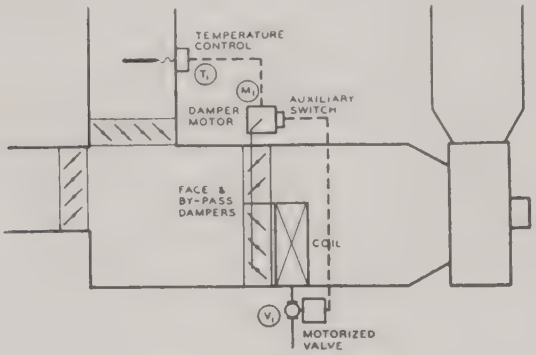


Fig. 4. Damper Control

be introduced except when the fan is not running. Where necessary, a second damper motor can simultaneously control exhaust dampers. This is often required in localities where larger quantities of outdoor air can be introduced economically for long periods of time.

In localities where excessively long periods of high **relative humidity** are likely to recur frequently, the motor position may also be affected by an outdoor humidistat (not shown) to prevent introducing large quantities of excessively humid air. This control arrangement depends somewhat on the flow characteristics of the dampers employed, to produce the desired mixed-air temperatures, but in general will give more uniform results than a thermostat located in the mixed air, unless the bulb of such a thermostat can be placed in the fan discharge. This is because of the difficulty of finding the proper location for the bulb in the mixed-air stream ahead of the heating or cooling equipment, so that it will be affected by the true average temperature.

It is obvious that where this or similar control systems control outdoor air in the winter, the heating and cooling coils must be of the non-freeze type.

Types of Systems

A. Control of Simple Chilled Water Unit

5. One of the simplest methods of controlling a single unit cooled by chilled water is through the use of face and bypass dampers as in **Fig. 4.** The damper motor controls these dampers so that as the bypass dampers open, the face dampers close. When wide open bypass dampers are not large enough to supply the full quantity of air to a system, it may be necessary to position the face dampers so they will not close completely in order that full quantity of air can get to the fan. An auxiliary switch on the damper motor operates to close a valve in the chilled water supply to the coil when the face dampers are completely closed and open it as soon as the face dampers start to open.

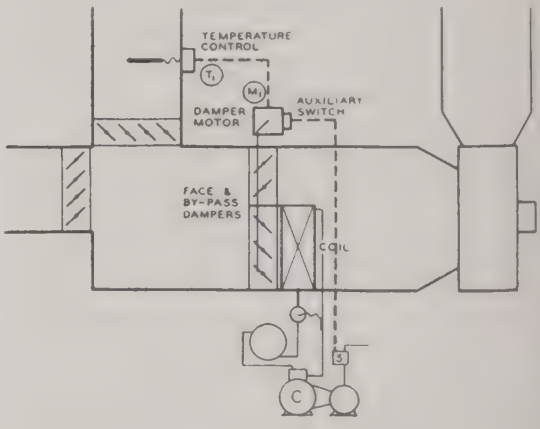


Fig. 5. Simple Control for Direct-Expansion Systems

This valve is an economy feature particularly desirable on well-water systems, where the water passing through the coils is wasted. However, it also saves considerable operating cost where units are served from a central water cooler. The damper motor itself is controlled by a temperature control in either the return-air to the unit or in the conditioned space. Usually the valve in the water supply is of the two-position type, since this results in better dehumidification under light load conditions.

Systems of this type employing modulating valves may suffer from lack of dehu-

midification at light loads unless the modulating range is held within the early part of the opening of the face damper.

B. Simple Control System for Direct-Expansion Unit or Central Systems

6. Fig. 5 shows a system somewhat similar to that of Fig. 4, but adapted for direct-expansion installations. As in the previous system, a return-air thermostat operates the damper motor which positions the face and bypass dampers.

Where only one or two units are cooled by a single compressor, the **linkage** to the **damper motors** should be so arranged that the minimum closure of the face dampers will permit enough air to pass over the coil to maintain a desirable suction pressure, one which will not cause serious frosting of the coil.

Frosting may be eliminated when compressors are equipped with unloading

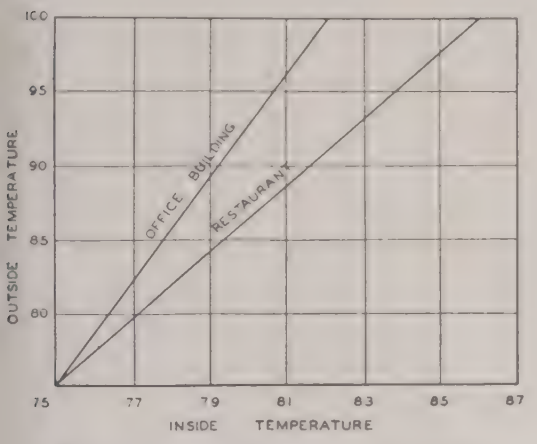


Fig. 6. Desirable Inside Temperatures

mechanisms which prevent low suction pressure, such as shown in **Fig. 14.**

When a **single cooling unit** operates from one compressor, an auxiliary switch on the damper motor may be employed to start and stop the compressor as shown. Either the damper motor itself or the compressor motor starter should take its power through the fan starter so that the system cannot run until the fan is operating. If the system is wired in this way, the damper motor must be of the automatic closing type.

Where **several units** operate from a single compressor, the auxiliary switch from the damper motor may alternately be connected to control a valve in the liquid line to the individual unit. This arrangement is not shown, but its utility is obvious.

A **direct-expansion** arrangement with the bypass controlled in this manner or-

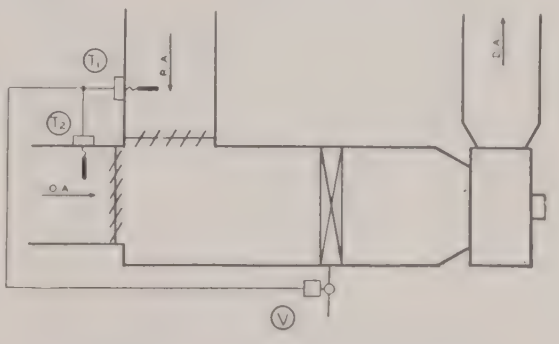


Fig. 7. Outdoor Temperature Control

dinarily gives relatively good humidity conditions at light loads. When the cooling equipment is carefully sized to the load conditions, it is often possible to employ a rather large differential in the return-air thermostat to obtain some fluctuation in the temperature in the conditioned spaces as the load varies. In this way, on the warmer days when the load is usually highest, the temperatures will be highest, and on the cooler days when the load is least, the temperatures will be at the lower end of the range. This is desirable, since it serves to vary the differential between inside and outside, according to weather conditions.

C. Outdoor Compensation

7. For **short-time occupancy,** it may be desirable to vary the inside conditions in proportion to outside temperature to reduce entering and leaving sensations of the people who come into the conditioned space. This also is often done for somewhat longer periods of occupancy, such as theaters and office buildings, but the variation is usually not so great in these cases.

Fig. 6 shows desirable inside temperatures for use with various outside temperatures, found to produce reasonable degrees of comfort in the northern and central zones of the United States. Such a schedule

may be set up automatically by means of a control system such as that in **Fig. 7.**

The **outdoor temperature controller** measures the outside temperature and determines the point at which the inside temperature controller will call for cooling if a two-position control is employed. If modulating control is used, the modulating range through which the valve will be positioned by the inside controller is varied. The return-air thermostat of Fig. 7 operates a valve to control the amount of cooling fluid admitted to the coil. Alternately, it could of course start and stop the compressor of a direct-expansion system.

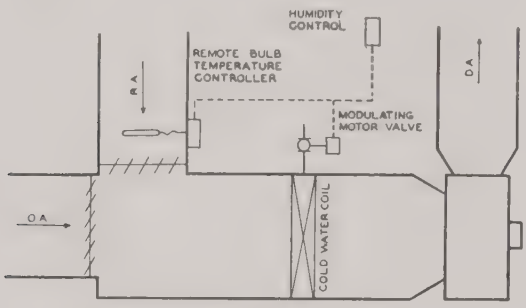


Fig. 8. Motorized Valve Control

As previously mentioned, inside temperatures cannot be expected to follow outdoor temperature fluctuations rapidly. Many engineers do not consider it desirable that they should. However, this type of control system will follow day-to-day trends in outdoor temperature, and has the advantage that when outdoor temperature is low, a lower inside temperature is held, which normally results in better relative humidity conditions.

D. Modulating Control with High Relative Humidity during Light Loads

8. The advantages of modulating control of the cooling fluid are fairly obvious, since rapid fluctuations in delivered air temperatures are eliminated, resulting in less disturbance to air distribution within the conditioned spaces and contributing to improved comfort. But where chilled water or brine is used as the cooling medium, the average temperature of the water in the coil is raised as its flow is throttled. This results in a smaller proportion of dehumidification, and on jobs where high humidities may be troublesome, uncomfortable conditions may be encountered, particularly at light loads.

To overcome this, the system of **Fig. 8** may be employed. The return-air temperature controller ordinarily operates to modulate the motorized valve to control the flow of cold water through the cooling coil. However, if the humidity rises to a predetermined high value, the humidistat causes the remote-bulb temperature controller to operate the valve in a two-position manner and thus provides a maximum of dehumidification during the cooling period, and yet the system is prevented from cooling to too low a temperature by the temperature controller.

In this way, the advantages of modulating control are maintained as long as practical, and the system reverts to two-position control whenever greater dehumidification is required. This system may also be used with direct-expansion coils by means of modulating attachments to the expansion valves or modulating liquid valves, but it should be noted that on most such systems, it is not necessary to revert to two-position operation, since the suction pressure automatically reduces as refrigerant is throttled, due to the re-established balance between coil capacity and machine capacity, which provides a greater ratio of dehumidification. The amount of reduction in suction pressure which is permissible is

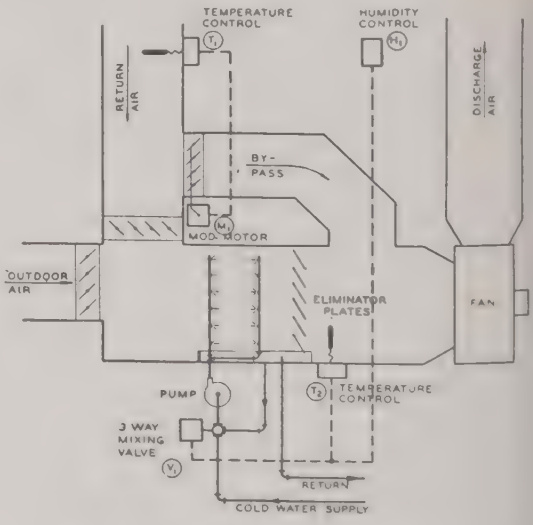


Fig. 9. Control of Air Washer Cycle

limited by the lowest practical coil temperature, and hence, even on direct-expansion systems, it is sometimes desirable to revert to two-position operation when humidity is high.

E. Air-Washer Control

9. Many medium and larger-sized systems employ air washers instead of extended surfaces for cooling. The degree of automatic control employed with air-washer equipment has increased in recent years, and **Fig. 9** shows a fairly complete control of the cooling cycle of an air-washer system. The outdoor air control shown in Fig. 3 could be applied very nicely to an air washer of this type, particularly during the cooling cycle. Cold water could be supplied from a system such as that shown in Fig. 1. There are times when a face damper located completely across the face of the air washer is necessary to obtain proper air pressure relationships so that the amount of air bypassing can be controlled.

The pump sprays water into the air stream, from which it is collected in the pan and returned to the pump suction. A three-way mixing valve is controlled by dew-point temperature controller T_2. This valve mixes the cold supply water and the water from the pan to give a constant dew point to the air leaving the eliminator plates. This occurs since the air is practically saturated in passing through the air washer and hence the temperature of the air equals its dew point.

The temperature within the conditioned space is controlled by return-air and bypass dampers operated by the damper motor M_1, controlled by the return-air temperature controller T_1.

With this much control, the relative humidity attained in the space will vary somewhat, since smaller quantities of air pass through the sprays and are treated to the same dew-point temperature at light loads. Where it is necessary to overcome this to give a more constant relative humidity, the h_s/h_L ratio can be varied considerably by compensating the action of the dew-point controller through a humidity control H_1, located in the conditioned space. This control measures the relative humidity in the space and determines the dew point to be maintained by T_2. The higher the relative humidity, the lower the dew point produced by the combined action of the three-way mixing valve and temperature controller T_2. Also, T_2 and H_1 could control a straightway valve in the cold water supply line with a check valve in the pump suction from the pan, and obtain the same results as with a three-way valve.

With this type of control, the lowest h_s/h_L ratio attainable is determined by the lowest cold-water supply temperature obtainable.

F. Blower Type Zone Installation

10. Fig. 10 shows a system suitable for some of the larger zone jobs employing a central blow-through type fan. With this arrangement it is usually practical to control the outdoor air and return-air dampers from the mixed-air temperature so that when no cooling or heating is required, the proportion of the two will be such as to give 65 F air whenever such a mixture is possible. The minimum outdoor air quantity would ordinarily be set by a manual switch and the outdoor air damper motor should be so connected that when the fan is stopped, the outdoor air dampers are closed tightly.

The temperature of the air leaving the fan will be at, say, 65 F, whenever outdoor air is sufficiently cool to produce this low a temperature in the mixture. Other-

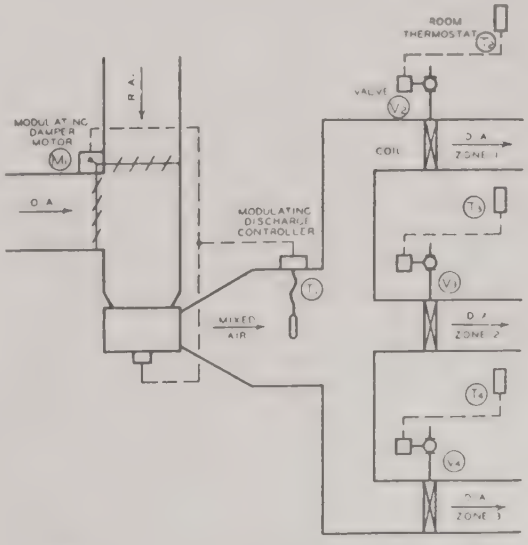


Fig. 10. Blower Type Zone Installation

wise, it will be at some higher value. This air is delivered through a duct system to the separate zones 1, 2 and 3, in each of which is located a cooling coil. The cooling provided by each coil is controlled by a room thermostat located in the corresponding zone.

Such a system might be very applicable to a building such as that shown in Fig. 2. The separate zones may be controlled either from modulating or two-position room thermostats, and individual coils may be equipped with bypass around the coils wherever desired. When so used, satisfactory temperature and relative humidity conditions should be experienced over normal ranges of load.

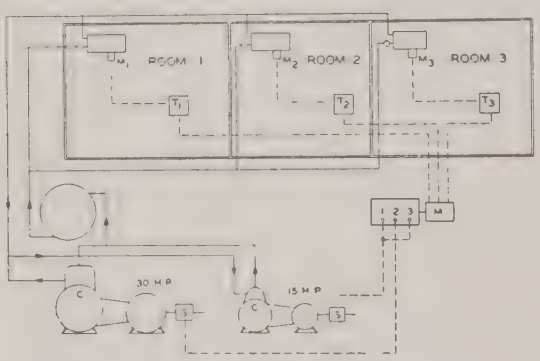


Fig. 11. Compressor Capacity Control

This system is also used for heating where zoning is necessary, by having the individual room thermostats control modulating steam valves on individual heating coils in the various ducts.

A common variation of the system shown on Fig. 10 is to use a fan discharging from a central duct system to individual zones where booster fans, individual air heaters and supply and bypass dampers are employed. This arrangement helps to reduce duct sizes in buildings.

On some installations, both cooling and heating coils are employed, so that the same system performs the dual function of heating and cooling. From the control point of view, it is better to have the air flow first through the cooling coil and then through the heating coil unless other considerations make this arrangement impractical. When this is done, it becomes entirely practical to heat in some zones while cooling in others or to cool whenever it is necessary to keep the humidity low in a given zone and to provide reheat in the heating coil of that zone to prevent excessive lowering of temperatures. Reheat could also be supplied by utilizing the waste heat of the condenser gases or the heat in the condenser water.

G. Control of Compressor Capacity for Zone Systems

11. Various methods are employed with zone systems such as that shown in Fig. 10 to vary compressor capacity. Compressor bypass valves, clearance pockets or two-speed motors may be automatically controlled to vary capacity, or a number of units of different horsepower may be used to provide various steps of capacity under automatic control. **Fig. 11** shows a system of the latter type in which one compressor is twice the capacity of the other. By turning on the smaller, then starting the larger and stopping the smaller, and then adding the smaller compressor, three steps of capacity are possible with only two compressors.

A step-controller is ordinarily employed to produce this sequence of capacity. The step-controller may be operated in response to suction pressure where conditions permit. However, there is less danger of short-cycling any of the steps of capacity if they can be cut in and out from room conditions.

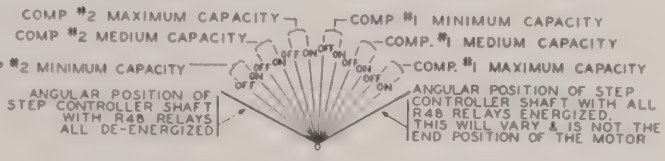
The system of Fig. 11 accomplishes this as follows. The room thermostats in the separate zones have a cumulative effect on the step-controller M so that the greater the demand for cooling, the greater will be the compressor capacity employed. At the same time, these thermostats in individual zones can control motors M on face and bypass dampers, which in turn can close valves in liquid lines to the individual cooling units if required.

This control system assures that the refrigeration capacity shall approximately match the load conditions. The position of the step-controller represents the sum of the loads in the various zones, and since these loads change rather slowly, the danger of short-cycling from one step of capacity to another is largely eliminated. It is well to protect a system of this type

from low suction pressure by a suction pressure switch, independent of the other controls.

The adjustment of the step controllers for applications of this type in which the summation of the total load is measured, should be as shown in **Fig. 12**. Note that this differs materially from the application discussed relative to Fig. 1.

Fig. 12



For pneumatic controls, the same operating sequence should be followed.

This method of controlling the step-controller is just as applicable when controlling compressor bypass valves or a multi-speed motor on a single compressor. If one zone has a greater load than another, the effect of its associated thermostat on the step-controller can be correspondingly greater than that of other zones.

Many compressors are now equipped with capacity control which unloads the individual cylinders successively by holding open their suction valves. This action is usually responsive to suction pressure, and thus the adaptation of compressor capacity to load is automatic. A typical mechanism of this type is shown in Fig. 14. An outstanding advantage of this method is that the mechanism can be allowed to respond to very small changes in suction pressure and thus any desired pressure can be accurately obtained whether the compressor has one, or ten, or more cylinders.

The dotted lines of Fig. 14 show the addition of **pneumatic control** equipment to vary the operating suction pressure in response to temperature or humidity as measured in the conditioned spaces, or wherever else desired. Similar means for adjusting the suction pressure with electrical controls may be used.

H. All-Year System

12. Fig. 13 shows a typical all-year system. Outdoor air and return air are proportioned according to the demand of temperature controllers T_2 and T_3, so that when outdoor temperature is either above or below a mean value, the amount of outdoor air is decreased until the minimum set by S_1 is reached.

The cooling cycle is controlled by T_1 and H_1 in the return air. The face and bypass dampers of the coil are affected only by the temperature of the return air, so that a definite low limit for the cooling

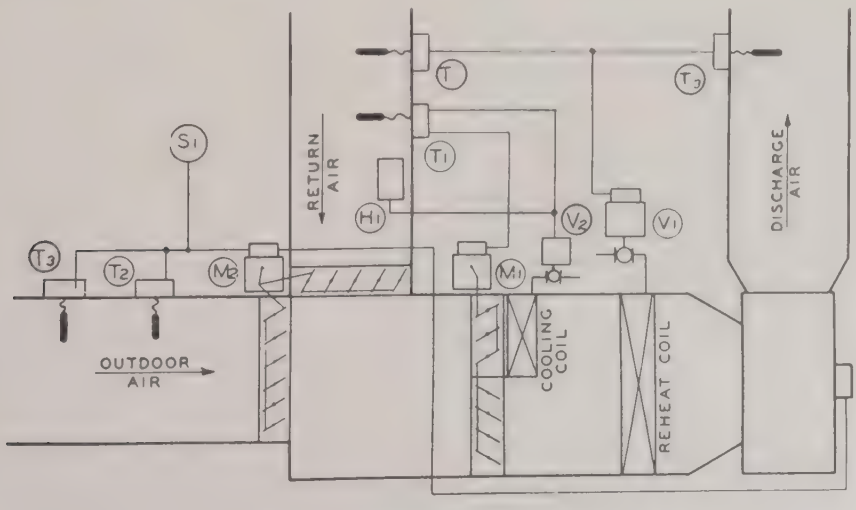


Fig. 13. Typical All-Year System

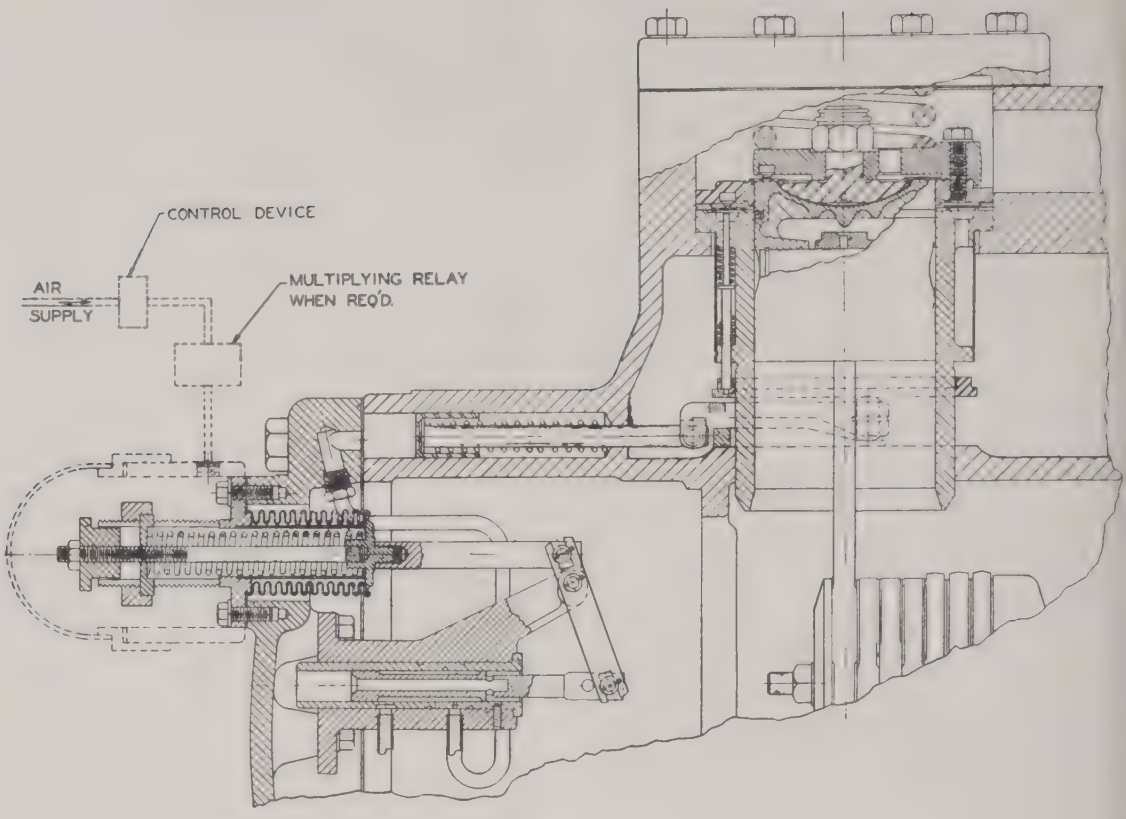


Fig. 14

cycle is provided. As the temperature nears its low limit, the cooling fluid to the coil is throttled for economy by T_1.

At any time when humidity is high, the humidistat H_1 keeps valve V_2 open to provide a greater ratio of dehumidification. If reheat is available all year, V_2 may be connected to M_1 to open the bypass dampers at least to a minimum position whenever V_2 opens wide, as by humidistat action. Further reduction in temperature causes return-air controller T_5 to begin to open the heating valve V_1 and provide the necessary heat to maintain constant temperatures. Usually a discharge temperature control T_4 is required in the winter time to prevent the discharge of air at an uncomfortably low temperature.

In the **cooling cycle,** this thermostat will sometimes need to be made inoperative, which can be done by a manual switch or by extra contacts in T which operate on a temperature rise somewhat above the point at which the heating valve closes.

This system, with the proper settings of T_1 and T, can be made to give automatic changeover from heating to cooling, but is also adaptable to manual changeover by the addition of the proper manual switches.

Winter humidity control can be supplied by the addition of humidifying sprays, pan-type humidifiers or other humidifying equipment, controlled by a return-air or space humidistat. Humidifying equipment, like the outdoor air motor, should take its power from the fan power supply so that it cannot operate except when the fan is running.

I. Refrigeration Storage

13. Since many air conditioning systems have rather short peak loads either from weather or occupancy, smaller refrigeration equipment can adequately condition such spaces if storage for refrigerating effect is provided and the smaller equipment allowed to run at least a portion of the time while cooling is not actually required.

Two types of systems are used—one in which a **water cooler,** such as shown in Fig. 1, is used to chill water and store it at above freezing temperature in a large in-

sulated storage tank. In this case, a control system such as shown in Fig. 1 can be used for the control of the refrigeration equipment. The thermostat in the water supply line to the units and storage tank is ordinarily set somewhat lower than would otherwise be the case. Control of the individual units can be accomplished from any of the systems applicable which are mentioned above.

In the second system, a special type of water cooler is employed, in which ice may be accumulated without danger of bursting tubing or otherwise damaging the cooler. In this case, control of the compressor is accomplished either from a remote-bulb temperature controller with the bulb so located with respect to the evaporator in the water cooler as to limit the thickness of ice formation, or it may be controlled from a timed cycle so that during a certain period of time, say at night, the refrigeration compressor will run continuously to build up ice.

PART VII

MISCELLANEOUS

Chapter *Page*

41. Glossary of Refrigeration—B. H. Jennings 659
The definitions of this glossary have been freely drawn and cross checked from many authoritative sources, and it is hoped the glossary represents usages accepted by the refrigeration industry. In preparing this glossary, the editor has received invaluable help from many individuals with the suggestions of Messrs. Glenn Muffly, F. H. Faust, Gayle B. Priester, and L. C. Bastian being most extensive.

42. Codes and Standards 681

41. GLOSSARY OF REFRIGERATION

absolute zero—the zero point on the absolute temperature scale, 459.7 degrees below the zero of the Fahrenheit scale, 273.16 degrees below zero on the Centrigrade scale.

absorbent—a material which has the ability to take up water vapor or some other vapor but which changes chemically, physically or both, during the cycle. Calcium chloride is an example of a solid-liquid material, while liquid materials include solutions of lithium chloride, lithium bromide and the ethylene glycols.

absorber—a device containing liquid for absorbing refrigerant vapor or other vapors. In an absorption system that part of the low side used for absorbing refrigerant vapor.

absorption—a process whereby a liquid substance can act in such manner as to receive another liquid or gaseous fluid which in turn becomes part of the common resulting solution.

absorption system—a refrigeration system in which the refrigerant gas evolved in the evaporator is taken up in an absorber and released in a generator upon the application of heat.

acceleration due to gravity—the rate of increase in velocity of a freely falling body ignoring the buoyant effects of the air. Its value varies with latitude and elevation. The International Standard taken at sea level and 45° latitude is 980.665 cm per second per second or 32.174 ft per second per second, symbol (*g*).

access doors—doors provided in a unit to permit inspection of the inside, as of plenum chambers.

accumulator—a storage chamber for low-side liquid refrigerant.

activated alumina—a form of aluminum oxide which absorbs moisture readily and is used as a drying agent.

adiabatic process—a thermodynamic process in which no heat is extracted from or added to the system of the process.

adsorbent—a material which has the ability to cause molecules of gases, liquids or solids to adhere to its internal surfaces without changing the adsorbent physically or chemically. Certain solid materials such as silica gel, activated carbon and activated alumina have this property.

adsorption—the adherence of molecules of dissolved substances, gases or liquids to the surfaces with which they are in contact.

aeration—a term generally employed with reference to air circulation or ventilation. In milk cooling a method where the milk flows over refrigerated surfaces exposed to the atmosphere. In ice manufacturing a misnomer for agitation by means of air.

agitator—a device causing turbulent motion in a fluid confined in a tank.

air-agitation system—a combination consisting of a power-driven blower, distributing piping and flexibly connected fittings for delivering air to the water in ice cans for the purpose of agitating the water and promoting the production of clear ice.

air alternator—a device which automatically switches the air from one side of the ice tank to the other.

air blast—forced air circulation.

air changes—the amount of air leakage is sometimes computed by assuming a certain number of air changes per hour for each room, the number of changes assumed being dependent upon the type, use and location of the room.

air circulation—natural or imparted motion of air.

air cleaner—a device used to remove airborne impurities (see air washers and air filters).

air conditioning—the simultaneous control of all or at least the first three of the following factors affecting both the physical and chemical conditions of the atmosphere within a structure; temperature, humidity, motion, distribution, pressure, dust, bacteria, odors, toxic gases and ionization.

air conditioning unit—an assembly of equipment for the control of at least the first three items enumerated in the definition of *air conditioning.*

air cooler, forced-circulation—a factory-made encased assembly of elements by which heat is transferred from air to evaporating refrigerant.

air laterals—pipe which supplies the air to the ice cans contained in the freezing tank.

air mains—pipe which carries air to the air laterals supplying ice cans contained in the freezing tank.

air, saturated—a mixture of dry air and saturated water vapor, both at the same dry-bulb temperature.

air tunnel—a refrigerated tunnel with rapid

air circulation through which the product to be frozen is passed.

air washer—a water-spray system or device for cleaning, humidifying or dehumidifying the air.

air washer and cooler—a vessel designed for passing agitating air through a body of chilled water for the purpose of cleaning and dehumidifying the air delivered to ice cans for agitation of the water therein.

ambient air—generally speaking, the air surrounding an object. In a domestic or commercial refrigerating system having an air-cooled condenser, the temperature of the air entering the condenser.

ammonia machine—shortened expression for a compression refrigerating machine using ammonia as a refrigerant. Similarly, sulfur-dioxide machine, etc.

analyzer—device in high side of absorption system for increasing the concentration of refrigerant in the vapor entering rectifier or condenser.

anemometer—instrument for measuring the velocity of air or other gas.

anticipating control—one which is artificially forced to cut in before it otherwise would, thus starting the cooling (or heating) before needed, to reduce the temperature fluctuation.

apparatus dew point—the dew-point temperature of the air leaving the conditioner unit (i.e., humidifier, dehumidifier, or surface cooler).

aspiration—production of movement in a fluid by suction.

atomize—reduce to fine spray.

attic fan—an exhaust fan to discharge air near the top of a building while cooler air is forced (drawn) in at a lower level.

available energy—the portion of the total energy which can be converted to work in a perfect engine.

back pressure—loose terminology suction pressure of refrigeration vapor in a system.

baffle—partition used to divert fluid flow.

balling—a name applied to the hydrometric scale used in measuring the strength of worts. This scale gives directly the percentage extract dissolved in the wort.

barometer—instrument for measuring atmospheric pressure.

benching machine—portable device for elevating and stacking ice blocks.

bimetallic element—one formed of 2 metals having different coefficients of thermal expansion, used as a temperature control device.

binary vapor cycle—a refrigerating cycle in which two separate refrigerants are used, one superimposed upon and augmenting the cycle of the other.

blanch—scald or sterilize, in case of vegetables to be canned, cooked or frozen, usually by dipping in a hot-water bath.

bleeder—pipe attached, as to a condenser, to lead off liquid refrigerant, parallel to main flow.

blower—enclosed fan device or low-pressure compressor used to impart motion to air.

boiler—a closed vessel in which steam is generated or in which water is heated, also a refrigerant evaporator.

boiling point—the temperature at which a liquid vaporizes upon the addition of heat, dependent upon the absolute pressure at the liquid-vapor surface.

booster—a compressor for very low pressures, usually discharging into the suction line of another compressor.

bore—inside diameter of a cylinder.

brazed—joined by fusion using a spelter on the order of brass. This is considered equivalent to hard soldering.

breaker—a relatively poor conductor of heat used to join the liner and outer shell of an internally refrigerated container such as a refrigerator. A frame making a thermal break around a door or its opening to retard heat flow to the interior of a refrigerator.

brine—any liquid cooled by a refrigerating system and used for the transmission of heat.

brine cooler—evaporator for cooling brine in an indirect system.

brine-spray system—refrigerating scheme for cooling by a mist or spray of brine.

brine tank—in an ice plant, the main freezing tank in which the cans are immersed while ice is being produced. In a brine circulating system, a storage tank or balance tank for brine. In domestic and commercial fields, a container surrounding the evaporator and filled with brine for storing refrigeration or for equalizing temperature at various points of the evaporator, especially in ice cream cabinets.

British thermal unit (Btu)—heat required to produce a temperature rise of 1 degree Fahrenheit in 1 lb of water. Strictly, 1 Btu = 251.996 I.T. calories, where the I.T. calorie is 1/860 kilowatt hour. The mean Btu is 1/180 of the energy required to heat water from 32 deg F to 212 deg F.

bunker—space in refrigerator given to ice or cooling element.

bypass—a pipe or *duct*, usually controlled by valve or *damper*, for conveying a fluid around an element of a system.

"C" factor—thermal conductance—the amount of heat expressed in Btu trans-

mitted in one hour through 1 sq ft of nonhomogeneous material for the thickness or type under consideration for a difference in temperature of 1 deg F between the two surfaces of the material. Conductance is usually used to designate the heat transmitted through such heterogeneous materials as plasterboard, hollow clay tile, etc.

cabinet—refrigerating container per se; mechanical refrigerator without refrigerating machine.

calibration—process of dividing and numbering the scale of an instrument; also of correcting or determining the error of an existing scale, or of evaluating one quantity in terms of readings of another.

calorie—heat required to raise the temperature of 1 gram of water 1 degree centigrade, actually from 4C to 5C. Mean calorie = 1/100 part of the heat required to raise 1 gram of water from 0 to 100 degrees centigrade. Great calorie or kilo calorie = 1,000 small calories.

calorimeter—device for measuring heat quantities, such as machine capacity, heat of combustion, specific heat, vital heat, heat leakage, etc.; also device for measuring quality (or moisture content) of steam or other vapor.

can dump—a device so arranged that ice cans conveyed from the tank may be placed therein approximately vertically, then tipped in order that the ice block may slide out. It may be equipped with sprinkling pipes for spraying the can and thawing the ice block free.

capacitor—a condenser type of motor, or the electrical condenser connected with it.

capacity—of a refrigerating compressor is the refrigerating effect in Btu per hour produced by the change in total enthalpy between the refrigerant liquid at a temperature corresponding to the pressure of the vapor leaving the compressor and the total enthalpy of the refrigerant vapor entering the compressor per hour. Of a condensing unit, the refrigerating effect in Btu per hour produced by the change in total enthalpy of refrigerant liquid leaving the unit and the total enthalpy of the refrigerant vapor entering the unit. Generally measured in tons per day or Btu per hour.

capacity, heat—the amount of heat necessary to raise the temperature of a body one degree. Numerically the mass multiplied by the specific heat.

capacity reducer—in a compressor, a device such as a clearance pocket, movable cylinder head, or suction bypass, by which compressor capacity can be adjusted without otherwise changing the operating conditions.

capillarity—the action by which the surface of a liquid where it is in contact with a solid (as in a slender tube) is raised or lowered.

capillary tube—a vapor lock device comprising a tube of small ID; also a small diameter tube connecting a thermostatic bulb with the bellows or diaphragm of a control device.

carbon dioxide ice—solid CO_2, dry ice.

carbonization—carbon deposits produced by oxidation of the lubricating oil, which may seriously interfere with the proper functioning of the compressor valves.

Carnot cycle—a sequence of operations forming the reversible working cycle of an ideal heat engine of maximum thermal efficiency. It consists of isothermal expansion, adiabatic expansion, isothermal compression and adiabatic compression to the initial state

cascade system—one having two or more refrigerant circuits, each with a pressure-imposing element, condenser and evaporator, where the evaporator of one circuit cools the condenser of the other (lower-temperature) circuit.

carrying freezer—cold storage freeze room, generally kept between −20 F and 20 F, to receive and hold frozen goods.

centigrade—thermometric system in which the freezing point of water is called 0° and its boiling point 100° at normal atmospheric pressure (14.696 psi).

central fan system—a mechanical indirect system of heating, ventilating, or air conditioning, in which the air is treated or handled by equipment located outside the rooms served, usually at a central location, and conveyed to and from the rooms by means of a fan and a system of distributing ducts.

central plant system—a system with two or more low sides connected to a single, central high side, a multiple system.

centifugal compressor—a nonpositive displacement *compressor* which depends for pressure rise, at least in part, on centrifugal effect.

centrifuge—device for separating substances of different densities by centrifugal action.

change of air—introduction of new, cleansed or recirculated air to conditioned space, measured by the number of complete changes per unit time.

change of state—change from one phase, such as solid, liquid, or gas, to another.

charge—amount of refrigerant in a system; charging, putting in charge.

chill—to apply refrigeration moderately, as to meats, without freezing.

chilling room—room where animal carcasses are cooled after dressing prior to cold storage.

chimney effect—the tendency in a duct or other vertical air passage for air to rise when heated owing to its decrease in density.

class of refrigerating system—formerly in extensive use, now becoming obsolete as a result of code change to classification rather than weight; refers to total weight of refrigerant contained; Class A system is one containing 1,000 lb or more of refrigerant. Class B system is one containing more than 100 lb but less than 1,000 lb of refrigerant. Class C system is one containing more than 20 lb but not more than 100 lb of refrigerant. Class D system is one containing more than 6 lb but not more than 20 lb of refrigerant. Class E system is one containing 6 lb or less of refrigerant.

clearance—space in cylinder not occupied by piston at end of compression stroke, or volume of gas remaining in cylinder at same point. Measured in percentage of piston displacement.

clearance pocket—in a compressor, a space of controlled volume to give the effect of greater or less cylinder clearance, thereby changing compressor capacity.

closed cycle—any cycle in which the primary medium is always enclosed and repeats the same sequence of events.

coefficient of expansion—numerical ratio of the increase in length or volume per degree rise in temperature.

coefficient of heat transmission—the amount of heat (Btu) transmitted from fluid to fluid in one hour per square foot of the wall, floor, roof or ceiling for a difference in temperature of 1 deg F between the air on the inside and that on the outside of the wall, floor, roof or ceiling.

coefficient of performance—ratio of the refrigeration produced to the work supplied, refrigeration and work each expressed in the same thermal units, i.e., Btu/Btu; cal/cal.

coil—a cooling or heating element made of pipe or tubing.

cold storage—a trade or process of preserving perishables on a large scale by refrigeration.

comfort air conditioning—the simultaneous control of all, or at least the first three, of the following factors affecting the physical and chemical conditions of the atmosphere within a structure for the purpose of human comfort; temperature, humidity, motion, distribution, dust, bacteria, odors, toxic gases and ionization, most of which affect in greater or lesser degree human health or comfort.

comfort chart—a chart showing effective temperature and humidities under which human comfort is probable; usually drawn on a psychrometric chart base.

comfort cooling—refrigeration for comfort as opposed to refrigeration for storage or manufacture.

comfort line—the effective temperature at which the largest percentage of adults feel comfortable.

comfort zone—(average) the range of effective temperatures over which the majority (50% or more) of adults feel comfortable. (Extreme) The range of effective temperatures over which one or more adults feel comfortable.

commercial refrigerator—reach-in or service refrigerator of commercial size with or without means of refrigeration.

commercial system—refrigeration system used in a commercial and/or a business place, such as a meat market, store, florist shop, hotel, office building, restaurant, candy shop, bakery, or other places of similar commercial enterprise, assembled and installed in the manufacturing and/or business portion of any building (code).

compression, compound—compression by stages in two or more cylinders.

compression, single-stage—compression in one stage.

compression system—refrigerating system in which the pressure imposing element is mechanically operated.

compressor, compound—a compressor in which compression is accomplished by stages, as in two or more cylinders.

compressor, double-acting—one which has two compression strokes per revolution of the crank shaft per cylinder, i.e., both faces of the piston are working faces.

compressor, refrigerant—a specific machine for compressing a given refrigerant vapor from an evaporating system and discharging it into a condensing system (code). (2) That part of a mechanical refrigerating system which receives the refrigerant vapor at low pressure and compresses it into a lower volume at higher pressure.

compressor, single-acting—one having one compression stroke per revolution of the crank for each cylinder.

compressor unit—a condensing unit less the condenser and liquid receiver.

concentration—a number specifying the composition of a solution with respect to

the constituent named, as pounds of salt per gallon of brine.

condenser, atmospheric—condenser operated with water which is exposed to the atmosphere.

condenser (liquefier)—a vessel or arrangement of pipe or tubing in which the vaporized (and compressed) refrigerant is liquefied by removal of heat.

condenser, water-cooled refrigerant—an assembly of elements by which the flows of refrigerant vapor and water are maintained in such a heat transfer relationship that the refrigerant vapor is condensed into a liquid and the water is heated.

condensing refrigeration effect—condensing heat rejection effect, less the heat added to the refrigerant vapor in the refrigerant compressor.

condensing unit—a specific refrigerating machine combination for a given refrigerant, consisting of one or more power-driven compressors, condensers, liquid receivers (when required), and the regularly-furnished accessories.

conductance, thermal—of a body per unit area between two isothermal surfaces is the rate of heat flow through the body per degree difference between the temperatures of the isothermal surfaces.

conduction—heat transmission, as by particle to particle in a homogeneous substance, or by contact of two substances.

conductor (heat)—a material capable of readily conducting heat. The opposite of an insulator or insulation.

connecting rod—a device connecting the piston to a crank and used to change rotating motion into reciprocating motion, or vice versa, as from rotating crankshaft to reciprocating piston.

conservation of energy—see first law of thermodynamics.

control—any device for regulation of a machine in normal operation, manual or automatic. If automatic, the implication is that it is responsive to changes of temperature, pressure, liquid level or with time.

controlled atmosphere storage—(gas storage) artificial addition of carbon dioxide to the atmosphere, particularly in large concencentration, with no attempt to regulate the amount of oxygen.

convection—transfer of heat by movement of fluid containing thermal energy.

convection, natural—circulation of gas or liquid (usually air or water) due to its own thermal expansion; stack effect. In water there is a zone of reverse flow due to thermal contraction as water is warmed from 32 to 39.2 degrees fahrenheit, the maximum density point of water being 4 degrees centrigrade or 39.2 degrees fahrenheit.

convector—a heat transfer surface designed to transfer its heat to surrounding air largely or wholly by convection. Such a surface may or may not be enclosed or concealed. When concealed and enclosed, the resulting device is sometimes referred to as a concealed radiator.

cooler—in cold storage practice, an insulated room maintained at 30 F or above.

or in commercial systems, a mechanical device (i.e., not a cool space or box), as *unit cooler* or *room cooler*, usually called a cooling element.

or cooling unit—specific air treating combination consisting of means for air circulation and cooling.

cooling (or heating) air conditioning unit—a specific air treating combination consisting of means for ventilation, air circulation, air cleaning and heat transfer, with control means for cooling (or heating) and maintaining humidity.

cooling of air—reduction in air temperature due to the abstraction of sensible heat as a result of contact with a medium held at a temperature lower than that of the air. Cooling may be accompanied by moisture addition (evaporation), by moisture extraction (dehumidification) or by no change whatever of moisture content. Moisture change, if present, is considered a secondary or by-product effect. The medium may be directly in contact with the air (as water, brine, or ice) or indirectly through a barrier wall (as cooling surface). When the latter method is used, and the surface temperature is held above the air dew point, only cooling occurs without moisture inter-change.

cooling element—heat transfer surface containing refrigerating fluid in location where refrigerating effect is desired.

cooling tower—device for lowering the temperature of water by evaporative cooling, in which the water is showered through a space through which outside air circulates.

cooling water—water used for condensation of refrigerant; condenser water.

copper plating—a film of copper deposited by electrical, immersion or other means on the surface of another material such as iron or steel. In refrigeration usually on compressor walls, pistons, discharge valves and shaft or seal.

core area—the total plane area of the portion of a grill, face, or register, bounded by a line tangent to the outer edges of the outer openings through which air can pass.

core sucking and filling system—a combina-

tion consisting of a pump, distributing piping, hose, and device by which water containing impurities may be withdrawn from the unfrozen core of an ice block, and by means of connection to the source of water supply the core may be refilled with pure water.

corkboard (insulation)—cork granules, cleaned, compressed and bonded by heat to a weight of about 0.65 lb per board foot.

corresponding values—simultaneous values of various properties of a fluid, such as pressure, volume, temperature, etc., for a given condition of fluid.

corrosive—having rusting or chemically destructive effect on metals (occasionally on other materials).

counterflow—in heat exchange between two fluids, opposite direction of flow, coldest portion of one meeting coldest portion of the other.

critical point—of a substance, state point at which liquid and vapor have identical properties; critical temperature, pressure and volume are those properties at the critical point. Above the critical temperature or pressure there is no line of demarcation between liquid and gaseous phases.

cryohydrate—a frozen mixture of water and a salt; a brine mixed in eutectic proportions to give the lowest freezing point.

crystal formation, zone of maximum—temperature range in freezing in which most of the freezing takes place (about 25 to 30 F for water).

cutting room (locker plants)—cold room where animal carcasses are cut up into commercial sizes such as rib roast, legs, etc.

cycle—complete course of operation of refrigerant back to a starting point, measured in thermodynamic terms (functions); also used in general for any repeated process or any system.

Dalton's Law of partial pressure—each constituent of a mixture of gases behaves thermodynamically as if it alone occupied the space. The sum of the individual pressures of the constituents equals the total pressure of the mixture.

damper—a valve, or plate, used to regulate the flow of air or other fluid.

damper (multiple louvre)—a damper having a number of adjustable blades.

decibel—a unit commonly used for expressing sound or noise intensities referred to an arbitrary reference level. It is defined as *db difference* $= 10 \log_{10}(P_1/P_0)$, where P_1 is the unknown intensity. P_0 the reference level is commonly taken as 10^{-16} watts per sq centimeter which corresponds to the threshold of hearing at 1,000 cycles per second.

deck, coil deck—insulated horizontal partition between refrigerated space and bunker.

decomposition—process of chemical change, breaking up of structures, spoilage.

defrosting—removal of accumulated ice from cooling element, thawing frozen foods.

defrosting cycle—a refrigeration cycle which permits cooling unit to defrost during off period.

degree-day—degrees × days; a unit, based upon temperature differences and time, used in specifying the nominal heating load in winter. For any one day there exist as many degree-days as there are degrees Fahrenheit difference in temperature between the average outside air temperature, taken over 24-hour period, and a temperature of 65 F.

dehumidification—the condensation of water vapor from air by cooling below the dew point or removal of water vapor from air by chemical or physical methods.

dehumidifier—an air cooler or washer used for lowering the moisture content of the air passing through it; an absorption or adsorption device for removing moisture from air.

dehumidifier, surface—an air conditioning unit, designed primarily for cooling and dehumidifying air through the action of passing the air over wet cooling coils.

dehumidifying effect—of a unit in Btu per hr; the difference between the moisture content, in pounds per hour, of the entering and leaving air multiplied by 1,060.

dehydration—the removal f water vapor from air by the use of absorbing or adsorbing materials. (2) The removal of water from stored goods.

dehydrator—a device used to remove water or water vapor from refrigerant.

density—mass or weight per unit of volume.

dense-air system—cold-air system maintained under pressure greater than atmospheric in which air is compressed, heat of compression dissipated, and the air, chilled by expansion and performance of work, can create useful refrigeration.

desiccation—drying by evaporation, generally due to low humidity.

design working pressure—the maximum allowable working pressure for which a specific part of a system is designed.

dew point—temperature at which condensation starts if moist air is cooled at constant pressure with no loss or gain of moisture during the cooling process.

dew-point rise—increase in moisture content (specific humidity) of air expressed in terms of rise in dew-point temperature.

diesel engine—a power plant used to change thermal energy to mechanical energy using the heat of compression to ignite the fuel.

differential (of a control)—the difference between cut-in and cut-out temperatures or pressu es.

diffusers, air—see grilles.

diffusion temperature difference—the temperature difference between the air temperature at supply opening and design indoor temperature.

dip tanks—a tank located convenient to the can dump and supplied with water in which the ice can is immersed to thaw the ice block loose f om the can.

direct connected—driver and driven, as motor and compressor, positively connected in line to operate at same speed.

direct expansion coils—coils using the direct method of refrigeration.

direct method of refrigeration—a system in which the evaporator is in direct contact with the material or space refrigerated or is located in air circulating passages communicating with such spaces.

displacement—(1) volume swept by piston per stroke. (2) Actual—the actual volume of gas at compressor inlet conditions moved in a given time, or (3) theoretical—the total volume displaced by all the pistons of a compressor per stroke in a definite time interval. Usually measured in cubic feet per minute.

distributor—device for dividing flow of liquid fluid between parallel paths in an evaporator, or in other types of heat transfer apparatus.

domestic refrigerator—refrigerator or mechanical refrigerator for a home, usually of size of 16 cu ft or less.

double-pipe heat exchanger—one in which two pipes are arranged concentrically, one within the other, and in which one fluid flows through the inner pipe and the other through the annulus between them.

double suction—split suction, valving arrangement on compressors for carrying two suction pressures, see multiple effect.

drier—a chemical compound capable of adsorbing, or reacting chemically with the moisture contained in the liquid or gaseous refrigerant-oil mixture.

drip—water which appears on thawing frozen food, water melting from evaporator, water dropping from a cooling surface.

dry air—air without its contained water vapor; air only.

dry-air cooler—removes sensible heat from the dehydrated air whenever it leaves the dehydrator at an elevated temperature.

dry bulb—temperature, temperature by ordinary thermometer (term used only to distinguish from wet-bulb temperature).

dry ice—solid carbon dioxide, CO_2 (proprietary term).

dry tons—the sensible heat load expressed in tons of refrigeration.

dual compression—see multiple effect.

dual effect control—one responsive to temperatures of two zones, or to two variable conditions.

dual-temperature brine system—in chilling beef the use of an initial brine temperature, followed by the lower brine temperature.

duct—a conduit or tube used for conveying air or other gas.

duct system—a series of ducts, elbows and connectors to convey air from one location to another.

dust—solid material in a finely divided state, the particles of which are large and heavy enough to fall with increasing velocity due to gravity in still air. For instance, particles of fine sand or grit, the average diameter of which is approximately 0.01 centimeter, such as are blown on a windy day, may be called dust.

dynamometer—device for applying and measuring power developed by an engine or motor.

ebullator—a device inserted in flooded evaporator tubes to prevent the evaporator from becoming oil bound or refrigerant liquid becoming quiescent at a pressure lower than its boiling point.

effective temperature difference—the difference between the room air temperature and the supply air temperature at the outlet to the room.

effective work—net mechanical energy required by, or load imparted to, piston of compressor.

efficiency, compression—ratio of hp required to compress adiabatically and reversibly all the vapor delivered by compressor (in single stage) to the ihp.

efficiency, mechanical—ratio of the compression energy or work of a compressor to the energy or work input.

efficiency, volumetric—(1) *Total*—ratio of the actual volume of gas moved by the compressor or pump to the actual displacement of the compressor or pump. (2) *Apparent*-ratio of length on indicator card of suction line to stroke. (3) *Due to cylinder heating* (also called real or no-clearance

volumetric efficiency)-ratio of the total to the apparent volumetric efficiency.

ejector—a device which builds up a high fluid velocity in a restricted area to obtain a lower static pressure at that point so that fluid from another source may be drawn in.

electric precipitators—devices for removing dust from the air by means of electric charges induced on the dust particles.

electric refrigerator—refrigerator, cooled by a system, operated by electric motor as distinct from an engine-operated, or a gas-operated or brine-cooled refrigerator.

electrical degree—the 360th part of the angle subtended at the axis of the machine, by two consecutive field poles of like polarity. One mechanical degree is thus equal to as many electrical degrees as there are pairs of poles in the machine.

emissivity—characteristic of a surface for giving off heat by radiation. Almost equal to the absorptivity. Ranges in value from zero for a perfect reflector to unity for a perfect black body.

emulsification—formation of an emulsion, i.e., a mixture of small droplets of two or more liquids which do not dissolve in each other.

enclosed vertical compressor—machine with vertical cylinder and enclosed crank, generally single-acting.

engine—prime mover, device for transforming fuel or heat energy into mechanical energy.

enthalpy—a thermodynamic property of a substance defined $h = u + Pv/J$ where u = internal energy, customarily in Btu per lb; P = pressure in lb per sq ft; and v is specific volume in cu ft per lb. $J = 778$ ft lb per Btu. Also called *total heat* or *heat content*.

entropy—entropy is the ratio of the heat added to a substance to the absolute temperature at which it is added for a reversible process. It is a thermodynamic property which, for practical purposes, is best defined by stating its principal functions: (1) during a reversible adiabatic change of state, entropy is constant; (2) during a reversible isothermal change of state, the heat absorbed is equal to absolute temperature times change of entropy.

enzyme—complex organic substances such as diastase, pepsin, etc., capable of transforming by catalytic action some other compound; a soluble ferment.

equalizer—piping arrangement on vertical enclosed compressors to equalize refrigerant gas pressure in crank case and suction; device for dividing the liquid refrigerant between parallel low side coils.

equivalent evaporation—the amount of water a boiler would evaporate, in pounds per hour, if it received and vaporized feed water at 212 F and atmospheric pressure as compared to actual evaporation.

eutectic mixture or solution—a mixture which melts or freezes completely at constant temperature and with constant composition. Its melting point is the lowest possible for mixtures of the given substances.

evaporation—change of state from liquid to vapor.

evaporative condenser—condenser in which heat absorbed by water is largely dissipated as latent heat by evaporation of the water into an unsaturated air stream in contact with it.

evaporative cooling—involves the adiabatic exchange of heat between air and a water spray or wetted surface. The water assumes the wet-bulb temperature of the air, which remains constant during its traverse of the exchanger.

evaporative equilibrium—of a wet-bulb instrument—the condition attained when the wetted wick has reached a stable and constant temperature (when the instrument is exposed to air at velocities over 900 fpm this temperature may be considered to approach the true wet-bulb temperature).

evaporator—that part of a system in which refrigerant is vaporized to produce refrigeration.

evaporator, dry-type—an evaporator of the continuous tube type where refrigerant from a pressure-reducing device is fed into one end and the suction line connects to the outlet end.

exfiltration—air flow outward through a wall, leak, membrane, etc.

exhaust opening—any opening through which air is removed from a space which is being heated or cooled, or humidified or dehumidified, or ventilated.

expansion coil—an evaporator constructed of pipe or tubing.

extended surface—heat transfer surface one side of which is increased in area by the addition of fins, discs or other means.

external equalizer—in a thermostatic expansion valve, a tube connection from the chamber containing the evaporation pressure actuated element of the valve to the outlet or the evaporator coil. A device to compensate for excessive pressure drop through the coil.

extruded—pushed out through a die. Bars of ice, metal rods, shapes and tubes are made by this method.

f—film or surface conductance, the amount of heat expressed in Btu transmitted by radiation, conduction, convection from a surface to the air surrounding it, or vice versa, in one hour per square foot of the surface for a difference in temperature of 1 deg F between the surface and the surrounding air. To differentiate between inside and outside wall (or floor roof or ceiling) surfaces, f_i is used to designate the inside film or surface conductance and f_o the outside film or surface conductance. The letter *f* is also used for the friction factor in fluid flow.

factor of safety—ratio of ultimate stress to designed working stress.

fahrenheit—a thermometric system in which 32° denotes freezing and 212° the boiling point of water under normal pressure at sea level (14.696 psi).

fan—strictly, device for imparting motion to air. Any device for imparting motion to air by means other than positive displacement.

fan, attic—fan, mounte l in roof or side wall for exhausting air from attic area.

fan economizers—devices which prevent the operation of the fan motor on a cold diffuser during the shut-down period after the coil has been defrosted.

fan exhaust—fan used to remove or exhaust air from a space.

filter—a device to remove solid material from a fluid.

filter press—a device for separating solid and liquid matter under pressure so that the solid residue is compressed into briquettes to facilitate removal.

fin—an extended surface to increase the heat transfer area, as metal sheets attached to tubes.

finned tubes—heat transfer tube or pipe with extended surface in the form of fins, discs or ribs.

first and second laws—of thermodynamics—first law states, in effect, the conservation of energy principle, particularly equating heat and mechanical energy and denying perpetual motion, insofar as it implies a creation of energy. Second law states that the quality of energy varies as to form, particularly that heat energy is only in part transformable into mechanical energy; it denies the possibility of a machine operating in a cycle and developing mechanical energy from a single source of heat.

flammability—the ability of a material to burn.

flammable refrigerant—any refrigerant which will burn when mixed with air, such as ethyl chloride, methyl chloride and the hydrocarbons.

flare fitting—a type of soft-tube connector which involves the flaring of the tube to provide a mechanical seal.

flash gas—the gas resulting from the instantaneous evaporation of refrigerant in a pressure-reducing device to cool the refrigerant to the evaporation temperature obtaining at the reduced pressure.

flash chamber—separating tank placed between the expansion valve and evaporator in a refrigeration system to separate and by-pass any flash gas formed in the expansion valve.

flash point—temperature of combustible material, as oil, at which there is a sufficient vaporization to igni e the vapor, but not sufficient vaporization to support combustion of the material.

flooded system—system in which only part of the refrigerant passing over the heat transfer surface is evaporated, and the portion not evaporated is separated from the vapor and recirculated.

flotation—a method of treating materials by floating in a liquid. An ice-making method in which ice floats away from the surface on which it has been frozen.

fluid—gas, vapor or liquid.

foaming—formation of a foam or froth of oil-refrigerant due to rapid boiling out of the refrigerant dissolved in the oil when the pressure is suddenly reduced. This occurs when the compressor starts operating and, if large quantities of refrigerant have been dissolved, large quantities of oil may "boil" out and be carried through the refrigerant lines.

forecooler—in an ice plant, device for cooling the water for ice-making before its enters the cans, precooler.

foul gas—see non-condensible gas.

free area—the total minimum area of the openings in a grille, face, or register through which air can pass.

free delivery type unit—a device which takes in air and discharges it directly to the space to be treated without external elements which impose air resistance.

freezer—in cold storage, an insulated room kept below 30 F—see (a) carrying freezer, (b) sharp freezer. Any device for freezing perishables.

freeze up—failure of a refrigeration unit to operate normally due to formation of ice at the expansion valve. The valve may be frozen shut or open, causing improper refrigeration in either case.

freezing point—temperature at which given liquid substance will solidify or freeze upon removal of heat. Ice point for water is 32 F.

freezing tank—a container kept at low temperatures for storing foodstuffs or for freezing foodstuffs.

freezing time—time for any complete freezing process to take place.

Freon-12—the trade name for dichlorodifluoromethane (CCl_2F_2), also called F-12.

frost back—the flooding of liquid from an evaporator into the suction line accompanied by frost formation on suction line in most cases.

frozen food—any food in which the contained water is in the solid form.

fusible plug—a device having a predetermined melting temperature member for the relief of pressure.

gage (gauge)—instrument for measuring pressure or liquid level. Also arbitrary scale of measurement for sheet metal thickness, wire and drill diameters, etc.

gage glass—device for showing a liquid level.

gas constant—the coefficient R in the perfect gas equation $pv = RT$.

gas refrigerator—refrigerator motivated by thermal energy of burning gas.

gaseous state of matter—one of three states characterized by greatest freedom of molecules and lack of any inherent fixed shape or volume.

generator—basic part of absorption system; a still provided with means of heating used to drive ammonia out of water.

glazing—of foods, freezing a coat of ice on frozen food by dipping in water, the latent heat of the coat being absorbed inwards.

graduated acting—term applied to a control instrument or device which functions to give throttling control, that is, operates between full on and full off position.

grid—a device for lifting a row of ice cans.

grille—a lattice or grating for delivery or intake opening of an air passage.

head—in hydraulics, the height of a column of fluid necessary to produce existing pressure at a point in the system. Also velocity head the pressure equivalent to fluid velocity.

head pressure—*operating pressure* measured in the discharge line at the outlet from the *compressor*.

heat of condensation—latent heat given up in changing from gas to liquid.

heat exchanger—apparatus in which heat is exchanged from one fluid to another through a partition.

heat, latent—heat characterized by a change of state of the substance concerned, for a given pressure and always at a constant temperature for a pure substance, i.e., heat of vaporization or of fusion.

heat of the liquid—enthalpy of a mass of liquid above an arbitrary zero.

heat of reaction—heat per unit mass or per mole of one of the reagents or products of reaction in a chemical reaction; exothermal if given off, endothermal if absorbed.

heat pump—a refrigerating system employed to transfer heat into a space or substance. The condenser provides the heat while the evaporator is arranged to pick up heat from air, water, etc. By shifting the flow of air or other fluid a heat pump system may also be used to cool the space.

heat rejection effect, condensing—that portion of the total refrigerant heat rejecting effect of a condenser which is used for condensing the entering refrigerant vapor to a saturated liquid at the entering refrigerant pressure.

heat rejection effect, total refrigerant—total useful capacity of a refrigerant condenser for removing heat from the refrigerant circulated through it.

heat transmission—any flow of heat; usually refers to conduction, convection and radiation combined.

hermetically-sealed unit—a sealed (hermetic-type) condensing unit is a mechanical condensing unit in which the compressor and compressor motor are enclosed in the same housing, with no external shaft or shaft seal, the compressor motor operating in the refrigerant atmosphere. The compressor and compressor motor housing may be of either the fully-welded or brazed type, or of the service-sealed type. In the fully-welded or brazed type, the housing is permanently sealed and is not provided with means of access for servicing internal parts in the field. In the service-sealed type, the housing is provided with some means of access for servicing internal parts in the field.

high side—parts of a refrigerating system under condenser pressure or higher.

hold-over—in an evaporator, the ability to stay cold after heat removal from the evaporator stops. A material used to store heat in latent or sensible form.

horizontal compressor—machine with horizontal cylinder, generally double-acting.

horsepower—unit of power in foot-pound-second system, work done at the rate of 550 ft lb per sec, or 33,000 ft lb per min.

hot-gas defrosting—use of high pressure or condenser gas in the evaporator or low side to effect removal of frost.

humidifying effect of a unit in Btu per hr—the latent heat of vaporization of water at the average evaporating temperature times the number of pounds of water evaporated per hour.

humid heat—increase of enthalpy per pound of dry air with its associated moisture under conditions of constant pressure and constant specific humidity.

humidifier—a device to add moisture to air.

humidifier—central—a device which humidifies air to be circulated through ducts in an air-conditioning system.

humidifier—room-spray type—an air humidifier which sprays water directly into the room.

humidify—to add water vapor to the atmosphere; to add water vapor or moisture to any material.

humidistat—control device responsive to atmospheric relative humidity, hygrostat.

humidity, absolute—concentration of water vapor in unit volume, as grains of H_2O per cu ft or per cu ft of air.

humidity ratio—see specific humidity.

humidity, relative—the ratio of the partial pressure of water vapor in the air to the pressure of saturated water vapor at the temperature of the air. This is closely equivalent at temperatures below 150 F to a more rigorous definition—the ratio of the density of water vapor in the air to the density of saturated water vapor at the air temperature.

humidity, specific—the weight of water vapor (steam) associated with one pound weight of dry air, sometimes called *humidity ratio.*

humidity, percentage—the ratio of the weight of water vapor mixed with a pound of dry air to the weight of saturated water vapor at the same temperature mixed with a pound of dry air, expressed in per cent. This ratio of specific humidities is also called *saturation ratio.*

hydrator—a covered container or compartment in a refrigerator having a higher relative humidity than the balance of the storage space.

hydrolysis—reaction of a material such as Freon-12 or methyl chloride with water. Acid materials in general are formed.

hydrometer—an instrument which, by the extent of its submergence, indicates the specific gravity of the liquid in which it floats.

hydrostatic pressure—the pressure due to the vertical height of a liquid column.

hygrometer—instrument responsive to humidity conditions (usually relative humidity) of the atmosphere.

hygroscopic—absorptive of moisture, readily absorbing and retaining moisture.

hygrostat—automatic control responsive to humidity.

ice box—refrigerator using ice as the primary cooling medium.

ice can—an open content top tank or can of properly formed and reinforced sheet metal which when submerged in a chilled anti-freeze solution in an ice tank will permit the freezing of its water into a block of ice of a desired form or weight.

ice-can frames—a metal frame or support in which a number of ice cans are fixed so as to enable them to be handled as a group rather than handling each can individually.

ice-can truck—a wheeled frame with arm and hoist suitably arranged to be moved over the ice-tank top for lifting individual cans from the tank.

ice crane, hoist and runway—a device consisting of a crane beam, with suitable carriage, carrying a movable hoist for lifting the ice cans or ice blocks, individually or in groups, and running on suitably supported runway rails all for the purpose of lifting the cans or ice blocks from the tank and carrying them to a point beyond the tank for disposal.

ice-making capacity—actual productive ability of a system making ice. This is less than the rated (ice-melting) capacity as some refrigeration is used in cooling the water to freezing point, cooling the ice below the freezing point, and overcoming heat leakage.

ice-making tank—a tank arranged with proper accessories to hold an evaporator and anti-freeze solution and ice cans, the cans being immersed in the solution which usually is in circulation around the evaporator and the cans.

ice-melting capacity—1. Refrigeration equal to the latent heat of fusion of a stated weight of ice at 144 Btu per lb. 2. Ice-melting equivalent—the amount of heat absorbed by one pound of ice at 32 F in liquefying to water at 32 F, 144 Btu.

ice point—temperature at which water freezes under normal atmospheric pressure (14.696 psia), +32 F, 0 C.

ice-tank grating and covers—a framework of wood, metal or combination of each, which positions the ice cans in the ice-making tank and upon which are supported removable covers, usually of wood, above the cans excluding dirt and the warmer air of the tank room.

indicated work—work equivalent of indicator card area for a reciprocating compressor or engine.

indicator diagram—pressure-volume diagram tracing condition of gas in a compressor or engine cycle, in terms of p and v; indicator card.

indirect method of refrigeration (or cooling)—a system in which a liquid, such as brine or water, cooled by the refrigerant, is circulated to the material or space refrigerated or is used to cool air so circulated.

induction units—high-pressure type—employ nozzles which produce a high velocity jet. The high velocity jet of primary air induces a flow of secondary air through coils located in the secondary air stream.

induction units—low-pressure type—essentially induction type convectors. They use a jet of conditioned air (or primary air) to induce into the unit a flow of room or secondary air which mixes with the primary air. The mixture is discharged into the room through a grille at the top of the unit. Heating coils are located in the secondary air stream for use in heating.

industrial air conditioning—air conditioning for other uses than comfort.

industrial system—a system used in the manufacture or processing of materials, such as ice-making plants, cold storage warehouses, ice cream plants, dairy plants, packing houses, chemical plants, and other places of similar industrial enterprise.

inert gas—one that experiences no change of state in a system or process, such as air, nitrogen, helium, etc., mixed with a volatile refrigerant.

infiltration—air flowing *inward* as through a wall, leak, etc.

inlet—suction, low side-pipe to, or port of compressor.

insulation, fill—granulated, shredded, or powdered materials, prepared from vegetable, animal, or mineral origin. It can come in bulk or batt form.

insulation, sound—acoustical treatment of fan housings, supply ducts, and other parts of system and equipment for isolating of vibration.

intercooling—removal of heat from compressed gas between compression stages.

internal (intrinsic) energy—the sum of all the kinetic and potential energies contained in a substance due to the states of motion and separation of its several molecules, atoms and electrons. It includes sensible heat (vibration energy) and that part of the latent heat that is represented by the increase in energy during evaporation.

irradiation—subjecting foods, etc., to radiations of special wavelengths, such as the 0.2537 micron wavelength, which kills certain bacteria.

is-, iso- —prefix meaning constant; as isothermal, constant temperature; isentropic, constant entropy; isobaric, constant pressure; etc., and in chemicals one having different characteristics but with the same number and kind of atoms.

jacket water—in a compressor, the water used for cooling the cylinder head and/or walls.

jacketing—surrounding by a confined bath or stream of fluid for temperature control or heat absorption.

Joule-Thomson effect—dT/dp of an actual gas in a process of throttling or expansion without doing work or interchanging heat.

"k" factor—thermal conductivity; the amount of heat expressed in Btu transmitted in one hour through 1 sq ft of a homogeneous material 1 in. thick for a difference in temperature of 1 deg F between the two sides of the material. The conductivity of any material depends on the structure of the material and its density. Heavy or dense materials, the weight of which per cubic foot is high, usually transmit more heat than light or less dense materials, the weight of which per cubic foot is low.

lag of temperature control—the delay in action of a temperature responsive element due to time required for the element to reach surrounding temperature.

latent heat—change of enthalpy during a change of state usually expressed in Btu per lb. With pure substances, latent heat is absorbed or rejected at constant temperature at any pressure.

leak detector—1. A device used to detect refrigerant leaks in a refrigerating system, or 2. Halide torch—a flame tester generally using alcohol and burning with a blue flame. When the sampling tube draws in Freon vapor, color of flame changes to bright green.

line—mainly used in reference to a line on characteristic psychrometric chart or indicator—liquid, pipe conveying liquid, etc.

liquefaction—change of state to liquid, generally used instead of condensation in case of substances ordinarily gaseous.

liquid line—the tube or pipe carrying the refrigerant liquid from the condenser or receiver of a refrigerating system to a pressure-reducing device.

liquid receiver—a vessel permanently connected to a system by inlet and outlet pipes for the storage of (liquefied) refrigerant.

liquid state of matter—one of three states characterized by limited freedom of molecules and by substantial incompressibility.

liquor—a solution in refrigeration, generally of ammonia and water.

load—the amount of heat per unit time imposed on a refrigerating system, or the required rate of heat removal.

load factor—ratio of actual mean load to a maximum load or maximum production capacity in a given period.

locker plants—cold storage establishments containing food storage boxes or lockers for individual users.

louvres—sloping overlapping boards or metal plates intended to permit ventilation and shed falling water.

low-pressure control—a thermostat (usually switch) actuated directly by refrigerant vapor pressure in the low side.

low-pressure system—in air conditioning, a distributing system delivering air to ordinary ventilating grilles at low velocities with low static losses through the supply grilles.

low side—parts of a refrigerating system under evaporator pressure.

low temperature—temperature below ordinary refrigerating plant requirements—say, from −30 F on down; also any part of a system below another parallel refrigerating level of temperature.

lyophilization—the process of dehydrating a frozen substance under conditions of sublimation.

machinery room—a room in which is permanently installed and operated a refrigerating system but not including evaporators located in a cold storage room refrigerator box, air-cooled space, or other enclosed space. Closets solely contained within and opening only into a room shall be considered a part of such room and not as a machinery room.

main—pipe for distributing to or collecting from various branches.

makeup water—water supplied to replenish as water replacing that lost by evaporation.

manifold—portion of main in which several branches are close together. Also, single piece in which there are several fluid paths.

manometer—U-tube for measuring pressure differences.

mean temperature difference—mean of difference between temperatures of a fluid receiving and a fluid yielding heat.

mechanical condensing unit—a complete high side of a refrigeration system including the motor in a unit assembly.

mechanical equivalent of heat—an energy conversion ratio of 778.16 foot pounds = 1 Btu.

mechanical draft cooling tower—cooling tower through which air is forced or induced by fans or other mechanical means.

melting—change of state from solid to liquid.

melting point—for a given pressure, the temperature above which solid state cannot exist.

metabolism—chemical changes in living cells by which energy is provided for vital processes.

meter—unit of length = 39.37 inches. Also instrument for measuring rates or integrating rates over a period of time.

micron—a unit of length, the thousandth part of one millimeter or the millionth of a meter.

modulating—of a control, tending to adjust by increments and decrements, also one modified by variation of a second condition.

moisture tons—latent heat load expressed in tons of refrigeration.

mole (mol)—weight of a substance equal, in the weight units used, to its molecular weight; as a gram-mole; pound mole, etc.

motive power—term used to express the source from which a device such as a diaphragm valve, relay or motor obtains its power (electric or fluid).

motor, air—an air-operated device which is used primarily for opening or closing dampers.

multiple can filler—an elevated tank having a compartment for each ice can to be filled. Each compartment is provided with adjusting devices so that it will be filled with a predetermined weight of water. The water is transferred to the ice cans through valved pipes.

multiple system—a system using the direct method in which refrigerant is delivered to two or more evaporators in separate rooms or refrigerators.

multi-stage compression—in 2 or more steps, as where the discharge of one compressor is connected with the suction of another.

multi-stage expansion—passing volatile refrigerant through 2 or more pressure-reducing devices, connected in series, usually with an evaporator between them operating at one pressure and a second evaporator fed through both devices, at a lower pressure.

natural-draft cooling tower—a cooling tower in which air is circulated by natural convection.

non-condensible gas—gas in a refrigerating system which does not condense at the temperature and partial pressure at which it exists in the condenser, and therefore imposes a higher head pressure on the system.

non-priming cooler—tubes are omitted from the top segment of the shell leaving a gas space above the tubes about equal to ¼ the inside shell diameter.

off-peak system—one with control which normally avoids use of power during peak load periods, usually having eutectic or water-ice hold-over means.

oil cooler—heat exchanger for the purpose of cooling oil in a lubrication system.

oil separator—device for preventing oil from entering the evaporator.

open shell-and-tube condenser—one in which the water passes in a film over the inner surfaces of the tubes, which are open to the atmosphere.

open-type compressor (or system)—A compressor with a shaft or other moving part extending through a casing to be driven by an outside source of power, thus requiring a stuffing box, shaft seal or equivalent rubbing contact between a fixed and moving part.

organic compound—originally a chemical compound produced by a life process. Now generally understood to include all compounds containing carbon.

outlets, ceiling—outlets installed in ceiling in order that a blanket of cool air be introduced at the top of the room, gradually settling evenly over the occupied zone.

outlets slotted—flat metal plates containing a number of long narrow slots or a single long narrow slot.

outlets, vaned—equipped with vertical and/or horizontal adjustable vanes.

output—capacity, duty, performance, net refrigeration produced by system, equal to load imposed. Motor output is electrical power input times motor efficiency.

outside air—external air, atmosphere exterior to refrigerated or conditioned space, ambient (surrounding air).

outside air opening—any opening used as an entry for air from outdoors.

over-run—in ice cream freezing, the ratio of the volume of ice cream to the volume of the mix used.

ozone—triatomic oxygen, O_3, sometimes used in air conditioning or cold storage as an odor eliminator, somewhat toxic.

packing—the stuffing around a shaft or valve stem to prevent fluid leakage.

packing plant—an establishment engaged in the slaughtering, dressing and processing of animals. Also used in connection with the processing of vegetables or fish.

parallel, connection in—system whereby flow is divided among 2 or more channels, from a common starting point or header.

pasteurization—heat treatment, usually at 131–158 F, for killing bacteria as in milk without greatly changing its chemical composition.

performance factor—the ratio of heat moved by a refrigerating system to the heat equivalent of the energy used.

phase—in thermodynamics one of the states of matter, as solid, liquid, gaseous; electrical, an alternating current whose alternations have a definite time relation to the rotational position of the alternator. In a polyphase machine, the phases are separated by 360 electrical degrees divided by the number of phases.

phosphorous pentoxide—an efficient drier material which becomes gummy on reacting with moisture and hence is not used alone as a drying agent in refrigerating systems.

pipe line refrigeration—service to group of buildings with a refrigerant supply from a central refrigerating plant.

plenum—chamber under pressure for receiving air before delivery to conditioned space or combustion system.

pneumatic—operated by air pressure.

polytropic-change—any set of changes in a gas represented by the equation $pV^n = \text{constant}$.

positive displacement compressor—a compressor in which increase in vapor pressure is attained by changing the internal volume of the compression chamber.

potentiometer—an instrument for measuring or comparing small electromotive forces such as those produced by thermocouples.

power—the rate of doing work usually expressed in horsepower, kilowatts, etc.

power consumption—power used multiplied by time, measured in kw hr, hp hr, etc. (Power is the rate of work; power consumption is work.)

precooler—cooler for the removal of sensible heat before shipping, storing, or processing; device for cooling a fluid before it enters some piece of apparatus.

precooling—refers to practice of preparing fruits and vegetables for shipment by cooling before or after they enter car.

preheating—in air conditioning to heat the air in advance of other processes.

pressure—(1) thermodynamically the normal force exerted by a homogeneous liquid or gas, per unit of area, on wall of the container. (2) Absolute—pressure referred to that of a perfect vacuum. It is the sum of gage pressure and atmospheric pressure. (3) Atmospheric—pressure exerted by atmosphere in all directions as indicated by a barometer. Standard atmospheric pressure is a pressure of 76 cm of mercury at 32 F. It is equivalent to 14.6959 psi.

(4) Back—or suction—pressure in suction line of evaporator. (5) Balance—pressure in a system or container equal to that which exists outside. (6) Critical—vapor pressure corresponding to the critical temperature. (7) Gage—pressure above atmospheric. (8) Partial—portion of total gas pressure of a mixture attributable to one component.

pressure drop—loss in pressure, as from one end of a refrigerant line to the other, due to friction, etc.

pressure equalizing—allowing high and low-side pressures to equalize or nearly equalize during idle periods as by use of an unloading valve or a vapor lock liquid control; or nearly equalizing inlet and discharge pressures on the compressor. In either case, to reduce starting torque load.

pressure-imposing element—every device or portion of the equipment used for the purpose of increasing the pressure upon the refrigerant.

pressure-limiting device—a pressure-responsive mechanism designed to stop automatically the operation of the pressure-imposing element at a predetermined pressure.

pressure relief device—a valve or rupture member designed to relieve excessive pressure automatically.

pressure relief valve—a valve held closed by a spring or other means and designed to relieve automatically pressure in excess of its setting.

pressure vessel—any fluid-containing receptacle. In a refrigerating system other than expansion coils, headers and pipe connections.

primary fluid—the refrigerant, to distinguish from secondary fluid, or brine.

psychrometer—instrument for measuring relative humidities by means of wet and dry-bulb temperatures. See hygrometer.

psychrometric chart—a graphical representation of the properties of steam and air mixtures.

psychrometry—use of psychrometer, but broadly speaking the science and practices associated with atmospheric air mixtures and their control.

pump down (refrigeration system)—the operation by which the refrigerant in a charged system is pumped into the liquid receiver.

purging—the act of blowing out gas from a refrigerant-containing vessel, usually for the purpose of removing non-condensibles.

purger—device for removing non-condensible gas from refrigerant condensers or for removing low concentration liquor from absorption system evaporators.

pyrometer—instrument for measuring high temperature.

quality, of wet vapor—fraction by weight of vapor in mixture of liquid and vapor.

quick freezer room—room kept at very low temperature for the purpose of freezing foodstuffs rapidly.

radiation—the transmission of heat through space by wave motion. The passage of heat from one object to another without warming the space between. The heat is passed by wave motion similarly to transmission of light.

ratio of compression—ratio of absolute pressures after and before compression.

raw water—in ice making, any water used for ice making except distilled water.

reach-in refrigerator—commercial refrigerator, service refrigerator.

reciprocating compressor—a positive displacement compressor with a piston or pistons moving in a straight line but alternately in opposite directions.

recirculated air—return air passed through the conditioner before being again supplied to the conditioned space.

rectifier—in refrigeration, externally cooled heat exchanger in high side of absorption system for condensing absorbent and separating it from refrigerant before passing to condenser. In electricity, a device for converting AC to DC current.

reexpansion line—curve on indicator card, representing path of fluid left in cylinder at end of stroke during return of piston prior to inlet.

refrigerant—the medium of heat transfer in a refrigerating system which picks up heat by evaporating at a low temperature and pressure and gives up heat on condensing at a higher temperature and pressure.

refrigerated plate method of freezing—heat transfer through the direct contact of refrigerated plates with the packaged product.

refrigerating engineering—technique of design, manufacture, application and operation of refrigerating machinery and its primary equipment. Refrigeration (except as exact measure in heat units) refers here to a more general science, more concerned with the use of coldness for commercial and other useful purposes.

refrigerating machine—broadly, the refrigerant system proper, as condenser, receiver, compressor, and evaporator of compression system; strictly the high side only. Also used to refer to compressor only.

refrigerating medium—this may mean refrigerating fluid, but usually refers to a cold

medium such as ice, solid CO_2, frozen brine, etc.

refrigerating system—a combination of interconnected refrigerant-containing parts in which a refrigerant is circulated for the purpose of extracting heat.

refrigerator—a container and means of cooling it (domestic) or a large container such as a storage refrigerator, service refrigerator, etc.

regain—of moisture, the amount absorbed by any material in per cent of weight of that material.

regelation—refreezing of water that has resulted from the melting of ice under pressure; does not require refrigeration.

regenerative heating (or cooling)—process of utilizing heat which must be rejected in one part of the cycle, to perform a useful function in another part of the cycle, by heat transfer.

register—grille supplied with control damper.

reheating air—in an air conditioning system the final step in treatment, in the event the temperature is too low.

resistance, thermal—the reciprocal of conductance, or of overall coefficient of heat transfer.

respiration—production of CO_2 and heat by ripening of perishables in storage, also the breathing process of animals.

return air—air returned from conditioned or refrigerated space.

reverse cycle—heating—see heat pump.

reversible cycle—theoretical thermodynamic cycle which can be completely reversed; e.g., Carnot cycle.

room dry bulb (dew point, etc.)—the dry-bulb (dew-point, etc.) temperature of the conditioned room or space.

room air conditioner—air conditioning element for a room, self-contained system for conditioning atmosphere for comfort.

rotary compressor—one in which compression is attained in a cylinder by rotation of a positive displacement member.

run-around system—may be direct air-to-air heat exchangers or conduits in which is circulated brine, water, or expanding and condensing refrigerant. Apparatus consists of a surface or spray dehumidifier with counterflow water (or brine) coils on both entering and leaving sides, connected in series with a circulating pump. Heat from the entering air is absorbed by the precooler and returned to the leaving air by the after-heater coil.

running out—of food juices, see drip.

rupture member—a device which will automatically rupture at a predetermined pressure.

safety head—in a compressor, a cylinder head held in place by a spring of such strength that it will not be compressed during normal operation, but will be moved by solid or liquid matter or abnormal gas pressure between it and the piston, thereby protecting the compressor.

salinometer—hydrometer calibrated in salt concentration.

saponify—turn to soap, as of oil in contact with refrigerant; properly, the decomposition of ester into an alcohol and an acid.

saturated air—air in which the partial pressure of water vapor is equal to the vapor pressure of water at the existing temperature.

saturated vapor—vapor in equilibrium with its liquid, that is, when the number of molecules passing in two directions through the surface dividing the two phases is equal.

saturation—the condition for coexistence in stable equilibrium of two or more distinct phases.

saturation ratio, μ—the ratio of the specific humidity of an air water-vapor mixture to the specific humidity of this mixture saturated with water vapor at the same temperature.

seal, bellows—metal bellows used in a shaft seal or in place of a packing for valves. Also used in long pipe lines instead of gaskets to compensate for expansion of the line with temperature.

sealed unit—a motor compressor assembly having the compressor contained within a gas-tight casing through which no shaft extends. Drive is usually by a motor within the same casing, but may be induced by external means.

secondary condenser—the condenser of a secondary system; also, a condenser and a secondary system where the condenser is cooled by the evaporator of the secondary system.

secondary fluid—fluid cooled by the refrigerant in indirect method of refrigeration.

secondary refrigerant—the volatile refrigerant used in a secondary system or a non-volatile liquid refrigerant such as brine. It might be air. Should be further defined as volatile, non-volatile or gaseous.

self-contained air conditioning or cooling unit—a condensing unit combined in the same cabinet with the other functional elements. Self-contained air conditioning units are classified according to the method of rejecting condenser heat (water-cooled, air-cooled and evaporatively cooled), method

of introducing ventilation air (no ventilation, ventilation by drawing air from outside, ventilation by exhausting room air to the outside, or ventilation by a combination of the last two methods), and method of discharging air to the room (free delivery or pressure type).

self-contained system—a complete factory-made and factory-tested system in a suitable frame or enclosure which is fabricated and shipped in one or more sections and in which no refrigerant-containing parts are connected in the field other than by companion or block valves. See also unit system.

sensible heat—heat which is associated with a change in temperature; specific heat exchange of temperature; in contrast to a heat interchange in which a change of state (latent heat) occurs.

sensible heat coolers—in the form of cooling surface using water, brine or direct expansion refrigerant. It is always located on the leaving side or the dehydrator, but frequently treats in addition a large volume of room air which is not circulated through the dehydrator for moisture reduction.

service refrigerator—commercial refrigerator of reach-in type.

shaft seal—a rubbing seal or stuffing box—used to prevent fluid leakage between the shaft and bearing of a compressor or other fluid moving device.

sharp freezer—cold storage freezer room, generally kept at −30 F to − 10 F, to receive unfrozen goods and freeze them.

shell-and-tube—pertaining to heat exchangers in which a nest of tubes or pipes, or a coil of tube or pipe, is contained in a shell or container. The pipe (or pipes) carries a fluid through it while the shell is also provided with an inlet and outlet for a fluid flow.

shell-type apparatus—a refrigerant-containing pressure vessel having tubes for the passage of a refrigerating fluid or one to be cooled.

sight glass—glass tube used to indicate the liquid level in tanks, bearings and similar equipment.

silica gel—a drier material having the formula, SiO_2.

single can fillers—a device connected by hose with the source of water supply and provided with a starting device and float actuated valve, such that when placed in an ice can and started, it will deliver to the can a predetermined volume of water and then stop the flow.

sludge—a decomposition product formed in a system due to impurities in the oil or to moisture. Sludge may be gummy or hard.

small machine—compressor, high side or complete system having small capacity; commercial machine

solenoid valve—a valve which is closed by gravity, pressure, or spring action and opened by the movement of a plunger due to the magnetic action of an electrically energized coil. A valve which is normally open and becomes closed when energized will be known as a *reverse-acting solenoid valve*.

solubility—of solid, liquid or gas A in solid or liquid B, the mass of A, per unit mass of B, contained in a saturated solution of A in B.

specific gravity—density compared to density of standard material; reference usually to water or to air.

specific heat—energy per unit of mass required to produce one degree rise in temperature, usually Btu per lb deg F numerically equal to cal per gram deg C.

specific volume—volume per unit of mass, usually cu ft per lb.

spray deck—overhead bunker where air is cooled and circulated by brine sprays.

spray method of freezing—refrigerated liquid is sprayed into an insulated enclosure containing the product to be frozen.

spray pond—arrangement for lowering the temperature of water by evaporative cooling, in contact with outside air, in which the water to be cooled is sprayed by nozzles into the space above a body of previously cooled water, and allowed to fall by gravity into it.

standard air—air weighing 0.075 lb per cu ft which is closely air at 68 F dry bulb and 50% relative humidity at a barometric pressure of 29.92 in. of mercury, or approximately dry air at 70 F at the same pressure.

standard conditions—operating conditions specified by codes under which standard ratings are obtained.

standard rating—of a refrigerating machine, is capacity, in tons or Btu per hr, when operating under standard conditions.

steam-jet system—high-pressure steam supplied through a nozzle and acting to eject vapor from the evaporator, thus maintaining the requisite low pressure on one side and producing a higher pressure on the other by virtue of compression in a following diffusion passage.

stop valve—a shut-off for controlling the flow of refrigerant.

storage room—any cold storage space, usually on larger plant scale.

stroke—of piston, length of travel, twice its crank radius.

strong, rich—liquor or solution, water with high concentration of another substance, as ammonia, salt, etc.

stuffing box—a packing gland surrounding a shaft, stem or rod to prevent leakage.

subcooling—process of cooling refrigerant below condensing temperature, for a given pressure. Also cooling a liquid below its freezing point, where it can exist only in a state of unstable equilibrium.

subcooling refrigeration effect—additional refrigeration effect made available by subcooling the refrigerant liquid.

sublimation—change of state directly from solid to gas without appearance of liquid.

submerged condenser—condenser piping submerged in bath of condenser water.

suction line—the tube or pipe which carries the refrigerant vapor from the evaporator to the compressor inlet.

summer air conditioning—cooling, dehumidifying, air distribution and air cleaning, when outside temperatures are above the inside or room temperature.

sun effect—solar energy transmitted into space through windows and building materials.

superheated vapor or gas—vapor at a temperature which is higher than the saturation temperature (i.e., boiling point) at the existing pressure.

surface conductance—the amount of heat transmitted by radiation, conduction, and convection from a surface to the air or liquid surrounding it, or vice versa, in one hour per square foot of surface for a difference in temperature of 1 deg F between the surface and the surrounding air or liquid.

surface cooling—a method of cooling air or other gas by passing over cold surfaces.

surge drum—see accumulator.

sweating—condensation of moisture from air on a surface which is below the dew-point temperature.

system—a refrigerating scheme and/or machine usually confined to those parts in contact with refrigerant; an arbitrarily chosen group of materials and devices set apart for analytical study.

temperature, effective—an arbitrary index of the degree of warmth or cold felt by the human body in response to temperature, humidity, and movement of the air.

temperature, room—temperature of any room, as for example (1) a room in which refrigerator is being operated or tested; (2) a room being conditioned for the comfort of occupants. Colloq.: used to mean ordinary temperature one is accustomed to find in dwellings.

temperature, saturation—of a fluid, boiling point corresponding to a given pressure; evaporation temperature; condensation temperature.

temperature, wet-bulb—equilibrium temperature of water evaporating into air when the latent heat of vaporization is supplied by the sensible heat of the air.

thaw needle—a small tube with valved head, connected by hose to a source of warm water or steam, which may be inserted in the air agitation tube in an ice can for the purpose of thawing the air agitation tube loose from the ice block.

thawing—changing free water, or contained water as in foods, from solid phase to liquid phase by the addition of heat.

thermal conductivity—the ability of a material to transmit heat from one point in the material to another. Indicated in terms of Btu per hr sq ft deg F per in. of thickness.

thermal valve—a valve controlled by a thermally responsive element, for example, a thermostatic expansion valve which is usually responsive to suction or evaporator temperature.

thermocouple—device for measuring temperatures utilizing the effect that an electromotive force is generated whenever two junctions of two dissimilar metals in an electric circuit are at different temperature levels.

thermodynamic properties—basic qualities used in defining the condition of a substance, such as temperature, pressure, volume, enthalpy, entropy.

thermodynamics—the science of heat energy and its transformations to and from other forms of energy.

thermometer—instrument for measuring temperature.

thermostat—automatic control device responsive to temperature.

thermostat, direct-acting—an instrument which permits passage of control air when temperature is higher than the temperature for which instrument is set and cuts off the air when temperature goes below the temperature for which instrument is set.

thermostat, reverse-acting—an instrument which permits passage of control air when temperature is lower than the temperature for which instrument is set and cuts off the air when temperature goes higher than the temperature for which the instrument is set.

throttling—of a fluid, an irreversible adia-

batic process which consists of lowering pressure by an expansion without work.

throw—the distance air will carry measured along the axis of an air stream from the supply opening to the position in the stream at which air motion reduces to 50 fpm.

ton of refrigeration—a rate of heat interchange of 12,000 Btu per hour; 200 Btu per min.

ton-day of refrigeration—the heat removed by a ton of refrigeration operating for a day, 288,000 Btu. Approximately equal to the latent heat of fusion or melting of 1 ton (2,000 lb) of ice.

total cooling effect of a unit—the difference between the total enthalpy of the dry air and water vapor mixture entering the unit per hour and the total enthalpy of the dry air and water vapor (and water) mixture leaving the unit per hour, expressed in Btu per hr.

total heat—obsolescent name for enthalpy.

total tons—the total heat load expressed in tons of refrigeration; the sum of the dry tons and the moisture tons.

transmission—in thermodynamics a general term for heat travel; properly, heat transferred per unit of time.

transmittance—when heat flows from one medium to another, through one or more intermediate resistances, the transmittance of the system is equal to the rate of heat flow per sq ft of a specified side of the boundary, divided by the difference in the temperature of the two surfaces.

tray freezing method—food arranged on shallow trays, placed in portable racks, are placed in insulated chambers and subjected to moving refrigerated air for freezing.

triple point—static point at which three phases of given substance (i.e. solid, liquid and gas) exist in equilibrium.

tubing, soft copper—deoxidized and dehydrated, seamless, soft copper; thoroughly annealed to assure quality for bending, flaring. Commonly used for Freon-12 and methyl chloride.

turbulent flow—fluid flow in which the fluid moves transversely as well as in the direction of the tube or pipe axis, as opposed to streamline or viscous flow.

"*U*" factor—overall coefficient of heat transmission; the amount of heat expressed in Btu transmitted in one hour per square foot of the wall, floor, roof or other surface for a difference in temperature of 1 deg F between the fluid on the inside and that on the outside of the surface.

unit—1. A factory-made, encased assembly of the functional elements indicated by its name, such as air conditioning unit, room cooling unit, humidifying unit, etc. or 2. Unit, hermetically sealed—refrigerating unit having no exposed mechanical driving connection and containing no shaft seal.

unit cooler—adapted from unit heater to cover any cooling element of condensed physical proportions and large surface, generally equipped with a fan.

unit system—a self-contained system which has been assembled and tested prior to its installation and which is installed without connecting any refrigerant-containing parts. See also self-contained system.

unloader—a device in a compressor for equalizing high and low-side pressures when the compressor stops, and for a brief period after it starts, in order to decrease the starting load on the motor.

usage load—sum of the door load and product load factors.

unit system, removable—a refrigerating system which is readily removable as one unit from the cabinet or space which it cools and from the building in which it is used without disconnecting any refrigerant-containing part of the system.

vacuum system—in refrigeration one which employs a vacuum to boil water at the temperature desired; one which employs evaporating water vapor as the refrigerant.

valve, back-pressure—one controlling flow in the evaporator vapor line because of a fluctuating suction pressure, it is necessary to prevent the evaporator temperature from falling below a pre-determined value.

valve, check—a valve allowing (refrigerant) flow in one direction only.

valve, direct-acting diaphragm—one which closes with the admission of fluid pressure to a diaphragm and opens when pressure is released.

valve, discharge—on a compressor, the valve which allows compressed refrigerant to flow from the cylinder to the discharge main.

valve, expansion—1. Valve for controlling the flow of refrigerant to the cooling element, or 2. Expansion valve, automatic—a device which regulates the flow of refrigerant from the liquid line into the evaporator to maintain a constant evaporator pressure, or 3. Thermostatic—a device to regulate the flow of refrigerant into an evaporator in order to maintain an evaporation temperature in a definite relationship to the temperature of a thermostatic bulb.

valve, emergency relief—a manually-operated valve for the discharge of refrigerant in case of fire or other emergency.

valve, float—1. Regulating valve controlled

by liquid level, or 2. Valve actuated by float in a liquid container.

valve, low-side float—a float valve operating by changes in level of low-pressure liquid, opens at low level and closes at high.

valve, king—stop valve between receiver and liquid main.

valve, packless—a valve which does not use packing to prevent leaks around the valve stem. Flexible material usually used to seal against leaks and still permit valve movement.

valve, pressure relief—a valve held closed by a spring or other means, which automatically relieves pressure in excess of its setting.

valve, purge—device to allow fluid to flow out of system, particularly non-condensible gases. Also called a drain valve.

valve, reducing—a valve which maintains a uniform pressure on its outlet side, irrespective of how the pressure on its inlet side may vary above the pressure to be maintained.

valve, reverse-acting diaphragm—opens with the admission of pressure to the diaphragm and closes when pressure is released.

valve, safety—same as pressure relief valve.

valve seat—the stationary portion of the valve which, when in contact with the movable portion, stops flow completely.

valve, stop—a shut-off valve other than a service valve for controlling the flow of the refrigerant.

valve, suction—in a compressor, the valve which allows refrigerant to enter the cylinder from the suction line, and prevents return flow.

vapor—a gas, particularly one near to equilibrium with the liquid phase of the substance. Usually used instead of gas for a refrigerant, and in general for any gases below the critical temperature.

vapor lock—formation of some vapor or all vapor in a liquid line reducing weight flow as compared to weight flow in liquid phase with the same pressure differential.

vapor lock device—an orifice; capillary tube or other device having a restricted passage of fixed size for liquid refrigerant. It restricts flow of vapor of that same liquid to a lower rate of flow (in lb per min) with the same pressure difference.

velocity—the time rate of motion of a body in a fixed direction. In the fps system it is expressed in units of feet per second.

velocity, critical—the velocity above which fluid flow is turbulent.

ventilation—the process of supplying or removing air by natural or mechanical means, to or from any space. Such air may or may not have been conditioned.

venturi meter—measuring device for flow of fluid.

vertical bunkers (refrigerated cars and trucks)—filled with ice, the air in the bunker is chilled and flows out of the bottom opening in the bunker into the space beneath the floor rack.

vital heat—heat generated by fruits and vegetables in storage, due to ripening.

volatile liquid—one which evaporates readily at atmospheric pressure and room temperatures.

walk-in box—storage refrigerator.

wall-section—cross section of wall arranged chiefly to reveal conductivity characteristics.

welded system—one in which all refrigerant joints are brazed or welded.

warm-air duct—in commercial refrigerators the passage through which warm air ascends from storage space to the bunker. In heating, ventilating and air conditioning, pipes or ducts for conveying warm air into or out of a space or room.

water forecooling tank—tank where inlet water is cooled prior to freezing.

water or brine cooler—a factory-made assembly of elements in which the water or brine and the refrigerant are in heat transfer relationship, causing the refrigerant liquid to evaporate and the water or brine to be cooled.

water vapor—used commonly in air conditioning parlance to refer to steam in the atmosphere.

wax—a material which may separate on cooling of oil-refrigerant mixtures. Wax may plug the expansion valve and reduce heat transfer of the coil.

weep—drip from frozen foods.

wet-bulb depression—difference between dry-bulb and wet-bulb temperature.

wet compression—a system of refrigeration in which some liquid refrigerant is mixed with vapor entering the compressor so as to cause the discharge vapors from the compressor to be saturated rather than superheated.

window unit—self-contained room cooler arranged to be supported in or connected with a window opening, circulating outside air over the high side and room air over the low side.

winter air conditioning—heating, humidification, air distribution and air cleaning, where outside temperatures are below the inside or room temperature.

wire drawing—restriction of area for a flowing fluid, causing a loss in pressure by

(internal and external) friction without loss of heat or performance of work; throttling.

wort—the unfermented infusion of malt that when fermented produces beer.

year 'round air conditioning system—one which ventilates, heats and humidifies in winter and cools and dehumidifies in summer the spaces under consideration, and provides the desired degree of air motion and cleanliness. The equipment usually comprises preheater, filters, spray or surface dehumidifier, reheaters and bypass if required, fan, system of distributing ducts, and necessary means of manual or automatic control.

42. CODES AND STANDARDS

The Codes and Standards listed below represent accepted practice, methods, or standards prepared and accepted by the organizations indicated. They are valuable guides for the practicing engineering determining test methods, ratings, performance requirements, and limits applying to equipment used in refrigeration and air conditioning. Copies can usually be obtained from the organizations listed in the reference column.

SUBJECT	TITLE	SPONSOR	REFERENCE
Air Conditioning	Code of Minimum Requirements for Comfort Air Conditioning (1938)	A.S.H.V.E. ASRE	A.S.H.V.E.
Air Conditioning (150,000 Btu/Hr or less)	Code and Manual for the Design and Installation of Warm Air Winter Air Conditioning Systems (1945)	N.W.A.H. & A.C.A.	N.W.A.H. & A.C.A. No. 7
Air Conditioning (Above 150,000 Btu/Hr)	The Technical Code for the Design and Installation of Mechanical Warm Air Heating Systems (1942)	N.W.A.H. & A.C.A.	N.W.A.H. & A.C.A.
Building Requirements	American Standard Building Requirements (1946)	N.H.A. U.S.P.H.S.	A.S.A. A53.1—1946
Cleaners (Air)	A.S.H.V.E. Standard Code for Testing and Rating Air Cleaning Devices Used in General Ventilation Work (1934)	A.S.H.V.E.	See A.S.H.V.E. Transactions Vol. 39, 1933, p. 225
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1945)	H.P. & A.C.N.A.	H.P. & A.C.N.A. Engrg. Stds. Sec. 2 Part V
Color Scheme (Piping)	Scheme for Identification of Piping Systems (1928)	A.S.M.E.	A.S.A. A13—1928
Coils	Proposed Commercial Standard for Rating and Testing Air Cooling Coils Using Non-Volatile Refrigerants (1945)	B.C.M.I. B.S.	B.S T.S. 4044
Compressors	Tentative ASRE Standard Methods of Rating and Testing Refrigerant Compressors	ASRE A.S.H.V.E. A.C.R.M.A.	ASRE Standard 23
Condensers	ASRE Standard Methods of Rating and Testing Evaporative Condensers	ASRE A.S.H.V.E. A.C. & R.M.A.	ASRE Standard 20
Condensers	ASRE Standard Methods of Rating and Testing Water-Cooled Refrigerant Condensers	ASRE A.S.H.V.E. A.C. & R.M.A.	ASRE Standard 22
Condensing Units	ASRE Standard Methods of Rating and Testing Mechanical Condensing Units (1940)	ASRE A.S.H.V.E. A.C. & R.M.A.	ASRE Standard 14
Condensing Units	Commercial Standard for Commercial Electric Refrigeration Condensing Units (1945)	B.S. S.R.C.A.	B.S. CS107-45
Conductivity	Standard Method of Test for Thermal Conductivity of Materials by Means of the Guarded Hot Plate (Tentative) (1942)	A.S.H.V.E. ASRE A.S.T.M. N.R.C.	A.S.H.V.E.

SUBJECT	TITLE	SPONSOR	REFERENCE
Control Equipment (Industrial)	Underwriter's Laboratories, Inc., Standard for Industrial Control Equipment (July 1938, reprinted Sept. 1945)	U.L.	U.L. Subject 508
Controls	Underwriter's Laboratories, Inc., Standard for Temperature Indicating and Regulating Equipment (Jan 1947)	U.L.	U.L. Subject 873
Convector	Recommended Commercial Standard for Testing and Rating Convectors (1946)	B.S. C.M.A. I.B.R.	B.S. TS-4169
Coolers (Air)	ASRE Standard Methods of Rating and Testing Forced Circulation and Natural Convection Air Coolers for Refrigeration (1945)	ASRE A.S.H.V.E. A.C.R.M.A. R.E.M.A.	ASRE Standard 25
Coolers	ASRE Standard Methods of Rating and Testing Water and Brine Coolers	ASRE A.S.H.V.E. A.C. & R.M.A.	ASRE Standard 24
Cooling Units	ASRE Standard Methods of Rating and Testing Self-Contained Air Conditioning Units for Comfort Cooling (1940)	ASRE A.S.H.V.E. R.M.A. R.E.M.A. A.C.M.A.	ASRE Standard 16
Exchangers (Heat)	Standards of Tubular Manufacturers Association (1941)	T.E.M.A.	T.E.M.A.
Fans	Definitions and Terms in Use by the Blower Industry (1946)	N.A.F.M.	N.A.F.M. Bulletin No. 105
Fans	Standard Test Code for Centrifugal and Axial Fans (1938)	N.A.F.M.[a] A.S.H.V.E.	N.A.F.M. Bulletin No. 103
Fans	N.E.M.A. Fan Standards (1944)	N.E.M.A.	N.A.F.M. Publ. 44–95
Heat Transfer (Walls)	A.S.H.V.E. Standard Test Code for Heat Transmission through Walls (1927)	A.S.H.V.E.	A.S.H.V.E.
Piping	American Standard Code for Pressure Piping (1942)	A.S.M.E.	A.S.A. B31.1—1942
Pumps	Hydraulic Institute Test Code for Centrifugal Pumps. Hydraulic Institute Test Code for Rotary Pumps (1943)	H.I.	H.I. Section F
Refrigeration (Equipment)	Underwriter's Laboratories, Inc., Standard for Air Conditioning and Commercial Refrigerating Equipment (Feb. 1946)	U.L.	U.L. Subject 207A
Refrigeration (Mechanical)	American Standard Safety Code for Mechanical Refrigeration (1939)	ASRE	ASRE Standard 15
Refrigeration (Unit Systems)	Underwriter's Laboratories, Inc., Standard for Unit Refrigerating Systems (Feb. 1946)	U.L.	U.L. Subject 207C

SUBJECT	TITLE	SPONSOR	REFERENCE
Refrigerators (Gas-Fired)	American Standard Approval Requirements for Refrigerators Using Gas Fuel (1941)	A.G.A.	A.S.A. Z21.19—1941
Refrigerators (Household)	American Standard Test Procedure for Household Electric Refrigerators (Mechanically Operated) (1944)	ASRE U.S.D.A.	A.S.A. B38.2—1944
Unit Heaters	American Standard Approval Requirements for Gas Unit Heaters (1940)	A.G.A.	A.S.A. Z21.16—1940
Unfired Pressure Vessels	Unfired Pressure Vessel Code (1946 with 1947 Addenda)	A.S.M.E.	A.S.M.E.
Vacuum Pumps	A.S.H.V.E. Standard Code for Testing and Rating Return Line Low Vacuum Heating Pumps (1934)	A.S.H.V.E.	A.S.H.V.E.

ABBREVIATIONS

A.C.M.A.[1]	Air Conditioning Manufacturers Association.
A.C.R.M.A.	Air Conditioning and Refrigerating Machinery Association.
A.F.A.	American Foundrymen's Association.
A.G.A.	American Gas Association.
A.G.A.E.M.[2]	Association of Gas Appliance and Equipment Manufacturers.
A.I.L.	Anthracite Industries Laboratory.
A.S.A.	American Standards Association.
A.S. of A.	Acoustical Society of America.
A.S.H.V.E.	American Society of Heating and Ventilating Engineers.
A.S.M.E.	American Society of Mechanical Engineers.
A.S.R.E.	American Society of Refrigerating Engineers.
A.S.T.M.	American Society for Testing Materials.
B.S.	National Bureau of Standards.
C.M.A.	Convector Manufacturers Association.
F.H.A.	Federal Housing Administration.
G.A.M.A.	Gas Appliance Manufacturers' Association.
H.I.	Hydraulic Institute.
H.P. & A.C.N.A.	Heating, Piping and Air Conditioning Contractors National Association.
I.B.R.	Institute of Boiler and Radiator Manufacturers Association.
I.C.H.A.M.	Institute of Cooking and Heating Appliance Manufacturers.
I.M.W.I.	Industrial Mineral Wool Institute.
I.U.H.A.	Industrial Unit Heater Association.
N.A.F.M.	National Association of Fan Manufacturers.
N.B.F.U.	National Board of Fire Underwriters.
N.E.M.A.	National Electrical Manufacturers Association.
N.F.P.A.	National Fire Prevention Association.
N.H.A.	National Housing Agency.
N.R.C.	National Research Council.
N.W.A.H. & C.A.	National Warm Air Heating and Air Conditioning Association.
O.B.I.[3]	Oil Burner Institute.
O.H.I.A.	Oil Heat Institute of America.
O.P.A.	Office of Price Administration.
P.F.M.A.	Propeller Fan Manufacturers Association.
R.E.M.A.	Refrigeration Equipment Manufacturers Association.
R.M.A.[1]	Refrigerating Machinery Association.
S.A.E.	Society of Automotive Engineers.
S.B.I.	Steel Boiler Institute.
S.R.C.A.	Standard Refrigeration Compressor Association.
T.E.M.A.	Tubular Exchanger Manufacturers Association.
U.L.	Underwriter's Laboratories.
U.S.D.A.	United States Department of Agriculture.
U.S.P.H.S.	United States Public Health Service.

[1] Superseded 1940 by A.C.R.M.A.
[2] Superseded 1945 by G.A.M.A.
[3] Superseded 1942 by O.H.I.A.

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REFRIGERATING DATA BOOK

Sixth Edition

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REFRIGERATION CLASSIFIED

REFRIGERATING DATA BOOK

Basic Volume

Sixth Edition, 1949

THE "REFRIGERATION CLASSIFIED" Section of the Data Book has been completely revised for this edition.

The list of headings is intended to include all items of materials, components and accessories, which may ultimately become, through manufacturing, assembling or installation on the job, integral parts of any type of refrigerator, air conditioner, or refrigerating or air conditioning system, as well as complete systems or package units.

For the convenience of the user, headings are arranged in alphabetical order. The main listings for each item appear under the most generally accepted name for that item. Other names for the same item are found in proper alphabetical order, with cross-references to the main listing. In some instances cross-references are found to other items which may not be the same but which may perform similar functions. In all instances the address of a source of supply is given together with its name.

Advertisers are indicated by bold-face type, followed by a reference to the page number on which their advertising appears.

Although the publishers assume no responsibility for accuracy or completeness of the listings, every effort has been expended to make the listings authentic and comprehensive.

Suggested additions, or advice regarding inaccuracies or omissions will be warmly welcomed.

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Whatever your refrigeration needs— Call in the expert—Call in Frigidaire!

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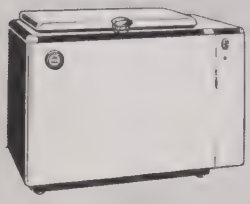

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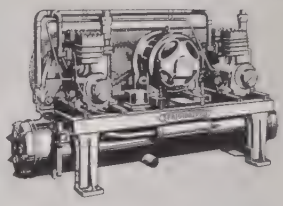

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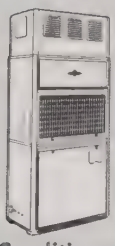

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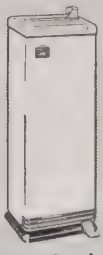

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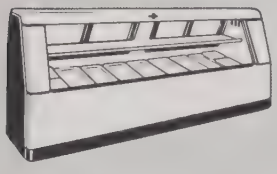

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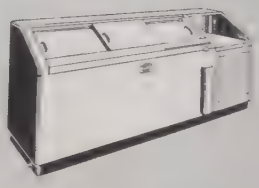

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These examples are typical of Frigidaire's wide variety of refrigeration and air conditioning products. All are engineered by Frigidaire, backed by over a quarter-century of experience in building more than 8 million refrigerating units – your assurance of *Refrigeration Security.*

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Haveg Corp., Marshallton, Del.
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
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ACCESS DOORS

Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
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E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.

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Mueller Brass Co., Port Huron, Mich.
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Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Hetzel Roofing Products Co., 27 Main St., Newark, N.J.
Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
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Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
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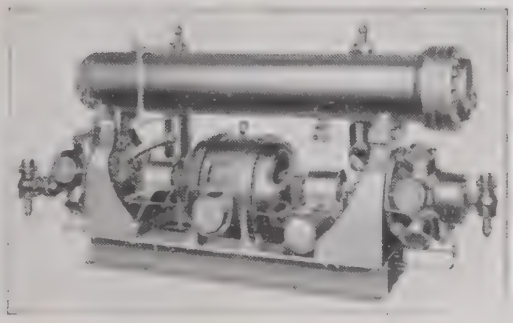
(C) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
Modine Mfg. Co., Racine, Wis.
(C) John J. Nesbitt, Inc., Holmesburg, Phila. 36, Pa. *(p. 13)*
(C) Niagara Blower Co., 6 E. 45th St., N.Y.C., 17 *(p. 96)*
(C) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
Reese & Long Refrigeration Products, Inc., 408 E. 25th St., N.Y.C. 10
Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Rigidbilt, Inc., 2505 S. Pulaski Rd., Chicago 23, Ill. *(p. 42)*
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59

(Continued)

CHRYSLER AIRTEMP
DIVISION OF CHRYSLER CORPORATION

- CENTRAL SYSTEM COOLING EQUIPMENT
- "PACKAGED" AIR CONDITIONERS
- MARINE COMPRESSOR UNITS

CENTRAL SYSTEM COOLING EQUIPMENT



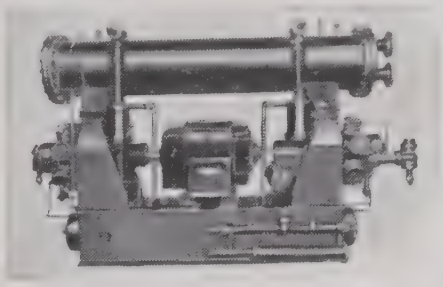
VARIABLE CAPACITY—RADIAL (DIRECT-DRIVE) COMPRESSOR—10 TO 75 H. P.

Chrysler Airtemp central system units offer variable capacities ranging from 10 to 75 H.P. The balanced Radial Compressor reduces weight and the direct drive from motor results in higher speeds . . . Available in three, five and seven cylinder combinations on single compressors and ten to fourteen on dual units. Automatic starting unloader permits "no-load" starting. Oil separator minimizes oil presence in condenser and evaporator. Compressors function efficiently with either evaporative or water cooled system. Shell and finned U-tube condenser is easily cleaned and uses city or cooling tower water.

"PACKAGED" AIR CONDITIONERS

FACTORY — OFFICE — STORE

Neutral color, two-tone crinkle finish 16-gauge bonderized cabinet with fully adjustable supply air grilles which are mountable front or back. Removable plenum chamber provides for ducts. Replaceable filters in large return air grille. Self aligning cushioned bearings suspend dynamically balanced fans driven by heavy duty resilient mounted motor. Continuous fins are securely bonded to all-copper cooling coil. Combination condenser and receiver is equipped with coil type spiral funnel water pass. Automatic water regulating valve. Condenser accommodates city or water tower cooling water. Direct connected Chrysler Airtemp Radial Compressor—both motor and compressor are sealed in oil. Unit has built-in thermostat and high-pressure cut-out. Fan operation can be separate. Heating coils can be added for year round service.



MARINE COMPRESSOR UNITS

VARIABLE CAPACITY — 10 TO 75 H. P.

Compressor, motor, high and low pressure controls, dehydrator and safety devices all mounted on a steel base—a single complete compressor unit ready to install and operate. The simple cylinder unloader mechanism which provides automatic variable capacity is exclusive with Chrysler Airtemp.

CHRYSLER AIRTEMP

AIRTEMP DIVISION OF CHRYSLER CORPORATION, DAYTON 1, OHIO

IN CANADA: THERM-O-RITE PRODUCTS, LTD., TORONTO

AIR CONDITIONING COILS (Continued)

Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
(C) B. F. Sturtevant Div., Westinghouse Elec. Corp. Hyde Park, Boston 36, Mass. *(p. 148)*
(C) Super Radiator Corp., 652 Stinson Blvd., Minneapolis 13, Minn.
Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(C) Trane Co., La Crosse, Wis. *(p. 14)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
(C) U.S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(C) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*
(C) Young Radiator Co., Racine, Wis. *(p. 16)*

AIR CONDITIONING COILS, DIRECT EXPANSION

Aerofin Corp., 410 Geddes St., Syracues, N.Y. *(p. 11)*
American Coils Co., 25 Lexington St., Newark 5, N.J.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Bush Mfg., 179 South St., W. Hartford 10, Ct. *(p. 221)*
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Hastings Air Conditioning Co., Inc., Hastings, Neb.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago, 18, Ill. *(p. 224)*
Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
Reese & Long Refrigeration Products, Inc., 408 E. 25th St., N.Y.C. 10
Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Rigidbilt, Inc., 2505 S. Pulaski Rd., Chicago 23, Ill. *(p. 42)*
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59
Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4 O,
B. F. Sturtevant Div., Westinghouse Elec. Corp.) Hyde Park, Boston 36, Mass. *(p. 148)*
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Trane Co., La Crosse, Wis. *(p. 14)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*
Young Radiator Co., Racine, Wis. *(p. 16)*

AIR CONDITIONING COILS, HEATING, HOT WATER

Aerofin Corp., 410 Geddes St., Syracuse, N.Y. *(p. 11)*
American Coils Co., 25 Lexington St., Newark 5, N.Y.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
E. K. Campbell Heating Co., 1809 Manchester St., Kansas City 3, Mo.
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
G & O Mfg. Co., 138 Winchester Ave., New Haven 8, Ct.
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
Modine Mfg. Co., Racine, Wis.
C. F. Moores Co., Inc., 1123 Ivy Hill Rd., Wyndmoor, Phila. 18, Pa.
John J. Nesbitt, Inc., Holmesburg, Phila. 36, Pa. *(p. 13)*
Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
Peerles of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
Reese & Long Refrigeration Products, Inc., 408 E. 25th St., N.Y.C. 10
Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Rigidbilt, Inc., 2505 S. Pulaski Rd., Chicago 23, Ill. *(p. 42)*
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59
Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4, O.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Super Radiator Corp., 652 Stinson Blvd., Minneapolis 13, Minn.
Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Trane Co., La Cross, Wis. *(p. 14)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*
Young Radiator Co., Racine, Wis. *(p. 16)*

AIR CONDITIONING COILS, HEATING, STEAM

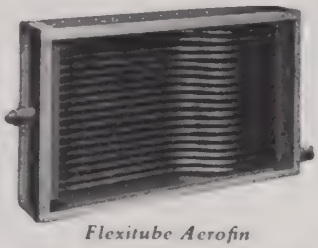
(F—Freeze Proof Construction available)

(F) Aerofin Corp., 410 Geddes St., Syracuse, N.Y.
(F) Airtherm Mfg. Co., 708 S. Spring Ave., St. Louis 10, Mo.
American Coils Co., 25 Lexington St., Newark 5, N.J.
(F) Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
(F) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
E. K. Campbell Heating Co., 1809 Manchester St., Kansas City 3, Mo.
(F) Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
(F) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(F) G & O Mfg. Co., 138 Winchester Ave., New Haven 8, Ct.
(F) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
Hastings Air Conditioning Co., Inc., Hastings, Neb.
(F) Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.

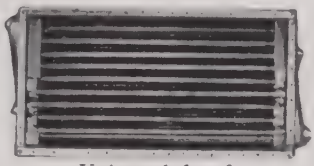
(Continued)

AEROFIN
Heat-Exchange Surface

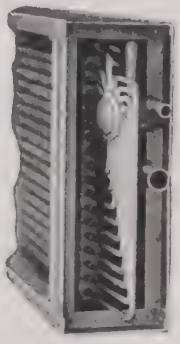
Cleanable Tube Unit with Removable Header



Flexitube Aerofin

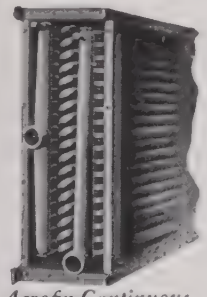


Universal Aerofin



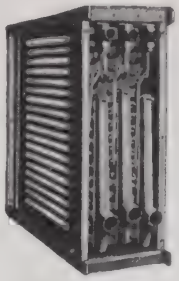
Direct Expansion Unit with Centrifugal Header

PROVEN PERFORMANCE



Aerofin Continuous Tube Water Coil

- **Aerofin Fan System Heat-Exchange Surface is adaptable to meet all practical requirements of refrigeration.**
- **Seventeen years of development, research and manufacturing have made Aerofin one of the foremost in the field.**
- **Aerofin has a complete line for direct expansion refrigerants and water.**
- **Aerofin Fan System Heat-Exchange Surface is an outstanding engineering achievement.**



Aerofin Direct Expansion Unit

- **Send for technical literature or consult any of our district offices for the solution of your particular problem.**

Aerofin Corporation

410 South Geddes Street, Syracuse 1, N. Y.

Chicago · Detroit · New York · Philadelphia · Dallas · Cleveland · Montreal

AIR CONDITIONING COILS, HEATING, STEAM (Continued)

(F) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
King Co., 902 N. Cedar St., Owatonna, Minn.
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p.227)*
Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
(F) McQuay, Inc., 1600 Broadway, N.E., Minneaplis 13, Minn. *(p. 225)*
(F) Marlo Coil Co., 6135 Manchester Ave., St. Louis, 10, Mo. *(p. 229)*
Modine Mfg. Co., Racine, Wis.
D. J. Murray Mfg. Co., 1002-3rd St., Wausau, Wis.
(F) John J. Nesbitt, Inc., Holmesburg, Phila. 36, Pa. *(p. 13)*
(F) Niagara Blower Co., 6 E 45th St., N.Y.C. 17 *(p. 96)*
(F) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
Reese & Long Refrigeration Products, Inc., 408 E. 25th St., N.Y.C. 10
Refrigeration Appliances, Inc., 918 W Lake St., Chicago 1, Ill.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Rigidbilt, Inc., 2505 S. Pulaski Rd., Chicago 23, Ill. *(p. 42)*
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59
Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4, O.
(F) B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Super Radiator Corp., 652 Stinson Blvd., Minneapolis 13, Minn.
Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(F)Trane Co., La Crosse, Wis. *(p. 14)*
(F) Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
(F) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(F) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*
(F) Young Radiator Co., Racine, Wis. *(p. 16)*

(C) Jaden Mfg. Co., Hastings, Neb.
(C,F) Kauffman Air Conditioning Corp., 4336 W. Pine Blvd., St. Louis, Mo.
(C,F) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
(C) King Co., 902 N. Cedar St., Owatonna, Minn.
(C) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
(C,F) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
(C,F) Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
(C,F) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(C,F) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
(C,F) Merchant & Evans Co., 2035 Washington Ave., Phila. 46, Pa.
(C,F) Modine Mfg. Co., Racine, Wis.
(C) D. J. Murray Mfg. Co., 1002-3rd St., Wausau, Wis.
(C,F) Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
(C,F) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(C,F) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
(C,F) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
(C,F) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
(C) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(C,F) Rigidbilt, Inc., 2505 S. Pulaski Rd., Chicago 23, Ill. *(p. 42)*
(C,F) Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59
(C,F) B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
(C) Tenney Engrg., Inc., 26 Ave B, Newark 5, N.J.
(C,F) Trane Co., La Crosse, Wis. *(p. 14)*
(C,F) Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
(C,F) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
(C,F) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(C,F) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(C,F) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(C,F) York Corp., York, Pa. *(p. 163)*
(C,F) Young Radiator Co., Racine, Wis. *(p. 16)*

AIR CONDITIONING FAN COIL UNITS
(C—Ceiling mounted; F—Floor mounted)

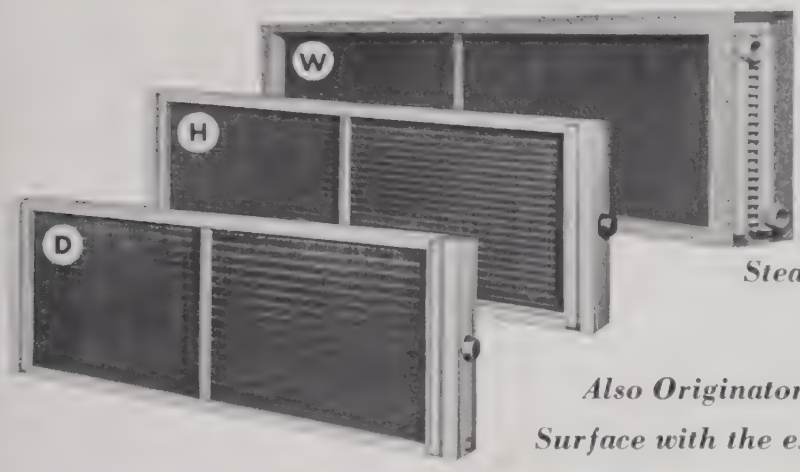
(F) American Coils Co., 25 Lexington, St., Newark 5, N.J.
(C,F) American Thermal Industries, Inc., 440 Illinois, Detroit 1, Mich.
(C,F) Baker Ice Machine Co., Inc., S. Windham, Me *(p. 191)*
(C)Betz Corp., 445 State St., Hammond, Ind.
(C,F) Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
(C,F) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
(C,F) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(C,F) Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
(C,F) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(F) Frick Co., Waynesboro, Pa. *(p. 47)*
(C) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(C,F) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
(C) General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
(C,F) Governaire Corp., P.O. Box 1654, Oklahoma City, Okla.
(C,F) Hastings Air Conditioning Co., Inc., Hastings, Neb.
(C,F) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 224)*

AIR DIFFUSERS (See also GRILLES also REGISTERS)

A-J Mfg. Co., 2119 Washington, Kansas City 8, Mo.
Air Devices, Inc., 17 E. 42nd St., N.Y.C. 17
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
Alton Mfg. Co., 1112 Ross Ave., Dallas 2, Tex.
American Coils Co., 25 Lexington St., Newark 5, N.J
Anemostat Corp. of America, 10 E. 39th St., N.Y.C
Barber-Colman Co., Rockford, Ill.
W. B. Connor Engrg. Corp., 114 E. 32nd St., N.Y.C. 16 *(p. 20)*
Hastings Air Conditioning Co., Inc., Hastings, Neb.
Knowles Mushroom Ventilator Co., 583 Upper Mountain Ave., Upper Montclair, N.J.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
C. F. Moores Co., Inc., 1123 Ivy Hill Rd., Wyndmoor. Phila. 18, Pa.
Pyle-Nat'l. Co. 1371 W. 37th St., Chicago 9, Ill. *(p. 17)*
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Tuttle & Baily, Inc., New Britain, Ct.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Waterloo Register Co., 113 E. 8th St., Waterloo, Ia.

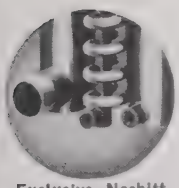
NESBITT SURFACE

FOR HEATING, DRYING, COOLING, DEHUMIDIFYING



Originators of FREEZEPROOF Heating Surface with Dual Steam-Distributing Tubes

Also Originators of Cooling (Water) Surface with the exclusive drain feature —and with the Surface pitched in the casing!



Exclusive Nesbitt Drainability Feature

Cleanable Water Surface with Removable Headers

SERIES W Continuous tube or cleanable tube water surface for air-cooling and de humidifying with cold water, or air heating with hot water. Constructed of copper tubes and plate-type aluminum fins. Single, double and half-serpentine arrangements. Available in wide range of sizes in three types: TYPE WD sections incorporate the exclusive Nesbitt drainability feature (illustrated at left) and have the surface pitched in the casing. Positive drainage of all tubes and return bends is insured, protecting the surface against winter freeze-ups. Publication 246. TYPE WB sections are for booster-heating or air-cooling applications where air volumes are relatively small and drainability unnecessary. Constructed same as Type WD sections except that the surface is set level in the casing and the drainability feature is omitted. Publication 246. TYPE WC sections employ standard Nesbitt Series W Surface cores, pitched in the casing, and are designed for applications where the tubes require periodic cleaning. Headers are constructed of close-grained gray cast iron with removable cover plates (illustrated at left) permitting inspection of the interior without disturbing connection piping. Drainability is obtained by removing drain tappings at both ends of the section. Available in single and double serpentine circuits in a range of sizes. Publication 25[illegible].

SERIES H Highly efficient blast-coil heating surface designed for general heating, ventilating, air-conditioning and drying in both high- and low-pressure steam systems. Copper tubes and headers, aluminum fins, lightweight like Series D Surface, but without steam-distributing tubes. Available in seven surface types, full range of sizes. Publication 248.

SERIES D Heating surface with Dual Steam Distributing Tubes. **Freezeproof plus.** Even distribution of the smallest amount of steam over the full length of the surface; therefore: UNIFORM discharge temperatures; perfect controllability with modulating valves; inherent protection against freezing. Ideally suited for preheating outdoor air. Copper tubes and headers, aluminum fins. Available in Type DS, with single steam headers, in various lengths; and Type DD, with double steam headers for greater lengths. Publication 247.

DIRECT EXPANSION COILS A complete line of direct expansion coils for air-cooling and dehumidifying will soon be available.

Nesbitt Surface is manufactured by

JOHN J. NESBITT, INC.

Sold by Leading Manufacturers of Fan System Apparatus

WRITE TO JOHN J. NESBITT, INC., PHILADELPHIA 36, PA. FOR CATALOGS, DATA AND DISTRIBUTORS

The TRANE Company

2030 Cameron Avenue, La Crosse, Wisconsin

A COMPLETE LINE OF HEATING, COOLING, AIR CONDITIONING AND AIR HANDLING EQUIPMENT

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Appleton, Wis.	Denver, Colo.	Memphis, Tenn.	Salt Lake City, Utah
Atlanta, Ga.	Des Moines, Iowa	Milwaukee, Wis.	San Antonio, Texas
Aurora, Ill.	Detroit, Mich.	Missoula, Mont.	San Francisco, Calif.
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Billings, Mont.	Flint, Mich.	New Orleans, La.	Sioux City, Iowa
Birmingham, Ala.	Gainesville, Fla.	New York, N.Y.	South Bend, Ind.
Boston, Mass.	Grand Rapids, Mich.	Oklahoma City, Okla.	Spokane, Wash.
Buffalo, N.Y.	Greensboro, N.C.	Omaha, Neb.	Toledo, Ohio
Canton, Ohio	Greenville, S.C.	Philadelphia, Pa.	Trumbull, Conn.
Chattanooga, Tenn.	Harrisburg, Pa.	Phoenix, Ariz.	Washington, D.C.
Chicago, Ill.	Houston, Tex.	Pittsburgh, Pa.	West Hartford, Conn.
Cincinnati, Ohio	Indianapolis, Ind.	Portland, Maine	Wichita, Kan.
Clarksburg, W.Va.	Jackson, Miss.	Portland, Ore.	Wilkes-Barre, Pa.
Clarksville, Tenn.	Kansas City, Kan.	Providence, R.I.	Wilmington, Del.
Cleveland, Ohio	Knoxville, Tenn.	Richmond, Va.	Worcester, Mass.

Sales Connections All Over The World

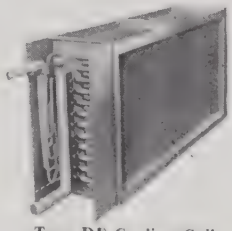
In Canada: Trane Company of Canada, Ltd., Mowat & King Sts., W., Toronto, Ont.

A COMPLETE LINE

The Trane Company builds a complete line of heating, cooling and air handling equipment. So comprehensive is this line that all major items for any application can be supplied by Trane. This undivided responsibility assures architect, engineer and contractor that each piece of equipment will work with the others to give peak efficiency for the over-all system. From basement equipment room to roof ventilator, its Trane equipment.

COOLING COILS

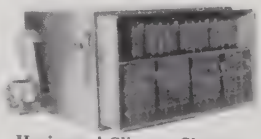
There is a Trane Extended Surface Coil for every comfort cooling or processing application. Trane Cooling Coils come in a wide variety of sizes and arrangements for use with either cold water or direct expansion refrigerants as the cooling medium.



Type DE Cooling Coil

CLIMATE CHANGERS

As its name implies, the Climate Changer is a unit type air conditioner for comfort or industrial processing applications. When the complete unit is used, the Climate Changer performs all six steps of the air conditioning process. Parts may be excluded, however, making it possible to use this unit for any desired phase of air conditioning such as heating, cooling, etc. Also available in multiple zone units.



Horizontal Climate Changer

SELF-CONTAINED AIR CONDITIONERS

For the smaller installation such as home, office, store or shop, these compact units discharge fresh, clean air into the zone of occupancy. Heating coils can be added for year 'round use. These handsomely rugged units are available in sizes from 3 through 10 hp.

Self-Contained Air Conditioner

EVAPORATIVE CONDENSERS

For installations in localities where the use of water for condensing the refrigerant in the air conditioning system is restricted, or where disposal of large quantities of water is a problem. These units condense refrigerants by the evaporative process using a constantly recirculated supply of water. The only water lost is that which is evaporated to cool the water being circulated. Sizes from 3 to 100 tons.

RECIPROCATING COMPRESSORS AND CONDENSER UNITS

Sound engineering, painstaking design, exhaustive testing and precision manufacture carried out with the finest materials available insure the quality and performance of the Trane Compressor and Condenser Units. Compact and perfectly balanced, they give maximum output using a minimum of space. Available in 26 sizes from 3 to 100 tons.

TURBO-VACUUM COMPRESSORS

The Trane Turbo-Vacuum Compressor is a centrifugal water chiller for the larger air conditioning jobs. This vibrationless water chiller is virtually a complete air conditioning equipment room placed in an easily installed single unit.

TRANE DRY EXPANSION CHILLERS

The Trane Dry Expansion Chiller is designed to provide chilled water for various comfort and process cooling jobs. In this unit the refrigerant is circulated in the tubes of a coil contained within the outer shell. Water is passed over the tubes and chilled. Trane Dry Expansion Chillers have over 3,000 variations in a wide range of capacities.

TRANE AIR WASHERS

Trane Air Washers are of strong waterproof construction, designed to give lasting trouble-free service. These units come in 5 foot, 7 foot and 9 foot lengths in one stage units. Two stage units are also available. Capacities of Trane Air Washers range from 3,000 to 330,000 cfm.

TRANE CENTRIFUGAL FANS

Recommended for all types of heating, cooling, ventilating and air handling applications. Available in both forward curved and backwardly inclined blade designs, belt or direct drive, single or double widths. Sizes 4½" to 89" wheel diameters.

TRANE EVAPORATIVE COOLERS

Designed for cooling fluids in a closed system, such as quenching oil, engine jacket water, engine lubricating oil, etc. Using this type of unit eliminates the possibility of contaminating the liquid being cooled and affords cooling without direct contact of fluids with the air.

TRANE COOLING TOWERS

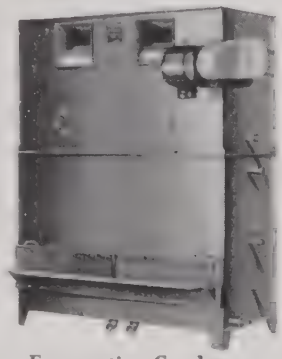
Trane Cooling Towers are available in a variety of sizes from 40 gpm to 360 gpm.

TRANE PRODUCT COOLERS

Trane Product Coolers for forced circulation of refrigerated air in walk-in boxes come in 10 sizes with a wide range of capacities.

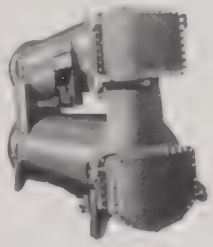
TRANE CIRCULATING PUMPS

The Trane Type SSU Pump is precision built for mounting directly on the driving motor. Available in five sizes from 1" through 2½" with maximum capacity of 250 gallons per minute against 80 feet, or 125 gallons per minute against a maximum head of 120 feet.



Evaporative Condenser KN Series Model

Model 25 CW Condenser



100-ton Turbo-Vacuum Unit

Trane Dry Expansion Chiller

Large FC Centrifugal Fan

UNITRANE

Incorporating modern beauty with the latest in air conditioning developments, the revolutionary new UniTrane represents the finest in multi-room air conditioning. UniTrane is ductless, employing a patented moisture controller that is excusively Trane to give completely independent moisture and temperature control. This feature makes UniTrane the only system that completely processes ventilation air within the unit, entirely eliminating expensive, space consuming ductwork. Each unit is independent from the other, giving full tenant control and increased flexibility.

TRANE AIR CONDITIONING MANUAL

Trane offers the engineering profession an unbiased textbook covering the fundamentals of air conditioning. The Manual not only shows how to design every type of system, but clarifies the underlying principles as well, enabling both student and engineer to reason out their own problems. Price $5.00.

TRANE REFRIGERATION MANUAL

Published primarily as an aid in understanding and correcting installation, maintenance and repair problems. Price $1.50.

OTHER TRANE EQUIPMENT

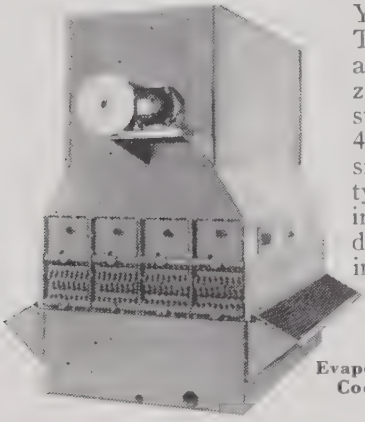
The complete Trane Line also includes —1. Trane Unit Ventilators for schoolroom air conditioning; 2. Trane Condensation and Centrifugal Pumps for a large variety of uses; 3. Trane Force-Flo Heaters for quiet heat and neat appearance; 4. Trane Railroad and Bus Air Conditioning Equipment of all kinds; 5. Trane Shell and Tube Heat Exchangers for cooling and heating vapors or liquids in a closed system; 6. Transformer Oil Coolers; 7. Convector-radiators; 8. Heating Coils; 9. Unit Heaters—propeller and blower; 10. Roof Ventilators; 11. Steam Heating Specialties; 12. Hot Water Specialties; 13. Wall-Fin Heaters.

Write today for Trane Condensed Catalog PB290 which describes completely all of the products listed here as well as providing sufficient data for their selection.

YOUNG RADIATOR COMPANY

Air Conditioning Equipment, Cooling Coils, Heating Units

Offices in Principal Cities · Dept. 188 · Racine, Wis., U.S.A.



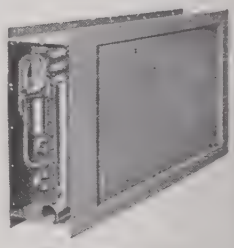
Evaporative Coolers

YOUNG AIR CONDITIONING UNITS are available in eight horizontal and eight vertical styles, in capacities from 400 to 16,625 CFM. Designed to fit every size and type installation for heating, cooling, humidifying, dehumidifying and filtering.

Air Conditioning Units

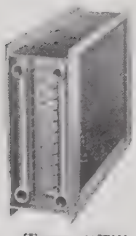
YOUNG EVAPORATIVE COOLERS combine the effect of forced air delivery with that of evaporation. Low water makeup rate is achieved and wind losses are eliminated. Various sizes for any requirement; capable of cooling temperatures within 10° wet bulb.

YOUNG DIRECT EXPANSION EVAPORATORS are designed for mechanical refrigerating systems that use Freon or methyl chloride as the cooling agent. Available in a variety of sizes, these units incorporate the new Young liquid distributor.



Evaporator Coil

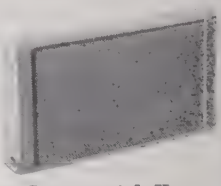
Type "K" Water Coil



Type "W" Water Coil

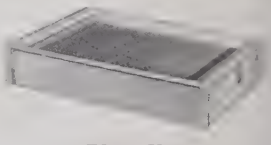
YOUNG COOLING and DEHUMIDIFYING COILS are made to be used with either water or brine. Can be installed for cooling alone or as a part of equipment handling complete or partial air conditioning. Type "K" coils have removable headers to simplify cleaning. Type "W" coils have continuous tubes for multi-pass circulation. Both types are available in a wide choice of sizes and capacities.

YOUNG COMMERCIAL HEAT TRANSFER UNITS provide the perfect heat transfer surface for application with heating and air conditioning installations where a compact and efficient coil is required.



Commercial Heat Transfer Unit

YOUNG BLAST UNITS and BOOSTER UNITS may be used for various applications in connection with forced air heating and conditioning systems. These units are completely encased heat transfer surfaces and are made in a wide range of sizes and capacities.



Blast Unit

YOUNG HEAT TRANSFER PRODUCTS

Young Quality

T. M. REG. U. S. PAT. OFF.

YOUNG RADIATOR CO.
Dept. 000, Racine, Wisconsin, U.S.A.
Sales and Engineering Offices in all Principal Cities

HEATING, COOLING AND AIR CONDITIONING PRODUCTS

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Gas, gasoline, Diesel engine cooling radiators • Jacket water coolers • Heat exchangers • Intercoolers • Condensers • Evaporative coolers • Oil coolers • Gas coolers • Atmospheric cooling and condensing units • Supercharger intercoolers • Aircraft heat transfer equipment

The PYLE-NATIONAL COMPANY

1371 West 37th Street, Chicago 9, Illinois

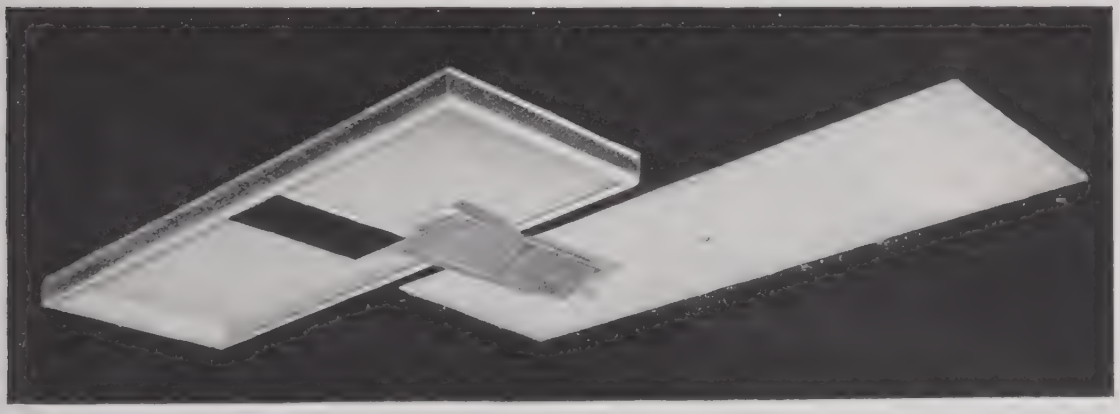
Representatives in Principal Cities

Manufacturers of

MULTI-VENT

PANELS

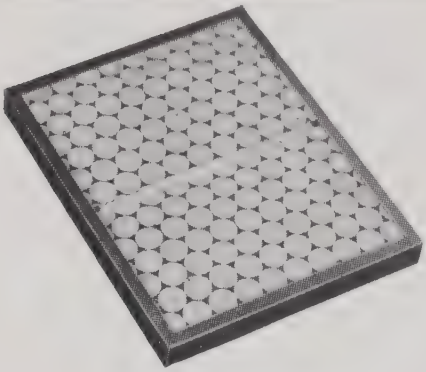
for Heating • Ventilating • Air Conditioning



Control Plate *Valve* *Distribution Plate*

MULTI-VENT Panels are used for the even distribution of warm, cool or fresh air within room spaces. The control plate frame is inserted in the overhead duct at the ceiling. The orificed adjustable air valve provides absolute displacement of static head, and may be set for varying quantities of air to enter the conditioned space, and at varying velocities or speed of entry, producing the comfort zone rate of air motion and degree of temperature desired by the occupant (provided the system includes the required auxiliaries). The perforated distribution plate, of large area, located in the ceiling, accomplishes a wide, gentle, uniform spread and dilution of the conditioned air and simultaneously provides panel heating or cooling. When used with the other correct auxiliaries Multi-Vent results in very uniform temperatures and maximum comfort.

Application data will be furnished on request



DUSTOP

*T. M. REG. U.S. PAT. OFF.

AIR FILTERS

—a FIBERGLAS* *product*

• DUST-STOP replaceable-type air filters provide a highly effective and economical medium for cleaning air of dust, dirt, and lint. They are used wherever air is moved mechanically—in heating, ventilating, and air-conditioning systems and units, as well as on many types of machinery and equipment to protect parts against abrasive dusts.

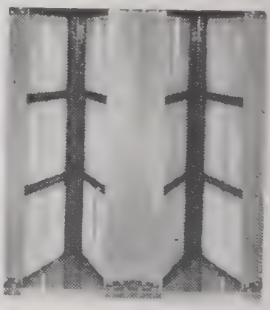
The DUST-STOP Air Filter is constructed of packs of glass fibers (Fiberglas) coated with an adhesive, faced with metal grilles and bound on the edges with a fiberboard frame. The fibers, being glass, are inorganic, chemically stable, resistant to heat, corrosive vapors and most acids . . . do not absorb the nonodorous, nonevaporating adhesive with which they are coated. Each impinged particle of dust is quickly soaked, acting as a wick to carry adhesive to other particles.

Dust-catching efficiency, ease of replacement, low cost and immediate availability of DUST-STOPS are principal reasons why these filters, in standard sizes, are engineered into heating and air-conditioning equipment by leading manufacturers.

FOR LARGE HEATING, VENTILATING AND AIR-CONDITIONING SYSTEMS

To provide maximum installation flexibility for any given CFM requirements, two types of easily assembled steel frames for DUST-STOP Air Filters are available. "L," or straight-front type frame, is used for conventional installations or where plenum depth is limited. "V" type is used where frontal area is restricted.

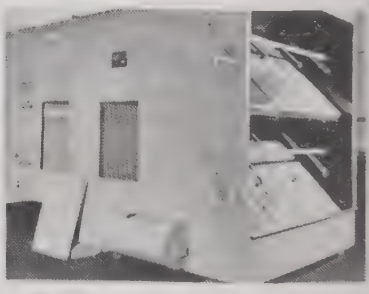
"L" Type Frame



"V" Type Frame

In most residential forced-warm-air furnaces and air conditioners, Dust-Stops are standard equipment.

Dust-Stop Filters are engineered into package air-conditioning units used in commercial establishments.



Fiberglas Aeration Packs are used as contact and eliminator mats in air-washing and humidity control units.

For Complete information see Sweet's Files or write—

OWENS-CORNING FIBERGLAS CORPORATION

2012 Nicholas Bldg., Toledo 1, Ohio

OFFICES: Atlanta, Baltimore, Boston, Buffalo, Chicago, Cincinnati, Cleveland, Columbus, Dallas, Denver, Detroit, Houston, Kansas City, Los Angeles, Milwaukee, Minneapolis, New York, Philadelphia, Pittsburgh, Portland, St. Louis, San Francisco, Seattle, Toledo, Tulsa, Washington, D.C.

AIR DISTRIBUTION PANELS

Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp. Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Pyle-Nat'l. Co., 1371 W. 37th St., Chicago 9, Ill. *(p. 17)*
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.

AIR FILTERS (See also AIR CLEANERS)

Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
Air Devices, Inc., 17 E. 42nd St., N.Y.C. 17
Air Filter Corp., 2514 W. Lisbon Ave., Milwaukee 5, Wis.
Air-Maze Corp., 5200 Harvard Ave., Cleveland 5, O.
American Air Filter Co., Inc., 215 Central Ave., Louisville 8, Ky.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Farr Co., 2615 Southwest Dr., Los Angeles, Cal.
Northern Blower Co., W. 65th St., South of Denison, Cleveland 2, O.
Owens-Corning Fiberglas Corp., 2012 Nicholas Bldg., Toledo 1, O. *(p. 18)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 129)*
Research Products Corp., 1015 E. Washington Ave., Madison 3, Wis. *(p. 19)*
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
H. J. Somers, Inc., 6063 Wabash Ave., Detroit 8, Mich.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
H. O. Threrice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
Universal Air Filter Corp., 332 W. Michigan St., Duluth 2, Minn.

AIR FILTER FRAMES

Air Devices, Inc., 17 E. 42nd St., N.Y.C. 17
Air Filter Corp., 2514 W. Lisbon Ave., Milwaukee 5, Wis.
Dollinger Corp., 1 Centre Park, Rochester 2, N.Y.
Farr Co., 2615 Southwest Dr., Los Angeles, Cal.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Owens-Corning Fiberglas Corp., 2012 Nicholas Bldg., Toledo 1, O. *(p. 18)*
Research Products Corp., 1015 E. Washington Ave., Madison 3, Wis. *(p. 19)*
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.

AIR PURIFICATION EQUIPMENT (See also LAMPS, BACTERICIDAL also ODOR ELIMINATION UNITS & SYSTEMS)

W. B. Connor Engrg. Corp., 114 E. 32nd St., N.Y.C. 16 *(p. 20)*
Hanovia Chemical & Mfg. Co., Chestnut St. & N.J.R.R. Ave., Newark 5, N.J.

AIR RECOVERY SYSTEMS

Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
W. B. Connor Engrg. Corp., 114 E. 32nd St., N.Y.C. 16 *(p. 20)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Tuco Products Corp., 30 Church St., N.Y.C. 7

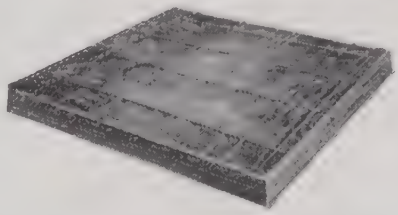
AIR SEPARATORS

W. D. Cashin Co., 69 A St., S. Boston 27, Mass.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Strong, Carlisle & Hammond Co., 1932 W. 3rd St., Cleveland 13, O.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

AIR WASHERS & SPRAY TYPE CONDITIONERS

Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
American Blower Corp., Div. of American Radiator Standard Sanitary Corp., 8111 Tireman Ave., Detroit, 32, Mich.
Bahnson Co., 1001 S. Marshall St., Winston-Salem 7, N.C.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Bayley Blower Co., 66th & Burnham Sts., Milwaukee 14, Wis.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E. Cleveland 14, O.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
E. K. Campbell Heating Co., 1809 Manchester St., Kansas City 3, Mo.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Drying System, Inc., 1810¼ Foster Ave., Chicago 40, Ill
Kauffman Air Conditioning Corp., 4336 W. Pine Blvd., St. Louis, Mo.
D. J. Murray Mfg. Co., 1002-3rd St., Wausau, Wis.
National Engrg. & Mfg. Co., 213 W. 19th St., Kansas City 8, Mo.
Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Ross Sprinkler Co., 34 Roberts St., Pasadena 3, Cal.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Spray Engrg. Co., 114 Central St., Somerville 45, Mass.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Trane Co., La Crosse, Wis. *(p. 14)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
York Corp., York, Pa. *(p. 163)*

RESEARCH
Self-Seal
AIR FILTERS



designed for the Air-Conditioning Engineer

Scientifically designed to give maximum filtering efficiency. Won't shed, pack or settle—removes more dust, pollen, lint by actual test. Write for information on—

RESEARCH SELF SEAL FILTER. CARDBOARD FRAME FILTER. DOMESTIC WASHABLE TYPE. FILTER MEDIA FOR SPECIAL SIZE APPLICATION.

RESEARCH PRODUCTS CORP.
Dept. R MADISON, WIS.

AIR PURIFICATION

THE THIRD DIMENSION IN REFRIGERATED STORAGE CONTROL

Successful applications in apple storages demonstrate that regulating air *quality*, as well as temperature and humidity, extends storage life and preserves quality of produce.

Activated carbon, properly applied, removes ripening gases, disease-causing gases and flavor-impairing odors. In Dorex units, it added 3 to 8 weeks to the life of apples in commercial storages. Tests on other produce show such promising results that commercial installations are now being made.

Gas and odor problem in storage

In seeking ways and means of further extending storage life and generally preserving produce quality, research in recent years has been directed toward a closer scrutiny of storage atmospheres in order to find a more efficient and economical solution to the gas problem.

The gases may merely be the escaping volatile aromas of oranges or onions, or they may be evolved as part of the ripening process as in the case of apples, or bananas, for instance. One of the outstanding features of the problem is the fact that the odors emanating from containers, storage rooms and produce which plague storage men and require costly policies of restriction and segregation also exist in the form of gases.

One of the prime offenders is ethylene, a ripening gas given off by a number of fruit varieties. In apples, for instance, it was found that as little as one percent of ripe fruit in a storage room would accelerate the ripening of other apples exposed to the same air.

Other gases, existing in such minute quantities that so far it has been impossible to identify them, emanate from fruit in storage and cause disease and distortion. Apples, for example, give off gases other than ethylene, which, if allowed to accumulate, will cause the disease known as apple scald.

Efforts to control the accumulation of these gases by dilution with outside air or by wrapping the fruit in mineral-oil impregnated paper has proved burdensome and expensive and, in most cases, has not satisfactorily solved the problem of "gas attack."

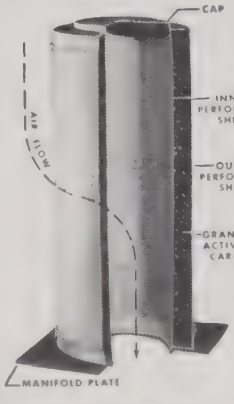
The solution to the whole problem, therefore, lies in finding an inexpensive and dependable way to rid the storage atmosphere of these damaging gases—or the third dimension of storage control —air purification.

Activated carbon traps gases and odors

It was found that the most practical way to remove the gases and odors is by adsorption—a natural phenomenon which occurs when airborne gases and vapors come in contact with the surface area of an adsorbing agent. An instantaneous condensation takes place, and the condensed impurities are held tenaciously until the agent is forced to give them up.

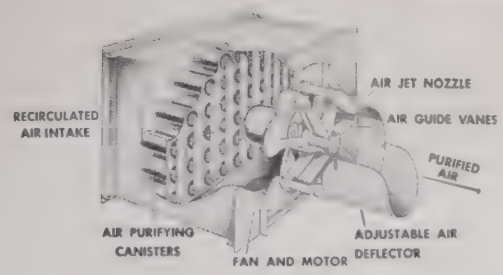
Activated carbon is the most efficient adsorbing agent known. For food storage purposes, however, a specially processed and activated carbon must be used which will meet the following specifications:

1. High activity (adsorptive ca-



Dorex Canister

The compact design of the precision fabricated Dorex Canister at left is the result of many years experience in air purification. The Canister is by far the best means for storage use. It holds the correct amount of carbon in a manner to provide a maximum area for decontamination, a minimum of air resistance, and uniform air flow through the carbon. The thousands of perforations in the surface of the canister subdivide the gas-laden storage air into finely divided streams which enables it to pass easily between the granules of the gas and odor adsorbing carbon.



The Dorex Air Purification Unit above is designed specifically for storages. In order to control the atmosphere adequately and economically, it was engineered to the following specifications:

1. **Constant purification and recirculation of *all* storeroom air.**
2. **Thorough mixing of purified air with storage room air.**
3. **Continuous operation independent of other equipment in the storage space.**
4. **Flexibility in location.**
5. **Self-contained unitary design to eliminate costly duct work or alterations.**

Dorex units are portable and can be floor mounted or hung from walls or ceilings. The directional air jet creates an individual pattern for mixing and distribution and avoids undue air impact on stored produce or fixtures. With the straightening vanes, the amount of "throw" can be adjusted up to a tight jet that reaches 90 feet away from the unit. The large quantity of air thus handled and the high aspiration it creates (five to six times the volume of supply air) results in a very efficient mixing of room and supply air. Dorex Storage Units are built in sizes and capacities to fit any storage space. All parts are either of noncorrosive metal or protected with corrosion-resistant coating.

pacity) for a wide range of gases and vapors in minute quantities.

2. High retentivity over the entire range of normal operating conditions.
3. Low retentivity for water vapor.
4. Extreme hardness to avoid dusting in handling and in service.
5. High apparent density (in the granular form)—not less than 0.45.
6. Adaptability to repeated reactivation without appreciable loss in activity or retentivity.

Outstanding record in apple storages

After more than five years of testing in laboratories and commercial apple storages, Dorex Units are now being used in every fruit district in the country. Reports from large scale users confirmed what the tests had proved, that:

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2. **Scald is reduced.**
3. **The use of oiled paper in packing is eliminated.**
4. **Flavor-impairing odors are removed.**
5. **Apples are more readily salable because of the increased firmness and improved quality of the fruit.**
6. **Continual access to the storage space is permitted for removal and storage of fruit.**

Broad application for other produce predicted

As this is being written (in the fall of 1948 in preparation for publication in the 1949 handbook), tests with a number of other varieties of produce such as pears, citrus fruits, eggs, nut meats and mushrooms, have shown such promising results that installations on a commercial scale have already been started on pears and are planned for others in the near future. Probably by the time you read this, they will already be in and operating successfully. (If you're interested, drop us a note and we'll bring you up to date.)

Since all odors are gases, Dorex Units are being used in increasing numbers as standard "housekeeping" equipment to prevent cross-transfer of odors when "mixed" storage becomes necessary. And by quickly clearing up residual odors, they eliminate the necessity of shutting down and airing out a room after odorous produce has been stored in it.

We have some interesting literature covering the storage applications and research with Dorex Storage Units. Write for your free copies so you can be fully informed on this economical method of extending storage life and protecting food quality.

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Driver-Harris Co., Harrison, N.J.
Duriron Co., Inc., Dayton 1, O.
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Galv-Weld Products, 324 E. 2nd St., Dayton 2, O.
Handy & Harman, 82 Fulton St., N.Y.C. 7
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New Jersey Zinc Co., 160 Front St., N.Y.C. 7
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
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AMMONIA MASKS (See GAS MASKS)

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Dri-Steam Products, Inc., 29 Broadway, N.Y.C. 6
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
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Farr Co., 2615 Southwest Dr., Los Angeles, Cal.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
National Engrg. & Mfg. Co., 213 W. 19th St., Kansas City 8, Mo.
Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Palmer Mfg. Corp., 2200 W. Fillmore, Phoenix, Ariz.
J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Kan. *(p. 81)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 129)*
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
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Tenney Engrg., Inc., 26 Ave. B, Newark 5, N. J.
Thermador Elec'l. Mfg. Co., 5119 District Blvd., Los Angeles, Cal.
Trane Co., La Crosse, Wis. *(p. 14)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*
Young Radiator Co., Racine, Wis. *(p. 16)*

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Thomas Beckett & Co., Inc., 2118 Griffin St., Dallas 2, Tex.
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Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

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Parks-Cramer Co., Box 444, Fitchburg, Mass.
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BACK PRESSURE, etc. (See SUCTION PRESSURE)

BACTERICIDAL LAMPS (See LAMPS, BACTERICIDAL)

BAFFLES (See also COIL & BAFFLE ASSEMBLIES)

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Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*

McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
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Bristol Brass Corp., Bristol, Ct.
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C. G. Hussey & Co., 2860-2nd Ave., Pittsburgh 19, Pa.
Mueller Brass Co., Port Huron, Mich.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*

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Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Henry Disston & Sons, Inc., Tacony, Phila. 35, Pa.
Jessop Steel Co., Green St., Washington, Pa.
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Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Weirton Steel Co., Weirton, W. Va.
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Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
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Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.

BARE IRON PIPE COILS (See also PIPE BENDING)

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Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Cyclops Iron Wks., 837 Folsom St., San Francisco 7, Cal.
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. *(p. 47)*
General Refrigeration Div., Yates-American Machine Co. Beloit, Wis.
Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Joseph E. Lewis & Co., Inc., 1315 Carroll St., Baltimore 30, Md.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
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Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4, O.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 163)*

BARE TUBE COILS
(A—Ammonia B—Other refrigerants)

(A,B) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
(A,B) Bush Mfg. Co., 179 South St., W. Harftord 10, Ct. *(p. 221)*
(A,B) California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
(A,B) Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
(A,B) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(A,B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(A,B) General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
(A,B) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
(B) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga.
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
(B) Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.

(B) McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
C. F. Moores Co., Inc., 1123 Ivy Hill Rd., Wyndmoor, Phila. 18, Pa.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
(A,B) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B) Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
(A,B) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
(B) Reese & Long Refrigeration Products, Inc., 408 E. 25th St., N.Y.C. 10
Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
(A,B) Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
(A,B) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(A,B) Rigidbilt, Inc., 2505 S. Pulaski Rd., Chicago 23, Ill. *(p. 42)*
(A) Roessing Mfg. Co., Sharpsburg St., Pittsburgh, Pa.
(B) Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
(B) B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(A,B) Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
(A,B) Vilter Mfg Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A,B) Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
(A) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 163)*

BAROMETERS

Bristol Co., Waterbury 91, Ct. (Recording only)
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*

BASES, CABINET, COMPRESSOR, etc.

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 124)*
Brasco Mfg. Co., Harvey, Ill.
Falstrom Co., 13 Falstrom Court, Passaic, N.J.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.

BASES, INSTRUMENT

Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Panelyte Div., St. Regis Paper Co., 230 Park Ave., N.Y.C. 17

BASES, MACHINERY

Colonial Iron Wks. Co., 17643 St. Clair Ave., Cleveland 10, O.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Lukenweld, Inc., Coatesville, Pa.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.

BASES, MOTOR

Ackermann Mfg. Co., Wheeling, W. Va.
Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 124)*
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.

BASES, MOTOR, PIVOTED

American Pulley Co., 4200 Wissahickon Ave., Phila. 29, Pa.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7

BASES, VIBRATION ABSORBING (See VIBRATION ABSORBING BASES, etc.)

BASKETS, WIRE (See also WIRE WORK)

Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
Haynes Stellite Co., Unit. of Union Carbide & Carbon Corp., Kokomo, Ind.
Kentucky Metal Products Co., Preston St. & Audubon Park, Louisville 4, Ky.
Locker Engrg. Co., 521 N. La Cienga Blvd., Los Angeles 36, Cal.
Midwest Metal Stamping Co., Kellogg, Ia.
C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
Union Steel Products Co., 448 Pine St., Albion, Mich.
United Steel & Wire Co., Battle Creek, Mich.
L. A. Young Spring & Wire Corp., 9200 Russell Sts., Detroit 11, Mich.

BATCH FREEZERS (See BLAST FREEZERS; also QUICK FREEZERS; also SHARP FREEZERS, etc.)

BAUDELOT COOLERS (See AERATORS; also WATER COOLERS, BAUDELOT TYPE)

BEARING MATERIAL (See also BABBITT METAL)

Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13 (Weld Metal)
Federated Metals Div., American Smelting & Refining Co., 120 Broadway, N.Y.C. 5
Mueller Brass Co., Port Huron, Mich.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.

BEARINGS (See also particular type)

Atlas Brass Foundry, 1901 Santa Fe Ave., Los Angeles 21, Cal.
Cleveland Graphite Bronze Co., 17000 St. Clair Ave., Cleveland 10, O. *(p. 27)*
Federal-Mogul Corp., Shoemaker & Lillibridge Sts., Detroit 13, Mich.
Link-Belt Co., 519 Holmes Ave., Indpls. 6, Ind.
National Lead Co., 111 Broadway, N.Y.C. 6
Nice Ball Bearing Co., 30th & Hunting Park Ave., Phila. 40, Pa.
Randall Graphite Products Corp., 609 W. Lake St., Chicago 6, Ill.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa.
Shenango-Penn Mold Co., Dover, O.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.

BEARINGS, BALL

Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
Bearings Industry Co., 17 W. 60th St., N.Y.C. 23
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Equitable Bearing Co., Inc., 1631 Cottage Grove Ave., Chicago, Ill.
Fafnir Bearing Co., 37 Booth St., New Britain, Ct.

Federal-Mogul Corp., Shoemaker & Lillibridge Sts., Detroit 13, Mich.
Gwilliam Co., 360 Furman St., Brooklyn, N.Y.
Jack & Heintz Precision Industries, Inc., Cleveland 1, O.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Link-Belt Co., 519 Holmes Ave., Indpls. 6, Ind.
Marlin-Rockwell Corp., 402 Chandler St., Jamestown, N.Y.
Messinger Bearings, Inc., D St., above Erie Ave., Phila. 1, Pa.
Miniature Precision Bearings, Inc., Carpenter St., Keene, N.H.
Moraine Products Div., Gen'l. Motors Corp., Dayton 1, O.
Nice Ball Bearing Co., 30th & Hunting Park Ave., Phila. 40, Pa.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.

BEARINGS, BALL, FOR DRAWERS

Locker Engrg. Co., 521 N. La Cienga Blvd., Los Angeles 36, Cal.
Nice Ball Bearing Co., 30th & Hunting Park Ave., Phila. 40, Pa.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa.
E. R. Wagner Mfg. Co., 4001 N 32nd St., Milwaukee, Wis.

BEARINGS, GRAPHITE, OILLESS OR SELF-LUBRICATING, etc.

Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
Atlas Brass Foundry, 1901 Santa Fe Ave., Los Angeles 21, Cal.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Bound Brook Oil-Less Bearing Co., Bound Brook, N.J.
Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Cleveland Graphite Bronze Co., 17000 St. Clair Ave., Cleveland 10, O. *(p. 27)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Graphite Metallizing Corp., 1050 Nepperhan Ave., Yonkers 3, N.Y.
Keystone Carbon Co., Inc., 1935 State St., St. Marys, Pa.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Moraine Products Div., Gen'l. Motors Corp., Dayton 1, O.
Randall Graphite Products Corp., 609 W. Lake St., Chicago 6, Ill.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
U. S. Graphite Co., Saginaw, Mich.

BEARINGS, NEEDLE

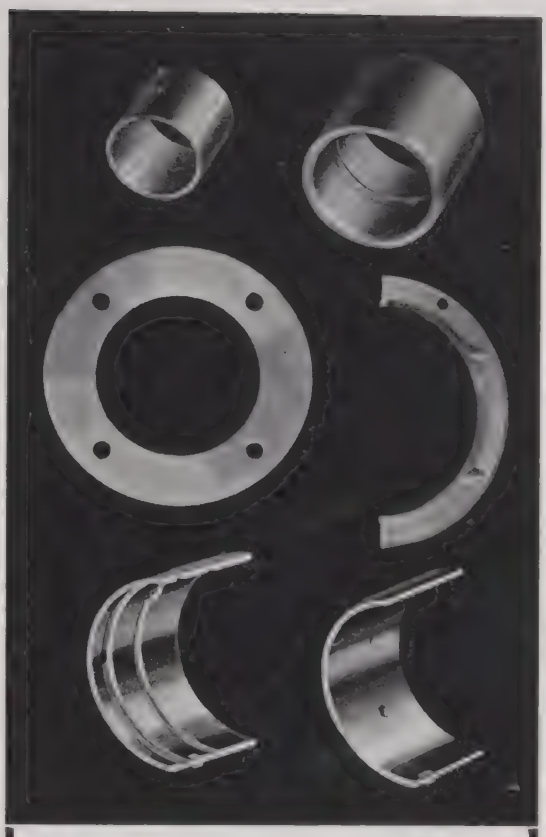
Fafnir Bearing Co., 37 Booth St., New Britain, Ct.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Orange Roller Bearing Co., 557 Main St., Orange, N.J.
Roller Bearing Co. of America, Trenton, N.J.

BEARINGS, RADIAL, THRUST, etc.

Aetna Ball & Roller Bearing Co., 4600 Schubert Ave., Chicago 39, Ill.
Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Cleveland Graphite Bronze Co., 17000 St. Clair Ave., Cleveland 10, O. *(p. 27)*
Equitable Bearing Co., Inc., 1631 Cottage Grove Ave., Chicago, Ill.
Fafnir Bearing Co., 37 Booth St., New Britain, Ct.
Gwilliam Co., 360 Furman St., Brooklyn, N.Y.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
Nice Ball Bearing Co., 30th & Hunting Park Ave., Phila. 40, Pa.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa.
Shenango-Penn Mold Co., Dover, O.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.

BEARINGS, ROLLER

Aetna Ball & Roller Bearing Co., 4600 Schubert Ave., Chicago 39, Ill.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.



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Equitable Bearing Co., Inc , 1631 Cottage Grove Ave., Chicago, Ill.
Fafnir Bearing Co., 37 Booth St., New Britain, Ct.
Gwilliam Co., 360 Furman St., Brooklyn, N.Y.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Hyatt Bearings Div., Gen'l. Motors Corp., Harrison, N.J.
Link-Belt Co., 519 Holmes Ave., Indpls. 6, Ind.
Messinger Bearings, Inc., D St., above Erie Ave., Phila 1, Pa.
Orange Roller Bearing Co., 557 Main St., Orange, N.J.
Roller Bearing Co. of America, Trenton, N.J.
Rollway Bearing Co., Inc., Syracuse, N.Y.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa.
Tyson Bearing Corp., Oberlin Rd., Massillon, O.

BEARINGS, SLEEVE

American Crucible Products Co., Lorain, O.
Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
Atlas Brass Foundry, 1901 Santa Fe Ave., Los Angeles 21, Cal.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Bound Brook Oil-Less Bearing Co., Bound Brook, N.J.
Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Cleveland Graphite Bronze Co., 17000 St. Clair Ave., Cleveland 10, O. *(p. 27)*
R. & J. Dick Co., Inc., Passaic, N.J.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Federal-Mogul Corp., Shoemaker & Lillibridge Sts., Detroit 13, Mich.
Graphite Metallizing Corp., 1050 Nepperhan Ave., Yonkers 3, N.Y.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Link-Belt Co., 2410 W. 18th St., Chicago 8, Ill.
McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.
Moraine Products Div., Gen'l. Motors Corp., Dayton 1, O.
National Formetal Co., 6539 Metta Ave., Cleveland 14, O.
National Lead Co., 111 Broadway, N.Y.C. 6
Randall Graphite Products Corp., 609 W. Lake St., Chicago 6, Ill.
Roller Bearing Co. of America, Trenton, N.J.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa.
Shenango-Penn Mold Co., Dover, O.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.

BEER COOLERS, DRAFT

Kooler Keg Systems, 11 Mill St., Belleville 9, N.J.
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. (p. *147*)

BEER COOLERS, KEG STORAGE & DISPENSING

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A,B) Bemco Mfg. Corp., 1504 Minor at Pike, Seattle 1, Wash.
Horace A. Carter, Inc., 16 E. Marshall St., Richmond 19, Va.
Corbin Cabinet Lock Co., Div. of American Hardware Corp., New Britain, Ct.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 6, Tex.
(A,B,C) Gem Refrigerator Co., 2539 Germantown Ave., Phila. 33, Pa.
(A,B,C) General Refrigerator & Store Fixtures Co., 856 N. Broad St., Phila., Pa.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B) Ideal Cooler Corp., 2953 Easton Ave., St. Louis 6, Mo.
(A) Kooler Keg Systems, 11 Mill St., Belleville 9, N.J.
(A,B,C) La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
(A,B) Master-Bilt Refrigeration Mfg. Co., 920 Palm St., St. Louis 7, Mo.
(A,B,C) Modern Appliance Co., 111 S. Ellsworth, San Mateo, Cal.
(A,B,C) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B) Paley Mfg. Corp., 244 Herkimer St., Brooklyn 16, N.Y.
(A,B) R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
(B,C) St. Louis Butchers Supply Co., 1545 N. 15 St., St. Louis 6, Mo.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(C) Supreme Metal Fabricators, Inc., 27 Rodney St., Brooklyn 11, N.Y.
(B) Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 178)*
(A,B) Uniflow Mfg. Co., Erie, Pa.
(C) United Refrigerator Mfg. Co., Inc., 350 Robert St., St. Paul 1, Minn.

BEER COOLING SYSTEMS

Allied Freezer Corp., 2822 Park Ave., N.Y.C. 51
Bemco Mfg. Corp., 1504 Minor at Pike, Seattle 1, Wash.
Corbin Cabinet Lock Co., Div. of American Hardware Corp., New Britain, Ct.
Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
Gem Refrigerator Co., 2539 Germantown Ave., Phila. 33, Pa.
Heat-X-Changer Co., Inc., 415 Lexington Ave., N.Y.C. 17 *(p. 240)*
Kooler Keg Systems, 11 Mill St., Belleville 9, N.J.
North Penn Co., 72-5th Ave., N.Y.C. 11
Paley Mfg. Corp., 244 Herkimer St., Brooklyn 16, N.Y.
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*

BEER LINE INSULATION (See INSULATION, BEER LINE)

BELLOWS, METALLIC

Bridgeport Thermostat Co., Inc., 1225 Connecticut Ave., Bridgeport 1, Ct.
Chicago Metal Hose Corp., Maywood, Ill.
Clifford Mfg. Co., 564 E. 1st St., Boston 27, Mass. *(p. 29)*
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill
Fulton Sylphon Co., Knoxville, Tenn.

BELLS (See ALARMS)

BELTS, WIRE

Audubon Wire Cloth Corp., Richmond St. & Castor Ave., Phila. 34, Pa.

BELTS & BELTING, FLAT

American Pulley Co., 4200 Wissahickon Ave., Phila. 29, Pa.
Baldwin Belting, Inc., 85 Chambers St., N.Y.C. 7
Bay State Belting Co., 234 Summer St., Boston, Mass.
Buffalo Weaving & Belting Co., 262 Chandler St., Buffalo 7, N.Y.
Chicago Belting Co., 113 N. Green St., Chicago 7, Ill.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
R. & J. Dick Co., Inc., Passaic, N.J.

(Continued)

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First with the Facts on Hydraulically-formed Bellows **HYDRON**

BELTS & BELTING (Continued)

Eagle Belting & Leather Co., 1401 Central Pkwy., Cin'ti. 4, O.
Fisher Leather Belting Co., Inc., 325 N. 3rd St., Phila. 6, Pa.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
L. H. Gilmer Co., 7230 Keystone St., Tacony, Phila. 35, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Holyoke Belting Co., 66 Winter St., Holyoke, Mass.
E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
Hudson Belting Co., 84 E. Worcester St., Worcester 4, Mass.
Manheim Mfg. & Belting Co., Manheim, Pa.
Milwaukee Leather Belting Co., 1114 N. Waters St., Milwaukee 2, Wis.
Missouri Belting Co., 1021 S. Grand Blvd., St. Louis 4, Mo.
Philadelphia Belting Co., 6th & Spring Garden St., Phila. 23, Pa
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
J. E. Rhoads & Sons, 35 N. 6th St., Phila. 6, Pa.
Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7
Ton-Tex Corp., 247 Pearl St., N.W., Grand Rapids 2, Mich.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
John M. Watt's Sons, 112 Walnut St., Phila., Pa.

BELTS & BELTING, ROUND

Baldwin Belting, Inc., 85 Chambers St., N.Y.C. 7
Chicago Belting Co., 113 N. Green St., Chicago 7, Ill.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Eagle Belting & Leather Co., 1401 Central Pkwy., Cin'ti. 4, O.
Fisher Leather Belting Co., Inc., 325 N. 3rd St., Phila. 6, Pa.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
L. H. Gilmer Co., 7230 Keystone St., Tacony, Phila. 35, Pa.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Hudson Belting Co., 85 E. Worcester St., Worcester 4, Mass.
Milwaukee Leather Belting Co., 1114 N. Water St., Milwaukee 2, Wis.
J. E. Rhoads & Sons, 35 N. 6th St., Phila. 6, Pa.
Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7
Ton-Tex Corp., 247 Pearl St., N.W., Grand Rapids 2, Mich.
John M. Watt's Sons, 112 Walnut St., Phila., Pa.

BELTS & BELTING, VEE

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American Pulley Co., 4200 Wissahickon Ave., Phila. 29, Pa.
Baldwin Belting, Inc., 85 Chambers St., N.Y.C.
Browning Mfg. Co., Inc., Maysville, Ky.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Dayton Rubber Mfg. Co., 2342 W. Riverview Ave., Dayton 1, O.
R. & J. Dick Co., Inc., Passaic, N.J.
Eagle Belting & Leather Co., 1401 Central Pkwy., Cin'ti. 4, O.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
L. H. Gilmer Co., 7230 Keystone St., Tacony, Phila. 35, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Hudson Belting Co., 85 E. Worcester St., Worcester 4, Mass.
Manheim Mfg. & Belting Co., Manheim, Pa.
Milwaukee Leather Belting Co., 1114 N. Water St., Milwaukee 2, Wis.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
J. E. Rhoads & Sons, 35 N. 6th St., Phila. 6, Pa.
Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7
Ton-Tex Corp., 247 Pearl St., N.W., Grand Rapids 2, Mich.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
John M. Watt's Sons, 112 Walnut St., Phila., Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

BENDING (See PIPE BENDING also PIPING PREFABRICATED)

BENDING, SMALL PARTS

American Brass Co., Waterbury 88, Ct.
Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
C. F. Moores Co., Inc., 1123 Ivy Hill Rd., Wyndmoor, Phila. 18, Pa.
Standard Valve Mfg. Co., 817 Albany St., Boston 19, Mass.
Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.

BENDS (See RETURN BENDS)

BERYLLIUM COPPER

Beryllium Corp., P.O. Box 1462, Reading, Pa.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.

BEVERAGE COOLERS (See BEER COOLERS also BOTTLED BEVERAGE COOLERS)

BEZELS

L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.

BIMETALS (See THERMOSTATIC BIMETALS)

BLADES, PROPELLER FAN

Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Torrington Mfg. Co., 70 Franklin St., Torrington, Ct. *(p. 105)*

BLAST COILS (See AIR CONDITIONING COILS)

BLAST FREEZERS (See also QUICK FREEZERS, etc.)

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Burge Ice Machine Co., 218 N. Jefferson St., Chicago 6, Ill.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
W. J. Finnegan Co., 7402 Santa Monica Blvd., Los Angeles 46, Cal. *(p. 116)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
King Co., 902 N. Cedar St., Owatonna, Minn.
Refrigeration Corp. of America, Div. of Noma Elec. Corp., 55 W. 13th St., N.Y.C. 11
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.

U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 165)*

BLENDERS, STEAM-WATER, etc.

Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 182)*

BLOCK TIN (See TIN)

BLOOD PLASMA REFRIGERATORS (See REFRIGERATORS, SPECIAL)

BLOWERS (See also particular type & FANS)

Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
E. H. Allen Co., 22 Dorrance St., Boston 29, Mass.
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
Bayley Blower Co., 66th & Burnham Sts., Milwaukee 14, Wis.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
E. K. Campbell Heating Co., 1809 Manchester St., Kansas City 3, Mo.
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Elliott Co., Jeanette, Pa.
Garden City Fan Co., 332 S. Michigan Ave., Chicago 4, Ill.
Grand Rapids Die & Tool Co., 1545 Madison Ave., S.E., Grand Rapids 7, Mich.
Hastings Air Conditioning Co., Inc., Hastings, Neb.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
Jaden Mfg. Co., Hastings, Neb.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
King Co., 902 N. Cedar St., Owatonna, Minn.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Jos. A Martocello & Co., 229 N. 14th St., Phila. 7, Pa. *(p. 157)*
Morrison Products, Inc., 16816 Waterloo Rd., Cleveland 10, O.
Herman Nelson Corp., 1824-3rd Ave., Moline, Ill.
Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
Peerless Elec. Co., 2000 W. Market St., Warren, O.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Redmond Co., Inc., Owosso, Mich.
Robinson Ventilating Co., Zelienople, Pa.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
F. A. Smith Mfg. Co., Inc., Union & Augusta Sts., Rochester 2, N.Y. (Small)
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Trane Co., La Crosse, Wis. *(p. 14)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

BLOWERS, AXIAL FLOW

Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Moore Co., 544 Westport Rd., Kansas City 2, Mo.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

BLOWERS, EXHAUST, VENTILATING, etc.

Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
E. H. Allen Co., 22 Dorrance St., Boston 29, Mass.
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
Bayley Blower Co., 66th & Burnham St., Milwaukee 14, Wis.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Campbell Heating Co., 3121 Dean Ave., Des Moines, Ia.
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7, Ill.
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
Garden City Fan Co., 332 S. Michigan Ave., Chicago 4, Ill.
Grand Rapids Die & Tool Co., 1545 Madison Ave., S.E., Grand Rapids 7, Mich.
Hastings Air Conditioning Co., Inc., Hastings, Neb.
Jaden Mfg. Co., Hastings, Neb.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks. Inc., 128 Linden St., Allentown, Pa
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Mayne Products Co., 324 Harries Bldg., Dayton 2, O.
Moore Co., 544 Westport Rd., Kansas City 2, Mo.
Morrison Products, Inc., 16816 Waterloo Rd., Cleveland 10, O.
National Engrg. & Mg. Co., 213 W. 19th St., Kansas City 8, Mo.
Herman Nelson Corp., 1824-3rd Ave., Moline, Ill.
Peerless Elec. Co., 2000 W. Market St., Warren, O.
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C., 59
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Schwitzer-Cummins Co., Indpls. 7, Ind.
F. A. Smith Mfg. Co., Inc., Union & Augusta Sts., Rochester 2, N.Y. (Small)
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Trane Co., La Crosse, Wis. *(p. 14)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

BLOWER COILS (See UNIT COOLERS)

BLOWER FILTER UNITS

Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Grand Rapids Die & Tool Co., 1545 Madison Ave., S.E., Grand Rapids 7, Mich.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
P. H. MacGirl Foundry & Furnace Wks., 413 E. Oakland Ave., Bloomington, Ill.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.

National Engrg. & Mfg. Co., 213 W. 19th St., Kansas City 8, Mo.
Herman Nelson Corp., 1824-3rd Ave., Moline, Ill.
Peerless Elec. Co., 2000 W. Market St., Warren, O.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Viking Air Conditioning Corp,. 5600 Walworth Ave., Cleveland 2, O.

BLOWER HOUSING & SCROLLS

Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Colonial Iron Wks. Co., 17643 St. Clair Ave., Cleveland 10, O.
Commercial Shearing & Stamping Co., 1775 Logan St., Youngstown, O.
Detroit Stamping Co., 350 Midland Ave., Detroit 3, Mich.
Erie Art Metal Co., 1602 E. 18 St., Erie, Pa.
Grand Rapids Die & Tool Co., 1545 Madison Ave., S.E. Grand Rapids 7, Mich.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc.,128 Linden St., Allentown, Pa.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Mayne Products Co., 324 Harries Bldg., Dayton 2, O.
Morrison Products, Inc., 16816 Waterloo Rd., Cleveland 10, O.
National Engrg. & Mfg. Co., 213 W. 19th St., Kansas City 8, Mo.
Peerless Elec. Co., 2000 W. Market St., Warren, O.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Schwitzer-Cummins Co., Indpls. 7, Ind.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.

BLOWER WHEELS

Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Grand Rapids Die & Tool Co., 1545 Madison Ave., S.E., Grand Rapids 7, Mich.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Mayne Products Co., 324 Harries Bldg., Dayton 2, O.
Morrison Products, Inc., 16816 Waterloo Rd., Cleveland 10, O.
National Engrg. & Mfg. Co., 213 W. 19th St., Kansas City 8, Mo.
Peerless Elec. Co., 2000 W. Market St., Warren, O.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Schwitzer-Cummins Co., Indpls. 7, Ind.
Torrington Mfg. Co., 70 Franklin St., Torrington, Ct. *(p. 103)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.

BODIES, TRUCK (See TRUCK BODIES, REFRIGERATED)

BOLTS (See also particular type)

Aircraft Screw Products Co., Inc., 47-23-35th St., Long Island City 1, N.Y.
Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Clark Bros. Bolt Co., Milldale, Ct.
Columbus Bolt Wks. Co., 291 Marconi Blvd., Columbus 16, O.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Wm. H. Haskell Mfg. Co., 24 Commerce St., Pawtucket, R.I.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C. 5
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
McKaig-Hatch, Inc., 125 Skillen St., Buffalo 7, N.Y.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
William H. Ottemiller Co., Pattison St., & M&P R.R., York, Pa.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

BOLTS, CARRIAGE, STOVE, etc.

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Bethlehem Steel Co., Bethlehem, Pa.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Clark Bros. Bolt Co., Milldale, Ct.
Columbus Bolt Wks. Co., 291 Marconi Blvd., Columbus 16, O.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

BOLTS, EXPANSION

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Chicago Expansion Bolt Co., 1338 W. Concord Place, Chicago 22, Ill.
National Lead Co., 111 Broadway, N.Y.C. 6
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.

BOLTS, LAG

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Bethlehem Steel Co., Bethlehem, Pa.
Clark Bros. Bolt Co., Milldale, Ct.
Columbus Bolt Wks. Co., 291 Marconi Blvd., Columbus 16, O.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Wm. H. Haskell Mfg. Co., 24 Commerce St., Pawtucket, R.I.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

BOLTS, MACHINE

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Bethlehem Steel Co., Bethlehem, Pa.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Clark Bros. Bolt Co., Milldale, Ct.
Columbus Bolt Wks. Co., 291 Marconi Blvd., Columbus 16, O.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Wm. H. Haskell Mfg. Co., 24 Commerce St., Pawtucket, R.I.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

BOLTS, PHILLIPS HEAD

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

BOLTS, STUD

Aircraft Screw Products Co., Inc., 47-23-35th St., Long Island City 1, N.Y.
Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Clark Bros Bolt Co., Milldale, Ct.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Wm. H. Haskell Mfg. Co., 24 Commerce St., Pawtucket, R.I.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
William H. Ottemiller Co., Pattison St., & M&P R.R., York, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

BOLTS, THROUGH

Bethlehem Steel Co., Bethlehem, Pa.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

52nd year making
FASTENERS
THE ATLAS BOLT AND SCREW CO.
BOLT MAKERS *Cleveland 10, Ohio* SINCE '90-
1108 IVANHOE ROAD

BOLTS, TOGGLE

Chicago Expansion Bolt Co., 1338 W. Concord Place, Chicago 22, Ill.
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
U. S. Expansion Bolt Co., York, Pa.

BOLTS, WELD

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Columbus Bolt Wks. Co., 291 Marconi Blvd., Columbus 16, O.
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

BOTTLED BEVERAGE COOLERS

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A,B,C) Airtex Corp., 333 N. Michigan Ave., Chicago 1 Ill.
(A,B,C) American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
(A,B) American Refrigerator & Machine, Inc., 2700 University, N.E., Minneapolis, Minn.
(A) Artkraft Mfg. Corp., Kibby St. & D.T.&I. R.R., Lima, O.
(A) Brunswick-Balke-Collender Co., 623 W. Wabash Ave., Chicago 5, Ill.
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(A,B) Cleveland Refrigerator Co., 2901 E. 65th St., Cleveland 4, O.
(A,B) Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
(A) Crosley Div., Aviation Corp., Cin'ti. 25, O.
(A,B) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
(B) Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila 37, Pa
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 6, Tex.
(A) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(A) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
(A,B,C) General Refrigerator & Store Fixtures Co., 856 N. Broad St., Phila., Pa.
(A,B,C) Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B,C) Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 179)*
(B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(A,B) Ideal Cooler Corp., 2953 Easton Ave., St. Louis 6, Mo.
(A) Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
(A,B) Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
(A,B,C) La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
(A,B,C) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
(A,B) Master-Bilt Refrigeration Mfg. Co., 920 Palm St., St. Louis 7, Mo.
(A,C) Modern Refrigerator Wks., 823 Milford St., Glendale 3, Cal.
(B) Morton Show Cases, Inc., Washington Courthouse, O.
(A) Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
(A,B,C) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B,C) Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
(A) Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.

(A,B) Perfecold, Inc., 1940 S. Main St., Los Angeles 7, Cal.
(A,B) R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
(A,B) Puffer-Hubbard Mfg. Co., Grand Haven, Mich.
(B) W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
Royal Store Fixture Co., 847 N. Broad St., Phila. 23, Pa.
(A) S & S Products, Inc., Lima, O.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
(A,B) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
(B) Star Metal Mfg. Co., Trenton Ave. & Ann St., Phila 34, Pa.
Stoddard Mfg. Co., 617-4th St., S.W., Mason City, Ia.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Supreme Metal Fabricators, Inc., 27 Rodney St., Brooklyn 11, N.Y.
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 178)*
(A,B) Uniflow Mfg. Co., Erie, Pa.
(A,B,C) United Refrigerator Mfg. Co., Inc., 350 Robert St., St. Paul 1, Minn.
(A,B) Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.
Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
(A,B) Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.
(A,B) Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.
(A,B) Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.

BOTTLED BEVERAGE COOLERS, ICE REFRIGERATED

Airtex Corp., 333 N. Michigan Ave., Chicago 1, Ill.
American Commercial Equip Co., 4150 Holly Knoll, Los Angeles 27, Cal.
Horace A. Carter, Inc., 16 E. Marshall St., Richmond 19, Va.
Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 6, Tex.
Gem Refrigerator Co., 2539 Germantown Ave., Phila. 33, Pa.
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 179)*
Ice Cooling Appliance Corp., 33 S. Clark St., Chicago 3, Ill.
La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
Modern Refrigerator Wks., 823 Milford St., Glendale 3, Cal.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
S & S Products, Inc., Lima, O.

BRACKETS (See BASES, MOTOR)

BRACKETS, SHELF

Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.

BRASS (See particular mill form, i.e., BAR, SHEET, etc.)

BRAZING

Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Queen City Steel Treating Co., 2980 Spring Grove Ave., Cin'ti. 25, O. (Elec. Furnace)

BRAZING RODS (See WELDING)

BREAD COOLERS, BAKERY

American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
Baker Perkins, Inc., Saginaw, Mich.
Drying System, Inc., 1810½ Foster Ave., Chicago 40, Ill.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Fred D. Pfening Co., 1075 W. 5th Ave., Columbus 8, O
C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
Union Steel Products Co., 448 Pine St., Albion, Mich.

BREAKER STRIPS

Brasco Mfg. Co., Harvey, Ill.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Micarta Fabricators, Inc., 5324 N. Ravenswood Ave., Chicago 40, Ill.
Panelyte Div., St. Regis Paper Co., 230 Park Ave., N.Y.C. 17
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
Tylac Co., Greeley & High St., Monticello, Ill.

BRINE CIRCULATORS (See AGITATORS)

BRINE COOLERS (See also particular type)

Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Audiffren Refrigerating Co., Proctor, Vt.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R. N.J., Newark 5, N.J. *(p. 239)*
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 240)*
John Nooter Boiler Wks Co., 1426 S. 2nd St., St. Louis 4, Mo.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Reynolds Mfg. Co., Inc., Springfield, Mo.
Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 19, Va.
Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill. (CO_2)
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

BRINE COOLERS, DOUBLE TUBE

Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
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Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
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Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill. (CO_2)
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BRINE COOLERS, MULTI-PASS

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Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
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Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill. (CO_2)
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BRINE COOLERS, SHELL & COIL

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill. (CO_2)
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
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BRINE COOLERS, SHELL & TUBE

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
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Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
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Davis Engrg. Corp., 30 Rockefeller Plaza, N.Y.C. 20
Downingtown Iron Wks., Downingtown, Pa. *(p. 59)*
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John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
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Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill. (CO)
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

BRINE SPRAY UNITS (See UNIT COOLERS)

BRINE TANKS

Arrow Tank Co., Inc., 16 Barnett St., Buffalo 15, N.Y.
Audiffren Refrigerating Co., Proctor, Vt.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Chattanooga Boiler & Tank Co., P.O. Box 110, Chattanooga, Tenn.
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
R. Munroe & Sons Mfg. Corp., 23rd & Smallman Sts., Pittsburgh 22, Pa.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Pittsburgh-Des Moines Construction Co., Neville Island Pittsburgh 25, Pa.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Quaker City Iron Wks., Aramingo Ave. & E. Tioga St., Phila., Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Stover Steel Tank & Mfg. Co., 100 S. Hancock St., Freeport, Ill.
Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.
York Corp., York, Pa. *(p. 163)*

BRINE TREATMENT (See also INHIBITORS)

American Sand-Banum Co., Inc., 9 Rockefeller Plaza, N.Y.C. 20 *(p. 36)*
Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Chemical Solvent Co., Box 487, Birmingham, Ala.
Dearborn Chemical Co., 310 S. Michigan Ave., Chicago Ill.

Solve The Corrosion
Problem In Ice and
Refrigeration Plants With

BLUE SEAL EMULSION SAND-BANUM

NON-POISONOUS
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Use regularly and be assured of corrosion-free brine cans and piping.

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BRINE TREATMENT
MAINTENANCE

WRITE FOR FOLDER

AMERICAN SAND-BANUM COMPANY, Inc.
9 Rockefeller Plaza, New York 20, N.Y.

National Aluminate Co., 6216 W. 66th Place, Chicago 38, Ill *(p. 244)*
North American Fibre Products Co., Standard Bldg., Cleveland 13, O.
Perolin Co., Inc., 10 E.40th St., N.Y.C. 16
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.

BRONZE (See particular mill form, i.e., BAR, SHEET, etc.)

BRONZE, BEARING STOCK

American Crucible Products Co., Lorain, O.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Federal-Mogul Corp., Shoemaker & Lillibridge Sts., Detroit 13, Mich.
Federated Metals Div., American Smelting & Refining Co., 120 Broadway, N.Y.C. 5
Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill.
Shenango-Penn Mold Co., Dover, O.

BRONZE PARTS (See also particular part)

American Crucible Products Co., Lorain, O.
Magnus Brass Mfg. Co., 533 Reading Rd., Cin'ti. 2, O.
P. R. Mallory & Co., Inc., 3029 W. Washington St., Indpls., Ind.
Shenango-Penn Mold Co., Dover, O.

BRUSHES, MOTOR

Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Graphite Metallizing Corp., 1050 Nepperhan Ave., Yonkers 3, N.Y.
Keystone Carbon Co., Inc., 1935 State St., St. Marys, Pa.
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Ohio Carbon Co., 12508 Berea Rd., Cleveland 11, O.
Pure Carbon Co., Inc., 445 Hall Ave., St. Marys, Pa.
Speer Carbon Co., St. Marys, Pa.
Superior Carbon Products Co., 9115 George Ave., Cleveland 5, O.

BULK ICE MAKERS (See ICE MAKERS, CABINET TYPE)

BULLETIN BOARDS

Acromark Co., 5 Morrell St., Elizabeth 4, N.J.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.

BUMPERS, RUBBER

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.

BUNKER SCREENS, REFRIGERATOR CAR

Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
Wheeling Corrugating Co., Wheeling, W. Va.

BUS AIR CONDITIONING SYSTEMS

Keco Industries, Inc., Union Trust Bldg., Cin'ti. 2, O.
Sterling Mfg. Co., 2523 Farnam St., Omaha, Neb.

BUSHINGS (See also particular type and BEARINGS, SLEEVE)

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Aircraft Screw Products Co., Inc., 47-23-35th St., Long Island City 1, N.Y.
Atlas Brass Foundry, 1901 Santa Fe Ave., Los Angeles 21, Cal.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Federal-Mogul Corp., Shoemaker & Lillibridge Sts., Detroit 13, Mich.
Graphite Metallizing Corp., 1050 Nepperhan Ave., Yonkers 3, N.Y.
Lunkenheimer Co., Beekman St., & Waverly Ave., Cin'ti. 14, O.
National Formetal Co., 6539 Metta Ave., Cleveland 14, O.
Shenango-Penn Mold Co., Dover, O.

BUSHINGS, FIBRE

Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Wilmington Fiber Specialty Co., P.O. Drawer 1028, Wilmington 99, Del.

BUSHINGS, PIPE (See also FITTINGS)

Buckeye Iron & Brass Wks., 324 E. 3rd St., Dayton 2, O.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Jarecki Mfg. Co., 1345 W. 12th St., Erie, Pa.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 255)*
Ravena Iron Co., 100 Main St., Ravena, N.Y.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 115)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

BUSHINGS, RUBBER

Bushings, Inc., 4358 Coolidge Highway, Royal Oak, Mich.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Stokes Molded Products, Inc., Taylor At Webster St., Trenton 4, N.J. (Hard)
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

BUSHINGS, RUBBER BONDED

Bushings, Inc., 4358 Coolidge Highway, Royal Oak, Mich.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard)
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

BUTANE

Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp. 30 E. 42nd St., N.Y.C. 17

CABINETS (See listings following, also particular type)

CABINETS, FOR AIR CONDITIONERS

Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y
Erie Art Metal Co., 1602 E. 18 St., Erie, Pa.
Falstrom Co., 13 Falstrom Court, Passaic, N.J.
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Radiant Steel Products Co., 200 Walnut St., Williamsport 28, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6

CABINETS, FOR HOUSEHOLD REFRIGERATORS

American Central Mfg. Corp., Div. of Aviation Corp., 18th & Columbia Sts., Connersville, Ind.
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Hotpoint. Inc., 5600 W. Taylor St., Chicago 44, Ill.
Rex Mfg. Co., Inc., Western Ave., Connersville, Ind.
Sanitary Refrigerator Co., Fond du Lac, Wis. *(p. 187)*
Seeger Refrigerztor Co., 850 Arcade St., St. Paul 6, Minn.
Stoddard Mfg. Co., 617-4th St., S.W., Mason City, Ia.
Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.

CABINETS, FREEZING (See HOME & FARM FREEZERS)

CABINETS, FROZEN FOOD (See HOME & FARM FREEZERS)

CABINETS, KEY

Locker Engrg. Co., 521 La Cienga Blvd., Los Angeles 36, Cal.

CABINETS, SEED GERMINATING

Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.

CABINET PARTS (See also particular part)

Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown N.Y.
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 215)*
Lindsay Corp., 1740-25th Ave., Melrose Park, Ill.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.
Motors Metal Mfg. Co., 5936 Milford Ave., Detroit 10, Mich.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*
Stanley Wks., 195 Lake St., New Britain, Ct.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.

CABINET TOPS & LIDS

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Dryden Rubber Co., 1014 S. Kildare Ave., Chicago 24, Ill.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
Luzerne Rubber Co., Trenton 9, N.J.
Panelyte Div., St. Regis Paper Co., 230 Park Ave., N.Y.C. 17
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.

CABLE, ELECTRIC (See also WIRE, ELECTRIC)

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
National Elec. Products Corp., Fulton Bldg., Pittsburgh 22, Pa.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

CALCIUM CHLORIDE

Barada & Page, Inc., Guinotte & Michigan Aves., Kansas City 1, Mo.
Henry Bower Chemical Mfg. Co., Gray's Ferry Rd. & 29th St., Phila. 46, Pa.
Columbia Chemical Div., Pittsburgh Plate Glass Co., 5th Ave. at Bellefield, Pittsburgh 13, Pa.
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Michigan Alkali Div., Wyandotte Chemicals Corp., Wyandotte, Mich.
Pennsylvania Salt Mfg. Co., 1000 Widener Bldg., Phila. 7, Pa.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Solvay Sales Div., Allied Chemical & Dye Corp., 40 Rector St., N.Y.C. 6
Tamms Silica Co., 228 N. La Salle St., Chicago 1, Ill.

CALCIUM SULFATE, ANHYDROUS (See DEHYDRANTS)

CAN FILLERS (See ICE MANUFACTURING ACCESSORIES)

CANDY CASES, REFRIGERATED

General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
C. F. Moores Co., Inc., 1133 Ivy Hill Rd., Wyndmoor, Phila. 18, Pa.
Refrigeration Corp. of America, Div. of Noma Elec. Corp., 55 W. 13th St., N.Y.C. 11

CANS, ICE (See ICE CANS)

CANVAS & CANVAS SPECIALTIES (See also DUCK)

Lehon Co., 4425 S. Oakley Ave., Chicago 9, Ill.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Webb Mfg. Co., 4th & Cambria Sts., Phila. 33, Pa.

CAPACITORS, ELECTRICAL

Aerovox Corp., New Bedford, Mass.
Cornell-Dubillier Elec. Corp., S. Plainfield, N.J
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
P. R. Mallory & Co., Inc., 3029 E. Washington St. Indpls., Ind.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Sprague Products Co., Marshall St., N. Adams, Mass.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

CAPS, WELDING

General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Walworth Co., 60 E. 42nd St., N.Y.C. 17

CARBON DIOXIDE (See also DRY ICE)

Air Reduction Sales Co., 60 E. 42nd St., N.Y.C. 17
Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
Mathieson Alkali Wks., Inc., 60 E. 42nd St., N.Y.C. 17

CARBON DIOXIDE ACCESSORIES

Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.

CARBON PRODUCTS (See also GRAPHITE PRODUCTS)

Ohio Carbon Co., 12508 Berea Rd., Cleveland 11, O.
Pure Carbon Co., Inc., 445 Hall Ave., St. Marys, Pa.
Speer Carbon Co., St. Marys, Pa.

CARBONATORS & CARBONATING EQUIPMENT

Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Hudson Products Co., Inc., 4400 St. Aubin, Detroit 7, Mich.
Liquid Carbonic Corp., 3100 S. Kezdie Ave., Chicago 23, Ill.
Paley Mfg. Corp., 244 Herkimer St., Brooklyn 16, N.Y.
Spacarb, Inc., 311 E. 23rd St., N.Y.C. 10
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*

CARTS, MATERIAL HANDLING (See TRUCKS)

CASTINGS (See also particular type)

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Bethlehem Steel Co., Bethlehem, Pa.
A. S. Campbell & Co., Inc., E. Boston 28, Mass.
Connecticut Precision Casting Co., New Canaan, Ct.
Haynes Stellite Co., Unit. of Union Carbide & Carbon Corp., Kokomo, Ind.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Jessop Steel Co., Green St., Washington, Pa.
Lakeside Malleable Castings Co., Racine, Wis.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Niles Tool Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Saginaw Malleable Iron Div., Gen'l. Motors Corp., Saginaw, Mich.
Shenango-Penn Mold Co., Dover, O.
Universal Foundry Co., 86 Pine St., Oshkosh, Wis.
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 4

CASTINGS, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Bound Brook Oil-Less Bearing Co., Bound Brook, N.J.
Canton Malleable Iron Co., 2400-13th St., N.E., Canton 5, O.
Cleveland Hardware & Forging Co., 3270 E. 79th St., Cleveland 4, O.
Connecticut Precision Casting Co., New Canaan, Ct.
Heat-X-Changer Co., Inc., 415 Lexington Ave., N.Y.C. 17 *(p. 240)*
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Red Spot Elec. Co., Tacoma, Wash.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
Universal Foundry Co., 86 Pine St., Oshkosh, Wis.

CASTINGS, BRASS, BRONZE, etc.

Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
American Crucible Products Co., Lorain, O.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Bethlehem Steel Co., Bethlehem, Pa.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Bound Brook Oil-Less Bearing Co., Bound Brook, N.J.
Bunting Brass & Bronze Co., 715 Spencer St., Toledo 9, O.
Canton Malleable Iron Co., 2400-13th St., N.E., Canton 5, O.
Connecticut Precision Casting Co., New Canaan, Ct.
Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Magnus Brass Mfg. Co., 533 Reading Rd., Cin'ti. 2, O.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Mueller Brass Co., Port Huron, Mich.
National Lead Co., 111 Broadway, N.Y.C. 6
Newman Bros., Inc., 666 W. 4th St., Cin'ti. 3, O.
Niles Tool Wks. Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Red Spot Elec. Co., Tacoma, Wash.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
Shenango-Penn Mold Co., Dover, O.
Universal Foundry Co., 86 Pine St., Oshkosh, Wis.
J. A. Zurn Mfg. Co., Erie, Pa.

CASTINGS, DIE

AC Spark Plug Div., Gen'l. Motors Corp., Flint 2, Mich.
Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
Badger Die Casting Corp., 201 W. Oklahoma Ave., Milwaukee 7, Wis.
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Chicago Expansion Bolt Co., 1338 W. Concord Place, Chicago 22, Ill.
Cleveland Hardware & Forging Co., 3270 E. 79th St., Cleveland 4, O.
Dollin Corp., 600 S. 21st St., Irvington 11, N.J.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich. *(p. 186)*
Jessop Steel Co., Green St., Washington, Pa.
Milwaukee Die Casting Co., 1021 N. 4th St., Milwaukee 3, Wis.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Sterling Die Casting Co., Inc., 743-39th St., Brooklyn 32, N.Y.
Titan Metal Mfg. Co., Bellefonte, Pa.

CASTINGS, LEAD

Alpha Metals, Inc., 363 Hudson Ave., Brooklyn 1, N.Y.
Belmont Smelting & Refining Wks., 330 Belmont Ave., Brooklyn 7, N.Y.
Chicago Expansion Bolt Co., 1338 W. Concord Place, Chicago 22, Ill.
National Lead Co., 111 Broadway, N.Y.C. 6

CASTINGS, MONEL; NICKEL & NICKEL ALLOY

Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C. 5
Shenango-Penn Mold Co., Dover, O.
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 4

CASTINGS, STEEL & IRON

Alloy Cast Steel Co., Rose Ave., Marion, O.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Bethlehem Steel Co., Bethlehem, Pa.
Canton Malleable Iron Co., 2400-13th St., N.E., Canton 5, O.
Connecticut Precision Casting Co., New Canaan, Ct.
Cooper-Bessemer Corp., Mt. Vernon, O.
Dayton Malleable Iron Co., P.O. Box 980, Dayton 1, O.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Jessop Steel Co., Green St., Washington, Pa.
Lakeside Malleable Castings Co., Racine, Wis.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
P. H. MaGirl Foundry & Furnace Wks., 413 E. Oakland Ave., Bloomington, Ill.
National Malleable & Steel Castings Co., 10600 Quincy Ave., Cleveland 6, O.
Niles Tool Wks. Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Pratt & Letchworth Co., 189 Tonawanda St., Buffalo 7 N.Y.
Saginaw Malleable Iron Div., Gen'l. Motors Corp., Saginaw, Mich.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
Shenango-Penn Mold Co., Dover, O.
Terre Haute Malleable & Mfg. Corp., 2030 N. 19th, Terre Haute, Ind.
Union Mfg. Co., New Britain, Ct.
Universal Foundry Co., 86 Pine St., Oshkosh, Wis.
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 4
J. A. Zurn Mfg. Co., Erie, Pa.

CEMENT, INSULATING

Airtex Corp., 333 N. Michigan Ave., Chicago 1, Ill.
American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Carney Co., Inc., 125 S. 5th St., Minneapolis 2, Minn.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
Locker Engrg. Co., 521 N. La Cienga Blvd., Los Angeles 36, Cal.
Mineral Insulation Co., 103rd & Southwest Highway, Chicago Ridge, Ill.
Munn and Steele, Inc., 130 Lister Ave., Newark 5, N.J.
National Gypsum Co., 325 Delaware Ave., Buffalo 2, N.Y. *(p. 133)*
Owens-Corning Fiberglas Corp., 2012 Nicholas Bldg., Toledo 1, O. *(p. 134)*
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Refractory & Insulation Corp., 120 Wall St., N.Y.C. 5
Robinson Insulation Co., Great Falls, Mont.
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Universal Zonolite Insulation Co., 135 S. La Salle St., Chicago 3, Ill. *(p. 135)*

CEMENT, RUBBER

Chase Industrial Refrigerator Equip. & Engrg. Co., 630 Reading Rd., Reading, Cin'ti. 15, O.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.

CEMENTS (See also ADHESIVES)

Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
B. F. Nelson Mfg. Co., 401 Main St., N.E., Minneapolis 13, Minn.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.

CHAIN, CONVEYOR, DRIVE, ROLLER, etc.

Baldwin Duckworth Div., Chain Belt Co., 369 Plainfield St., Springfield, Mass.
Conveyor Co., 3260 E. Slauson Ave., Los Angeles 11, Cal.
Corbin Screw Corp., New Britain, Ct.
Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp. Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Diamond Chain Co., Inc., 402 Kentucky Ave., Indpls. 7, Ind.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
Morse Chain Co., Div. of Borg-Warner Corp., Ithaca, N.Y.
Ramsey Chain Co., Inc., 900 Broadway, Albany 1, N.Y
Transue & Williams Steel Forging Corp., Alliance, O.
Watson Flagg Machine Co., 845 E. 25th St., Paterson 3, N.J.

CHANNELS (See also SHAPES; SHAPES, STRUCTURAL)

Bethlehem Steel Co., Bethlehem, Pa.
Brasco Mfg. Co., Harvey, Ill.
Carnegie-Illinois Steel Corp., U. S. Steel Corp., Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala

CHARGING LINES (See also FLEXIBLE CONNECTIONS; also TUBING, FLEXIBLE)

Aeroquip Corp., 300 S. East Ave., Jackson, Mich. *(p. 83)*
American Brass Co., Waterbury 88, Ct.
Chicago Metal Hose Corp., Maywood, Ill.
Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill. *(p. 55)*
Henry Valve Co., Melrose Park, Ill. *(p. 106)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Seamlex Co., Inc., 4123-24th St., Long Island City 1, N.Y. *(p. 218)*
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p 108)*
Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill.
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
York Corp., York, Pa. *(p. 163)*

CHECK VALVES, AIR, WATER

Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
James B. Clow & Son, 201 N. Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.

Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
R. & J. Dick Co., Inc., Passaic, N.J.
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Edward Valves, Inc., Subsidiary of Rockwell Mfg. Co., 1200 W. 145 St., E. Chicago, Ind.
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Gas & Oil Industry Labs., Inc., 4 Paine Ave., Irvington 11, N.J.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
Hancock Valve Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Jarecki Mfg. Co., 1345 W. 12th St., Erie, Pa.
Jenkins Bros., 80 White St., N.Y.C. 13
Kennedy Valve Mfg. Co., Elmira, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Lunkenheimer Co., Beekman St., & Waverly Ave., Cin'ti. 14, O.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Ross Sprinkler Co., 34 Roberts St., Pasadena 3, Cal.
Swaby Mfg. Co., 2330 W. Cermak Rd., Chicago 8, Ill.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
White Flomatic Corp., P.O. Box 267, Hoosick Falls, N.Y.
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 12, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
J. A. Zurn Mfg. Co., Erie, Pa.

CHECK VALVES, REFRIGERANT
(A—Ammonia; B—Other refrigerants)

(B) Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
(A,B) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
(A,B) Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
(A) Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
(A) Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
(B) Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill. *(p. 55)*
(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(A,B) Henry Valve Co., Melrose Park, Ill. *(p. 106)*
(A,B) Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
(B) Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
(B) Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
(B) Mueller Brass Co., Port Huron, Mich.
(B) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(B) Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A) Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
(A,B) Watson-Stillman Co., Roselle, N.J. *(p. 113)*
(B) Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 163)*
J. A. Zurn Mfg. Co., Erie, Pa.

CHILLERS, CONTINUOUS

Votator Div., Girdler Corp., Louisville 1, Ky. *(p. 71)*

CIRCUIT BREAKERS

Allen-Bradley Co., Milwaukee 4, Wis.
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
I.T.E. Circuit Breaker Co., 19th & Hamilton St., Phila., Pa.
Roller-Smith Div., Realty & Industrial Corp., Bethlehem, Pa.
F. A. Smith Mfg. Co., Inc., Union & Augusta Sts., Rochester 2, N.Y.
Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

CIRCULATORS, AIR (See FANS, CIRCULATING)

CLAMPS, CONDUIT & CABLE (See also FITTINGS, ELECTRIC CONDUIT)

General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
Tinnerman Products, Inc., 2038 Fulton Rd., Cleveland 13, O.

CLAMPS, PIPE (See also HANGERS, PIPE also SUPPORTS)

California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
U. S. Expansion Bolt Co., York, Pa.
Walworth Co., 60 E. 42nd St., N.Y.C. 17

CLIPS (See also FASTENERS; also SUPPORTS)

CLOCKS

Ashcroft Gauge Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
Marshalltown Mfg. Co., Marshalltown, Ia.

CLOSERS, COLD STORAGE DOOR

Chicago Spring Hinge Co., 1500 Carroll Ave., Chicago 7, Ill.
Jamison Cold Storage Door Co., Hagerstown, Md. *(p. 45)*
Kason Hardware Corp., 127 Wallabout St., Brooklyn 6, N.Y.
Streator Products Corp., 508 N. 8th St., Fairfield, Ia.
York Corp., York, Pa. *(p. 163)*

CLOTH, WIRE (See WIRE CLOTH)

CLUTCHES

Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Hilliard Corp., 102 W. 4th St., Elmira, N.Y.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Mercury Clutch Corp., 1520-12th St., S.W., Canton 6, O.
Morse Chain Co., Div. of Borg-Warner Corp., Ithaca, N.Y.
Stearns Magnetic Mfg. Co., 667 S. 28 St., Milwaukee 4, Wis. (Magnetic)
Twin Disc Clutch Co., Racine, Wis.
Watson Flagg Machine Co., 845 E. 25th St., Paterson 3, N.J.

COATINGS & COMPOUNDS, PROTECTIVE, PLASTIC, etc. (See also FINISHES)

American Chemical Paint Co., Ambler, Pa. *(p. 41)*
American Pipe & Construction Co., Box 3428 Terminal Annex, Los Angeles 54, Cal.
Brooklyn Varnish Mfg. Co., Inc., 50 Jay St., Brooklyn 1, N.Y.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Chemical Solvent Co., Box 487, Birmingham, Ala.
Colonial Alloys Co., Ridge Ave. & Crawford St., Phila. 29, Pa.
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Ernecke & Salmstein Co., 1611 N. Sheffield Ave., Chicago 14, Ill.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.

Metal Treating Chemicals

FOR CORROSION-PREVENTION AND PAINT-BONDING ON STEEL, ALUMINUM AND GALVANIZED IRON

"GRANODINE"
"DURIDINE"
"ALODINE"
"ZINODINE"
"LITHOFORM"
"DEOXIDINE"

Details will be sent on request.

American Chemical Paint Co.

AMBLER, CHEMICALS ACP PROCESSES PENNA.

Galv-Weld Products, 324 E. 2nd St., Dayton 2, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Grand Rapids Varnish Corp., 1350 Steele Ave S.W., Grand Rapids 2, Mich.
Haveg Corp., Marshallton, Del.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
A. C. Horn Co., Inc., 43-36-10th St., Long Island City, N.Y.
Inertol Co., Inc., 470 Frelinghuysen Ave., Newark 5, N.J.
B. F. Nelson Mfg. Co., 401 Main St., N.E., Minneapolis 13, Minn.
Nox-Rust Chemical Corp., 2429 S. Halsted St., Chicago 8, Ill.
Oakite Products, Inc., 22 Thamex St., N.Y.C. 6
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Quigley Co., Inc., 427-5th Ave., N.Y.C. 17
Reilly Tar & Chemical Corp., 1615 Merchants Bank Bldg. Indpls. 4, Ind.
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
L. Sonneborn Sons, Inc., 88 Lexington Ave., N.Y.C. 16
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
Tuco Products Corp., 30 Church St., N.Y.C. 7
United Chromium, Inc., 51 E. 42nd St., N.Y.C. 17
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.
Wailes Dove-Hermiston Corp., 448 South Ave., Westfield, N.J.
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
Wilbur-Williams Co., 43 Leon St., Boston, Mass.
X-Pando Corp., 43-15-36th St., Long Island City 1, N.Y.

COCKS

Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Jerguson Gage & Valve, 87 Fellsway, Somerville, Mass.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Mueller Co., 512 W. Cerro Gordo St., Decatur 70, Ill.
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

COIL & BAFFLE ASSEMBLIES
(A—Ammonia; B—Other refrigerants)

American Coils Co., 25 Lexington St., Newark 5, N.J.
(A,B) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
(A,B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
Cold Control, Inc., 111 Broadway, N.Y.C. 6
(B) Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
(A,B) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(B) Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
(B) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
(B) McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
(B) Melco Mfg. Co., Inc., Grand Ave., Ridgefield, N.J.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
(A,B) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(B) Paley Mfg. Corp., 244 Herkimer St., Brooklyn 16, N.Y.
(A,B) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(A) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
(B) Reese & Long Refrigeration Products, Inc., 408 E. 25th St., N.Y.C. 10
Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
(A,B) St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
(B) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(A,B) Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A,B) Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

COIL STOCK (See STRIP)

COILS (See particular type, i.e., FINNED COILS PLATE COILS; UNIT COOLERS; etc.)

COILS, SPECIAL
(A—Ammonia; B—Other refrigerants)

(A,B) Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*

COILS, SPECIAL (Continued)

(A,B) Aerofin Corp., 410 Geddes St., Syracuse, N.Y. *(p. 11)*
(B) Allied Freezer Corp., 2822 Park Ave., N.Y.C. 51
(A,B) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
(B) Betz Corp., 445 State St., Hammond, Ind.
(A,B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
(A) California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
(B) Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
(A,B) Crane Co., 836 Michgan Ave., Chicago 5, Ill. *(p. 109)*
(A,B) Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
(A,B) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(A,B) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(A,B) General Refrigeration Div., Yates-American Machine Co., Beloit, Wis
(A,B) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 224)*
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
(B) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
(B) Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
(B) Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
(B) McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
(B) Melco Mfg. Co., Inc., Grand Ave., Ridgefield, N.J.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
(A,B) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B) Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
(A) John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
(B) Paley Mfg. Corp., 244 Herkimer St., Brooklyn 16, N.Y.
(A) Pittsburgh Pipe Coil & Bending Co., 61 Bridge St. Etna P.O., Pittsburgh 23, Pa.
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
(B) Reese & Long Refrigeration Products, Inc., 408 E. 25th St., N.Y.C. 10
(A,B) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
(A,B) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 123)*
(A,B) Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
(A,B) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(A,B) Rigidbilt, Inc., 2505 S. Pulaski Rd., Chicago 23, Ill. *(p. 42)*
(A) Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
(B) Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59
(B) Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
(A,B) St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
John Simons Co., Inc., 50 Church St., N.Y.C. 7
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
(B) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(A,B) Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.

(Continued)

HEAT EXCHANGE EQUIPMENT

FOR ALL TEMPERATURES

★ CUSTOM ENGINEERED COILS
FOR ALL REFRIGERANTS

★ UNIT COOLERS
STANDARD • WATER DEFROST

★ PRODUCT COOLERS
FOR HIGH & LOW TEMPERATURE

★ BRINE SPRAY UNITS

★ AIR CONDITIONING UNITS
FLOOR AND CEILING TYPE

★ AIR CONDITIONING COILS
DIRECT EXPANSION & WATER

★ HEATING COILS

★ SPECIAL COILS
DESIGNED FOR PROCESS WORK

RIGIDBILT INC.

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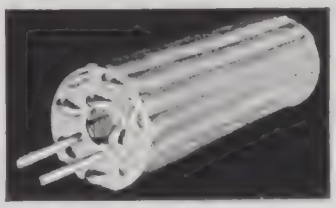
ROME

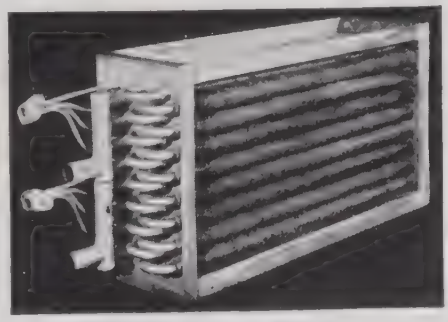
ROME Helical Fin Tubing. Over 100 sizes for all air heating and air cooling equipment—Straight lengths—U Bends—Continuous Coils.

ROME Water Cooled Condenser Coils. High efficiency, jointless construction. Manufactured to customers' specifications.



ROME Water Cooled Condensers—Tube within a tube type. Complete range of sizes.

ROME also manufactures: Air Cooled Condensers, Oil Coolers, Blast Air Cooling Coils, Blast Air Heating Coils, Concealed Copper Radiation (Convectors) and other similar Heat Transfer Equipment.



Specify ROME Heat Transfer Products for Satisfactory Performance

The Rome-Turney Radiator Co.

Rome, N.Y.

COILS, SPECIAL (Continued)

(B) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A) Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
(A,B) Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 163)*

COLD CONTROLS (See THERMOSTATS)

COLD STORAGE DOORS & WINDOWS

American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
Butcher Boy Cold Storage Door Co., Div. of Western Fixture & Equip. Co., 170 N. Sangamon St., Chicago 7, Ill.
Cleveland Refrigerator Co., 2901 E. 65th St., Cleveland 4, O.
Corbin Cabinet Lock Co., Div. of American Hardware Corp., New Britain, Ct.
Cork Insulation Co. Inc., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
Cruse Refrigerator Co., Inc., 504 Main St., Louisville 2, Ky.
Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Frosty Foods Equip. Co., 305 Benson Blvd., Sioux City 15, Ia.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
Jamison Cold Storage Door Co., Hagerstown, Md. *(p. 45)*
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
Masterfreeze Corp., Sister Bay, Wis.
Matthews Refrigerator & Door Co., 5103 S.E. Powell Blvd., Portland 6, Ore.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.
C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.
Streator Products Corp., 508 N. 8th St., Fairfield, Ia.
Sub Zero, Inc., 41 E. 42nd St., N.Y.C. 17
York Corp., York, Pa. *(p. 163)*

COLD STORAGE DOOR HARDWARE (See HARDWARE, COLD STORAGE DOOR; also CLOSERS)

COLLARS, SHAFT

Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.

COLUMNS, WATER (See GAUGE GLASSES)

COMPARATORS, CHROMATE, pH, etc.

National Aluminate Co., 6216 W. 66th Place, Chicago 38, Ill. *(p. 244)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*

COMPOUNDS, CAULKING

Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Black Swan Mfg. Co., 2235 W. Grand Ave., Chicago 12, Ill.
Chamberlin Co. of America, 1254 LaBrosse St., Detroit 26, Mich.
Jesse G. Croll & Son, 2606 Germantown Ave., Phila. 33, Pa.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hetzel Roofing Products Co., 67 Main St., Newark, N.J.
Lehon Co., 4425 S. Oakley Ave., Chicago 9, Ill.
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O.
B. F. Nelson Mfg. Co., 401 Main St., N.E., Minneapolis 13, Minn.
Pecora Paint Co., Inc., 3rd & Lawrence Sts., Phila. 40, Pa. (Also Glazing)
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Presstite Engrg. Co , 3900 Chouteau Ave., St. Louis 10, Mo.
Reilly Tar & Chemical Corp., 1615 Merchants Bank Bldg., Indpls. 4, Ind.
L. Sonneborn Sons, Inc., 88 Lexington Ave., N.Y.C. 16
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
Tamms Silica Co., 228 N. La Salle St., Chicago 1, Ill.
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
X-Pando Corp., 43-15-36th St., Long Island City 1, N.Y.

COMPOUNDS, CORROSION INHIBITING (See BRINE TREATMENT; also INHIBITORS, CORROSION; also WATER TREATING)

COMPOUNDS, GASKET, PIPE JOINT, etc.

Arco Co., 7301 Bessemer Ave., Cleveland 4, O.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada.
Black Swan Mfg. Co., 2235 W. Grand Ave., Chicago 12, Ill.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Crane Packing Co,. 1800 Cuyler Ave., Chicago 13, Ill.
Jesse G. Croll & Son, 2606 Germantown Ave., Phila. 33, Pa.
Dearborn Chemical Co., 310 S. Michigan Ave., Chicago, Ill.
Federal Process Co., 501 Erie Bldg., Cleveland 15, O.
Galv-Weld Products, 324 E. 2nd St., Dayton 2, O.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
Goetze Branch, Johns-Manville Products Corp., Allen Ave., New Brunswick, N.J.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hetzel Roofing Products Co., 67 Main St., Newark, N.J.
A. C. Horn Co., Inc., 43-36-10th St., Long Island City, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Macksons Co., 127 Cedar St., N.Y.C. 6
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Pecora Paint Co., Inc., 3rd & Lawrence Sts., Phila. 40, Pa.
Permatex Co., Inc., 1720 Ave. Y, Brooklyn 29, N.Y.
Presstite Engrg. Co., 2900 Chouteau Ave., St. Louis 10, Mo.
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Radiator Specialty Co., 1700 Dowd Rd., Charlotte, N.C.
Reilly Tar & Chemical Corp., 1615 Merchants Bank Bldg., Indpls. 4, Ind.
Smooth-On Mfg. Co., 572 Communipaw Ave., Jersey City 4, N.J.
Star Products Co., 130 Walnut Ave., N.Y.C. 54
Tuco Products Corp., 30 Church St., N.Y.C. 7
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
Utility Lab., Inc., 330-36th St., Brooklyn 32, N.Y.
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.
"X" Labs., Inc., 25 W. 45th St., N.Y.C. 19
X-Pando Corp., 43-15-36th St., Long Island City 1, N.Y.

COMPOUNDS, MASTIC (See also ENAMEL, BITUMASTIC)

American Bitumuls Co., 200 Bush St., San Francisco 4, Cal.

(Continued)

JAMISON COLD STORAGE DOOR CO.

Oldest and Largest Builders of Cold Storage Doors in the World

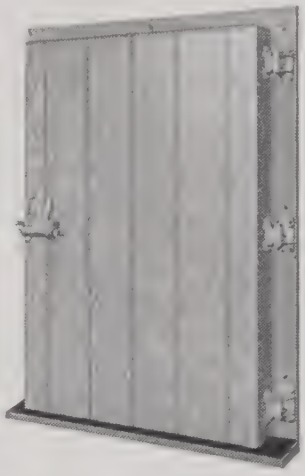
Hagerstown, Maryland

SALES BRANCHES AND DISTRIBUTORS IN PRINCIPAL CITIES

◀ JAMISON LO-TEMP DOOR

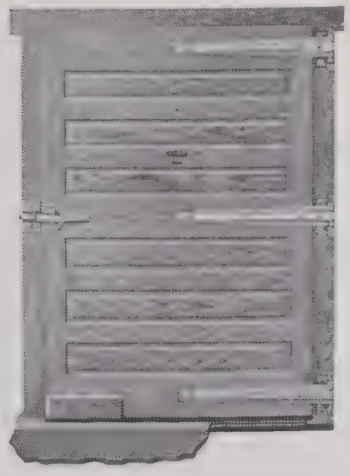
Tested under rigorous industrial use for several years. This door was developed for sub-freezing and sharp freezing operations where an infitting door is desired. Furnished with Lo-Temp gasket (sponge-rubber core with a water impervious skin), an exceptionally efficient seal is achieved. This gasket not only assures a uniform seal against the frame casing but plug seals the space between door and frame. Equipped with Jamison E-Z Open Two Point Fasteners and Adjusto-flex Hinges, a combination that produces uniformly distributed gasket pressure when the door is closed. Door is insulated with 6" or more of cork board or semi-rigid board form insulation, as specified. Cork board laid in hot asphalt; semi-rigid board form encased in water-proof paper. The front and edges of the door, as well as the frame casing, are metal clad. This feature provides protection from the deteriorating effects of moisture precipitation and helps maintain sanitary conditions.

JAMISON-BUILT STEVENSON SUPER-FREEZER DOOR ▶



It has been standard practice for many years to use this door (Over-lap Type) in sub-freezing and sharp freezing operations and rooms in which frozen food products are stored. Jamison "Zero-seal" gasket forms a soft, resilient, moisture impervious, insulating seal between door and frame. Door is insulated with 6" or more of cork board or semi-rigid board form insulation as specified. Cork board laid in hot asphalt, semi-rigid board form encased in water-proof paper.

Door is furnished with frame casing, without jambs. Jamison Adjustable Spring Hinges and Super-Freezer Door Fastener act together to compress the gasket to a uniformly tight seal. Door is protected with 20 gauge galvanized steel on the warm side, front or back, to retard moisture infiltration. Bulletin 126-B.



◀ JAMISON STANDARD DOORS—

Cooler and Freezer

Adaptable over a broad range of uses. Built of finest grade lumber, fully reinforced to form a solid operating unit. Jamison Adjustoflex Hinges and Wedgetight Fasteners are used. Door is shipped hung in frame complete with all hardware ready for installation in your door bucks.

COOLER DOORS—insulated with 4" loose fill insulation encased in water-proof paper.

FREEZER DOORS—insulated with 4" and 6" of cork board or semi-rigid board form insulation, as specified. Cork board laid in hot asphalt, semi-rigid board form encased in water-proof paper. Bulletin 122-E.

Other Jamison-Built Doors to meet your requirements

Sharp Freezer Doors . . . Track Doors . . . Standard Doors . . . Vestibule Doors . . . Dutch Doors . . . Auto-Close Doors . . . Can Passing Vestibules . . . Horizontal and Vertical Sliding Doors . . . Loading Dock Doors . . . Ice Passing Chutes . . . Cold Storage Windows . . . Refrigerator Fronts.

COMPOUNDS, MASTIC (Continued)

Arc Co., 7301 Bessemer Ave., Cleveland 4, O.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Lehon Co., 4425 S. Oakley Ave., Chicago 9, Ill.
Munn and Steele, Inc., 130 Lister Ave., Newark 5, N.J. (Mascalt)
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O.
Pecora Paint Co., Inc., 3rd & Lawrence Sts., Phila. 40, Pa.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17

COMPOUNDS, WATERPROOFING

American Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
A. C. Horn Co., Inc., 43-36-10th St., Long Island City, N.Y.
Inertol Co., Inc., 470 Frelinghuysen Ave., Newark 5, N.J.
Lehon Co., 4425 S. Oakley Ave., Chicago 9, Ill.
North American Fibre Products Co., Standard Bldg., Cleveland 13, O.
Pecora Paint Co., Inc., 3rd & Lawrence Sts., Phila. 40, Pa.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Reilly Tar & Chemical Corp., 1615 Merchants Bank Bldg., Indpls. 4, Ind.
Smooth-On Mfg. Co., 572 Communipaw Ave., Jersey City 4, N.J.
L. Sonneborn Sons, Inc., 88 Lexington Ave., N.Y.C. 16
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
Tamms Silica Co., 228 N. La Salle St., Chicago 1, Ill.
Tuco Products Corp., 30 Church St., N.Y.C. 7
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.

COMPOUNDS, WATER TREATING (See WATER TREATING)

COMPRESSORS, AIR, GAS, etc.

Beach-Russ Co., 50 Church St., N.Y.C. 7
Binks Mfg. Co., 3114 Carroll Ave., Chicago 16, Ill.
Brunner Mfg. Co., Utica 1, N.Y. *(p. 63)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Cooper-Bessemer Corp., Mt. Vernon, O.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
DeVilbiss Co., 300 Phillips Ave., Toledo 1, O.
Fuller Co., Box 420, Catasauqua, Pa. *(p. 172)*
Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Kleen Air, Inc., 719 Raymond Ave., St. Paul 4, Minn.
Lynch Mfg. Corp., 3600 Summit St., Toledo 1, O.
Nash Engrg. Co., 358 Wilson Rd., S. Norwalk, Ct.
Niles Tool Wks. Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Quincy Compressor Co., 217 Maine St., Quincy, Ill.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Union Steam Pump Co., Battle Creek, Mich.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
Yeomans Bros. Co., 1433 N. Dayton St., Chicago 22, Ill.

COMPRESSORS, AMMONIA

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Clark Bros. Co., Inc., Olean, N.Y.
Cooper-Bessemer Corp., Mt. Vernon, O.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Cyclops Iron Wks., 837 Folsom St., San Francisco 7, Cal.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Fuller Co , Box 420, Catasauqua, Pa. *(p. 172)*
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
Yeomans Bros. Co., 1433 N. Dayton St., Chicago 22, Ill.
York Corp., York, Pa. *(p. 163)*

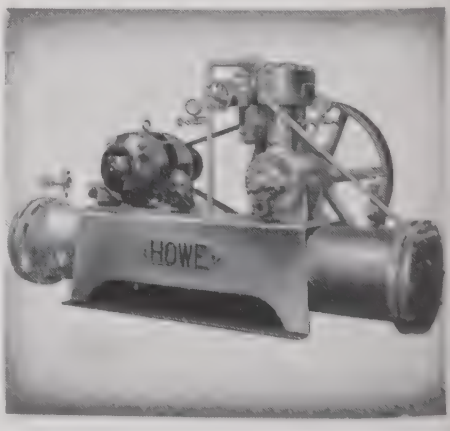
COMPRESSORS, BOOSTER

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 163)*

HOWE REFRIGERATION EQUIPMENT

Efficiently Designed for EVERY Purpose

35 years of specialization have made Howe synonymous with rugged, efficient refrigeration equipment for every purpose. Serving dairies, food processors and quick freezing ice manufacturers, cold storage plants, meat packers, and other industries. Whether you need a new ice machine, condensors, coolers, coils or tanks, a ½ to 150-ton compressor, or a complete plant, Howe engineers know how.



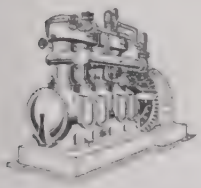
Write your requirements • Distributors in principal cities.

HOWE ICE MACHINE CO.

2821 MONTROSE AVENUE • CHICAGO 18, ILLINOIS

Frick Company

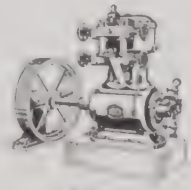
Waynesboro FRICK **Pennsylvania**



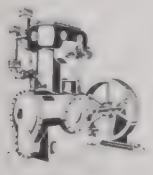
Large 4-Cylinder Compressors, for Ammo. or Freon-12

AMMONIA REFRIGERATION

Capacities from ½ to 1000 tons in one unit. Enclosed compressors feature water jacket over cylinder heads, outboard bearing beyond wheel, force-feed oiling, and patended Flexo-Seals. Capacity controls, dual-pressure cylinders, etc., supplied on order. Booster compressors save power for low temperature Frick-freezing systems. See Bulletins 104, 112, 516, and 651.



2-Cyl. Ammonia Compressors

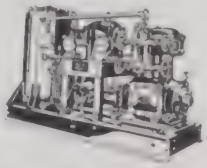


2-Cyl. Booster Compressor

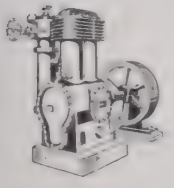
LOW-PRESSURE REFRIGERATION

Units from ¼ through 15 hp. Air and water cooled, for Freon-12. Separate machines in sizes 5¾″×4″ through 17¾″ × 12″, four-cylinder. Produce one ton of refrigeration per hp. expended, on air conditioning work. Equipped with Flexo-Seals, force-feed lubrication, etc. New "Eclipse" machines run up to 900 rpm, give big capacities in small space, have many exclusive features. See Bul. 97, 100, 508, 651.

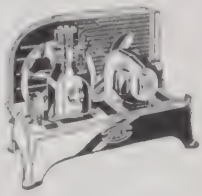
Frick "New Eclipse" Compressors for Freon-12 have 2, 3, 4, or 6 Cylinders, 4⅜″ by 4¼″



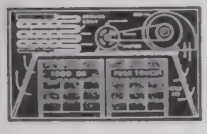
Combined Ammonia Unit



2-Cyl. Freon-12 Compressors



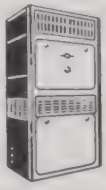
Low Pressure Unit



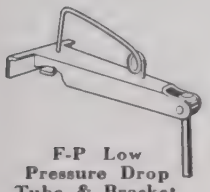
Section thru Frick-Freezing System

AIR CONDITIONING SYSTEMS

Complete systems, as well as unit conditioners and refrigeration for use with equipment built by others. Ask for bulletins 502, 503, 504, and 505, describing air conditioning with ammonia refrigeration and with Freon-12, and illustrating the patented Frick air conditioning systems, among others.



Frick Unit Air Conditioner



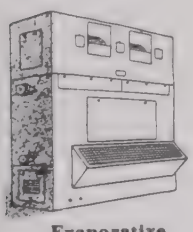
F-P Low Pressure Drop Tube & Bracket For Ice Plants

ICE-MAKING SYSTEMS

Loose can or group lift; brine race with Vertiflow unit or brine cooler systems; simple or dual effect; manual or automatic control. ALL WELDED ice cans, lockers, ice saver dumps, etc.

Bulletins 50, 127, and 145.

Frick-Knickerbocker Lockers



Evaporative Condenser

PLANT EQUIPMENT AND SERVICE

Condensers, coolers, coils, controls, valves and fittings, purgers, oil, etc. Bulletins 140 to 670. Catalog M covers valves and fittings.

Branch Offices and Distributors in principal cities everywhere. Complete service: literature, estimates, layout, manufacture, installation, test, and maintenance.

Bulletins Sent on Request

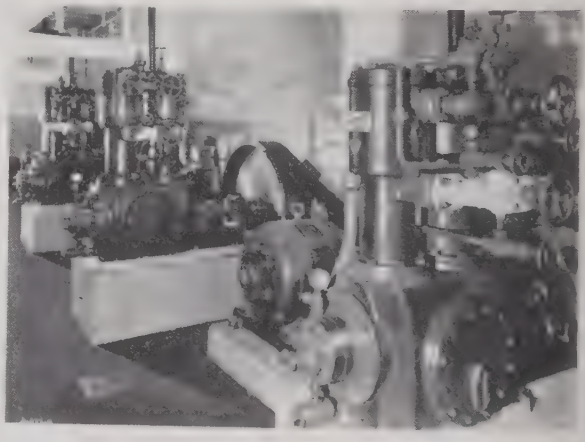
ECONOMICAL—DEPENDABLE

REFRIGERATION

Battery of CP Ammonia Compressors, two 3 cyl. 9 x 9's, two 2 cyl. 9 x 9's and two 2 cyl. 6½ x 6½'s in the Manhattan installation of Dairymen's League Cooperative Assn., Inc., New York City.

A COMPLETE LINE OF COMPRESSORS

Creamery Package makes a type and size compressor to suit every need. You will find CP "Refrigeration Headquarters" specializes in efficient and dependable equipment for the dairy and food industries. And above all you will find it a "Headquarters" with complete refrigeration service from factory to you.



Battery of three CP Ammonia Compressors at Belvidere, Illinois, plant of the Dean Milk Company.

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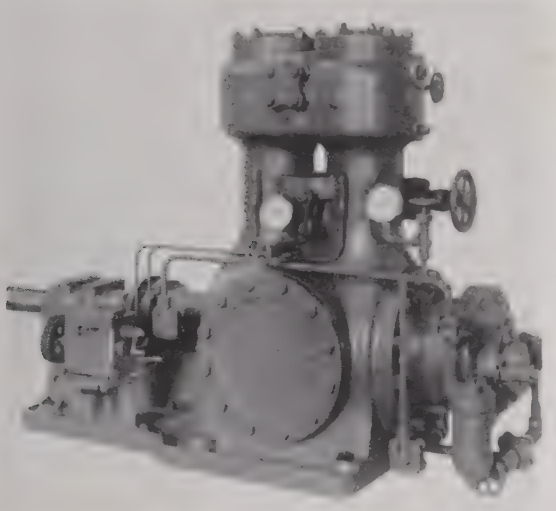
Ice Making and Refrigerating Plants of All Capacities
Air Conditioning Equipment—PakIcers—Quick Freezing Systems



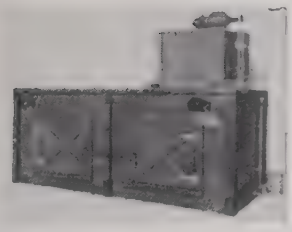
COMPRESSORS

Modern design, engineered ruggedness, assured dependability—all characterize long-lived Vilter Compressors and allied equipment.

AMMONIA Compressors: Vertical units built in sizes from 4 to 100 tons each—horizontal compressors, from 50 tons upward, for electric or steam drive.

BOOSTER Compressors: For economical production of low temperatures. Available in vertical sizes from 60 to 600 cubic feet per minute capacity. Horizontal two-stage units for larger capacities.

FREON Compressors: Vertical units from ¼ h.p. to 150 h.p. Modern design. Rotary type shaft seal. Quiet and efficient trouble-free operation.



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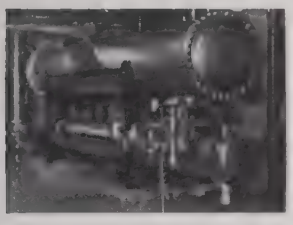
Vertical Crystalform PakIce units in sizes 1 to 6 ton ice per 24 hours. Larger Horizontal units from 10 to 30 tons each—both crystalform and briquette PakIce.



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Condensers and coolers. All sizes from one ton upward. Vertical or Horizontal design. Freon or Ammonia.

—OTHER VILTER PRODUCTS—

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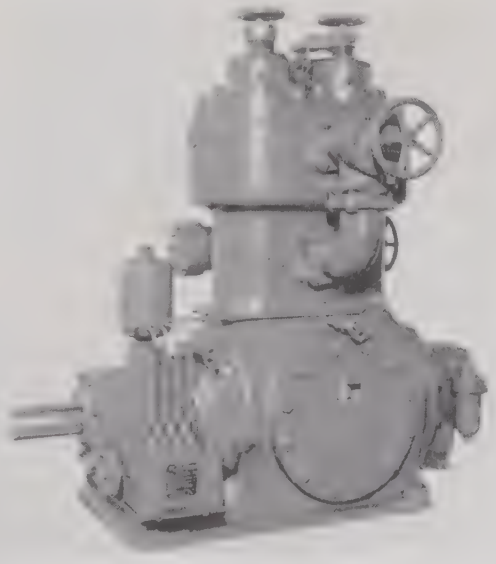
- VALVES AND FITTINGS

CATALOGS AND BULLETINS AVAILABLE ON REQUEST



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The leading periodical in refrigeration and air conditioning. Comprehensive, readable, authoritative, scientifically sound, it presents current developments in refrigeration and air conditioning in an interesting and unbiased style.

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New York 18, New York

COMPRESSORS, CARBON DIOXIDE

Frick Co., Waynesboro, Pa. *(p. 47)*
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 149)*
Wittenmeier Machinery Co., 850 Spaulding N. Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

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Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Trane Co., La Crosse, Wis. *(p. 14)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
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COMPRESSORS, FREON, METHYL, etc. (See also CONDENSING UNITS)

Airtemp Div., Chrysler Corp., 1119 Leo St., Dayton 1, O. *(p. 9)*
American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
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Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 163)*

COMPRESSOR PARTS, DRAWN OR STAMPED (See also STAMPINGS; also DRAWN SHAPES, etc.)

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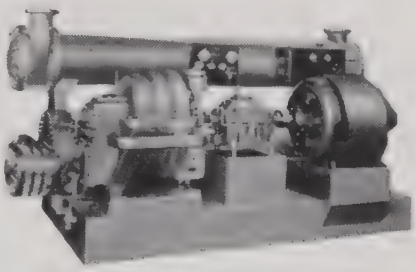
Centrifugal Refrigerating Machines

For large scale air conditioning, low temperature process refrigeration, direct or indirect cooling of water, brines, hydrocarbons and other liquids, and for liquefaction of vapors and gases. The Carrier Centrifugal is a compact, self-contained refrigerating machine, comprising centrifugal multi-stage compressor, evaporator, drive and condenser designed for a single foundation. Any standard motor or turbine drive may be applied. Capacities range from 100 to 1200 tons.

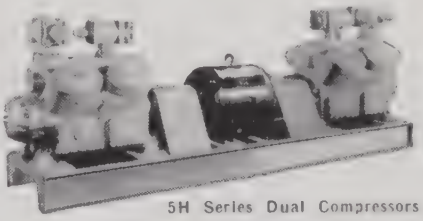


17M Centrifugal Refrigerating Machine

Reciprocating Compressors and Condensing Units

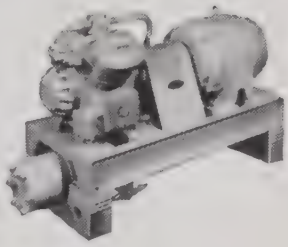
Carrier Customized Refrigerating Units cover the complete capacity range from 5 to 100 horsepower, in uniform load increments between sizes to permit proper unit selection for any load requirement. Many advanced design features such as built-in automatic load leveler for unloaded starting and perfect load balance, make them particularly adaptable to air conditioning and process refrigeration applications.

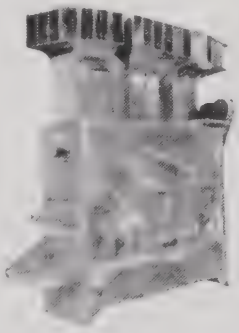
Direct drive dual units are available in 60, 75 and 100 horsepower capacities for use with water or evaporative cooled condensers. Compact condensing units, direct or v-belt drive, water or evaporative cooled arrangements, are available in 5, 7½, 10, 15, 20, 25, 40 and 50 horsepower.



5H Series Dual Compressors

5F and 5H Series Condensing Units





15Q Dry Coil Cold Diffuser

Blast Freezers and Cold Diffusers

For food freezing and storage, meat packing operations and other industries requiring medium and low temperature cooling. Units are available in suspension or floor models, dry coil and spray types in range of capacities up to 36 tons.

Evaporative Condensers

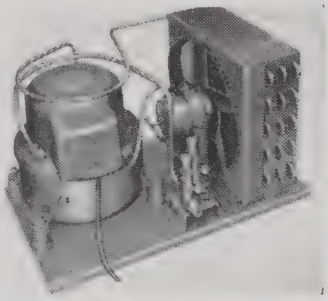
For use with refrigerating compressors in place of water cooled condensers or cooling towers. Compact, sturdy sectionalized construction, suitable for indoor or outdoor installation. Simplify water supply and disposal problems. Available in four standard sizes in capacities of 10 to 75 tons.

9Q Evaporative Condenser

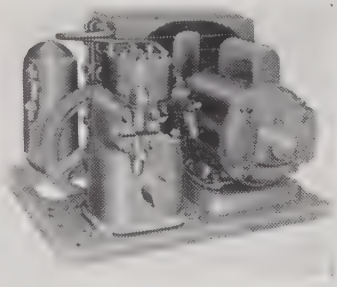
Write for descriptive literature on any of the above equipment

UNIVERSAL COOLER CONDENSING UNITS

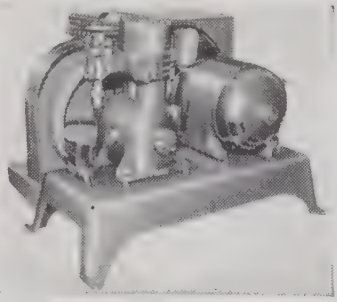
FOR EVERY REFRIGERATION REQUIREMENT



Hermetic Units
Split Phase and Capacitor Types
⅛ H. P. to ½ H. P.



Self-Contained
1/6 H.P. to 1 H.P.



Remote Type
1/4 H.P. to 15 H.P.

Universal Cooler offers an unsurpassed range of types and sizes for every refrigeration application such as

- Frozen Food Cabinets
- Food Storage Refrigerators and Display Cases
- Ice Cream Cabinets and Soda Fountains
- Water and Beverage Coolers
- Air Conditioning
- Commercial Refrigeration Equipment
- Truck Refrigeration Equipment
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- Special Applications

Write us today about your present or anticipated condensing unit requirements. Let us help solve your problems—the Universal Cooler way.

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of Curtis Manufacturing Company

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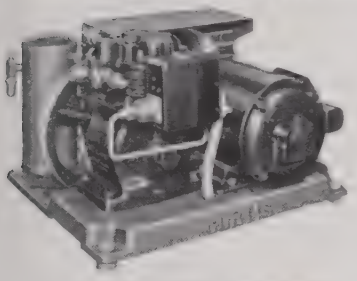
Established 1854

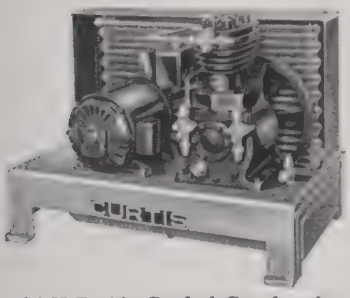
Condensing Units from 1/4 to 30 H.P.

Unit Coolers and Evaporator Coils.

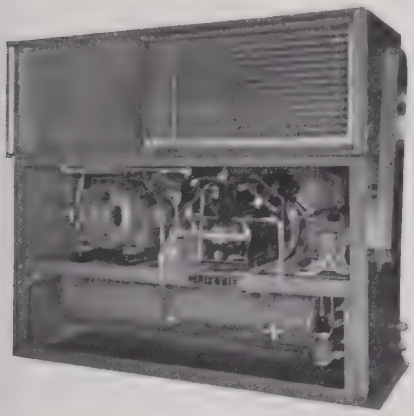
For Refrigeration and Air Conditioning Requirements



¼ to ½ H.P. Self Contained Condensing Unit.



1½ H.P. Air Cooled Condensing Unit. Other sizes from ¼ to 3 H.P.



7½—10—15 ton Remote or Central Type Air Conditioner.

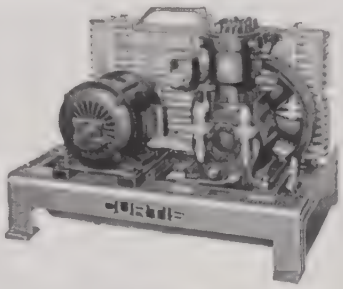
Commercial Refrigeration

Air cooled condensing units from ¼ to 3 H.P., inclusive, and water cooled units from ⅓ to 30 H.P., inclusive. All models available for either Freon (F 12) or Methyl Chloride. Mechanical advantages include Timken Bearings, Centro-Ring Positive Pressure lubrication.

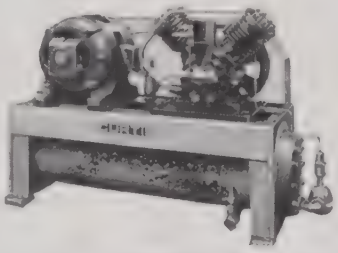
Special models are available for ice cream, frozen food cabinets and for the dairy industry.

Air Conditioning

For today's essential Air Conditioning requirements Curtis offers complete packaged, refrigerated air conditioning units, requiring only water and electrical connections to install. Cools, dehumidifies, circulates and filters the air. Eliminates costly installation expense. Adaptable for heating.



2 H.P. Water Cooled (Counterflow) Condensing Unit. Other sizes from ⅓ to 2 H.P.



15 H.P. Cleanable Shell and Tube Condensing Unit. Other sizes from 3 to 30 H.P.

3 and 5 ton Packaged Type Air Conditioner

CURTIS REFRIGERATION
AIR CONDITIONING COMMERCIAL

R560

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Jos. A. Martocello & Co., 229 N. 14th St., Phila. 7, Pa. *(p. 157)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

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Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 159)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill. *(p. 55)*
Flow Controls, Inc., 1821 W. North Ave., Chicago 22, Ill.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
York Corp., York, Pa. *(p. 163)*

CONDENSERS (See particular type following)

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American Coils Co., 25 Lexington St., Newark 5, N.J.
Brown Fintube Co., Elyria, O.
Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Fedders-Quigan Corp., 57 Tonawanda St., Buffalo 7, N.Y.
G & O Mfg. Co., 138 Winchester Ave., New Haven 8, Ct.
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Hussmann Refrigeration, Inc., 2401 N. Leffingwell St. Louis 6, Mo. *(p. 87)*
Kirsch Co., Sturgis, Mich. *(p. 54)*
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
McQuay Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
Oakes North Chicago Div., Houdaille-Hershey Corp., 1900 Foss Park Ave., N. Chicago, Ill. *(p. 54)*
Paley Mfg. Corp., 244 Herkimer St., Brooklyn 16, N.Y.
Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.

CONDENSERS

Economical methods of manufacture, modern production lines, engineering service and outstanding developments make this organization an excellent source of supply for production quantities of refrigerator condensers and evaporators of the household type.

HOUDAILLE-HERSHEY CORP.
North Chicago Division, North Chicago, Illinois

Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
Trane Co., La Crosse, Wis. *(p. 14)*
Universal Cooler Div., International Detrola Corp., 299 Joseph St., Marion, O. *(p. 52)*
Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

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Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown Pa,.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
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York Corp., York, Pa. *(p. 163)*
Young Radiator Co., Racine, Wis. *(p. 16)*

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Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

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Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Harry Cooling Towers, Inc., West St., Doylestown, Pa. *(p. 80)*
Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
Standard Refrigeration Co., 232 S. Hoyne Ave., Chicago 20, Ill.
York Corp., York, Pa. *(p. 163)*

AIR COOLED CONDENSERS

Kirsch Condensers are ruggedly built for trouble-free, permanent service. Hydrogen brazing of continuous tubing and specially designed fins guarantees durable bond, assures unrestricted flow of cooling agent through leak-proof tubes. Available in static, cross-over and manifold types . . . in single and multi-bank designs with 1/4", 5/16" and 3/8" tubing. Write for information on the application of Kirsch condensers to your needs.

KIRSCH COMPANY
Refrigerator Condenser Division
Sturgis, Michigan

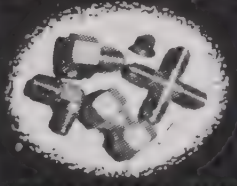
FOR THE BEST
SPECIFY ELECTRIMATIC

QUICK COUPLERS
...no tools needed

REFILLABLE AND
NON-REFILLABLE DRIERS

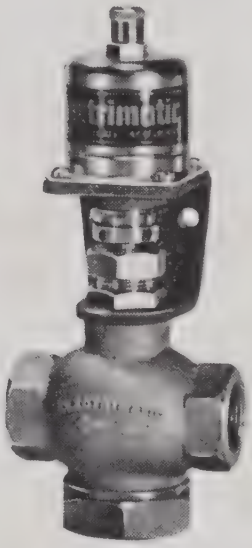
ALL-BRASS STRAINERS

FLEXIBLE REFRIGERATION
HOSE CHARGING LINES



FORGED FLARE NUTS
AND FITTINGS

Electrimatic
TYPE WP
WATER REGULATING VALVES



For Freon • Methyl Chloride • Sulphur Dioxide

No other valves do a more efficient job of controlling the flow of cooling water to the condensers of refrigerating machines. Electrimatic Type WP Valves not only start and stop the water, but also feed the proper amount of cooling water to secure the correct condensing pressure without wasting water.

Electrimatic Type WP may be installed at either the inlet or outlet water connection of the condensing unit, and may be serviced without breaking water or pressure connections. The actuating connection can be made to either the compressor, condenser or receiver. Every valve precision engineered, individually inspected and tested. Sizes ⅜ in., ½ in. and ¾ in. FPT.

Now Available for Immediate Shipment

Also available—Electrimatic's full line of other sizes and types of automatic valves. Ask us for full particulars.

Electrimatic

2100 INDIANA AVE., CHICAGO 16, ILL. CANADA—2025 ADDINGTON AVE., MONTREAL

CONDENSERS, DOUBLE PIPE

California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

CONDENSERS, DOUBLE TUBE

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Halstead & Mitchell, Bessemer Bldg., Pittsburgh 22, Pa. (Cleanable) *(p. 57)*
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Reese & Long Refrigeration Products, Inc., 408 E. 25th St., N.Y.C. 10
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.
Standard Refrigeration Co., 232 S. Hoyne Ave., Chicago 20, Ill.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 43)*
York Corp., York, Pa. *(p. 163)*

CONDENSERS, ELECTRIC (See CAPACITORS)

CONDENSERS, EVAPORATIVE (See EVAPORATIVE CONDENSERS)

CONDENSERS, SHELL & COIL

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Kramer Trenton Co., Olden & Bruening Aves., Trenton 5, N.J. *(p. 227)*
Long Mfg. Div., Borg-Warner Corp., 12501 Dequindre St., Detroit 12, Mich.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
Standard Refrigeration Co., 232 S. Hoyne Ave., Chicago 20, Ill.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Trane Co., La Crosse, Wis. *(p. 14)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15 N.Y.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 163)*

CONDENSERS, SHELL & TUBE

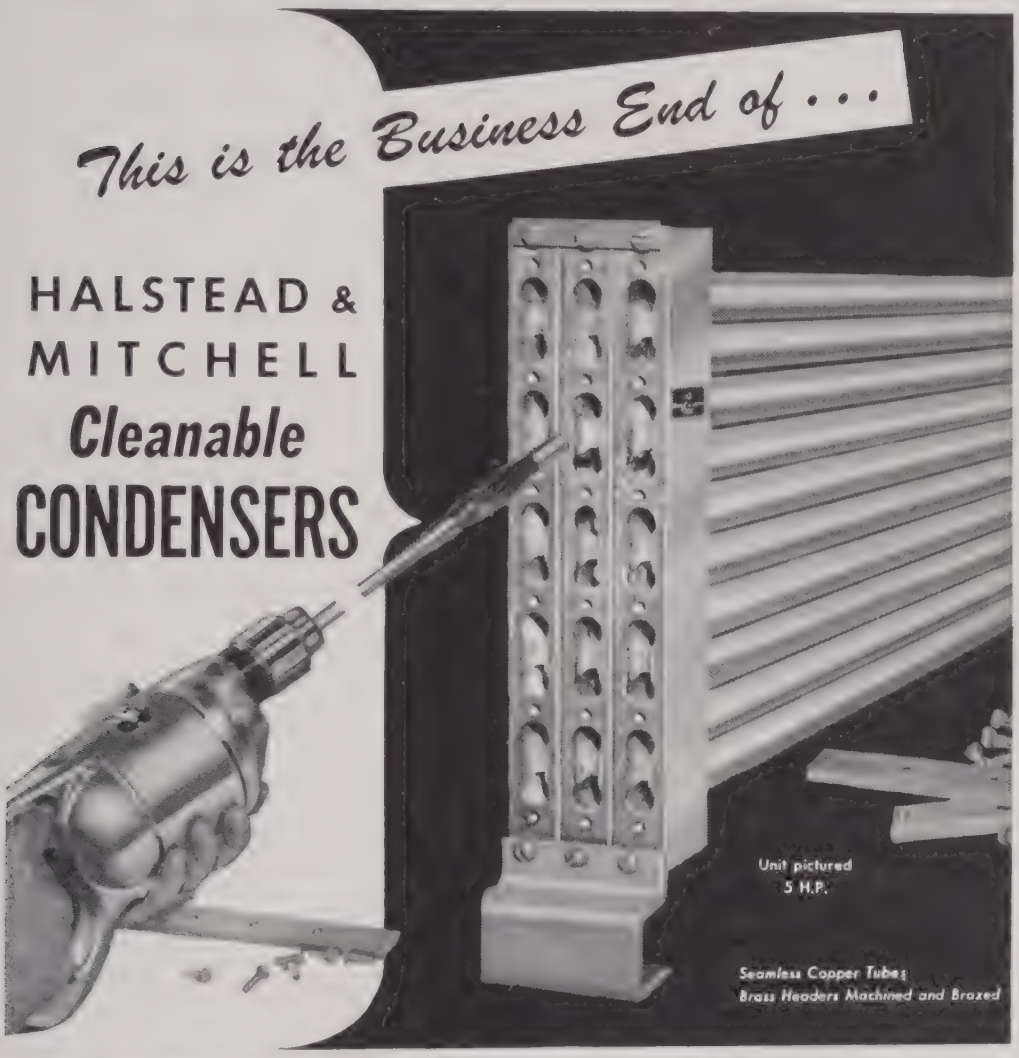
Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Chattanooga Boiler & Tank Co., P.O. Box 110, Chattanooga, Tenn.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlen Ave., St. Louis 20, Mo. *(p. 53)*
Davis Engrg. Corp., 30 Rockefeller Plaza, N.Y.C. 20
J. P. Devine Mfg. Co., 920 Shawnee Ave., Mt. Vernon, Ill.
Downingtown Iron Wks., Downingtown, Pa. *(p. 59)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
W. J. Finnegan Co., 7402 Santa Monica Blvd., Los Angeles 46, Cal. *(p. 116)*
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.H. *(p. 65)*
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Lummus Co., 420 Lexington Ave., N.Y.C.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Quaker City Iron Wks., Aramingo Ave. & E. Tioga St. Phila., Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave. Chicago 18, Ill.
Reynolds Mfg. Co., Inc., Springfield, Mo.

(Continued)

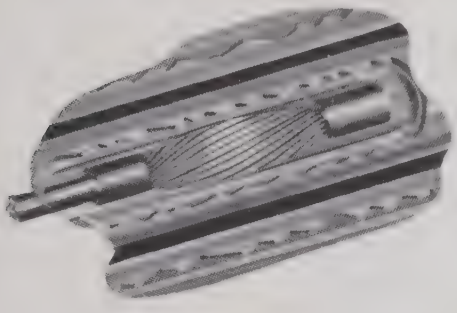
DOUBLE-TUBE COUNTER-FLOW WATER-COOLED CONDENSERS

This is the Business End of . . .

HALSTEAD & MITCHELL *Cleanable* CONDENSERS



Service Engineers and commercial users throughout the refrigeration industry are now specifying HM Condensers for replacement and conversion orders. These new HM units combine two features never before obtainable in tube-within-a-tube water-cooled condensers; (1) They're CLEANABLE . . . the water tubes are easily accessible at both ends for the spiral cleaning tool to restore the interior water surfaces to "new-unit" efficiency. (2) A TRUE-COUNTER-FLOW relationship is achieved between the coolant and the refrigerant through a unique seamless copper tube-within-a-tube construction that makes obsolete most types of similar water-cooled condensers. Thus, water and space requirements are reduced substantially and a most economical operation is obtained.



HM Condensers available from ½ to 10 H. P. from wholesalers in principal cities.

HM Halstead & Mitchell

OFFICES: Bessemer Building, Pittsburgh 22, Pa.

The Patterson-Kelley Company, Inc.

108 Burson Street **East Stroudsburg, Pa.**

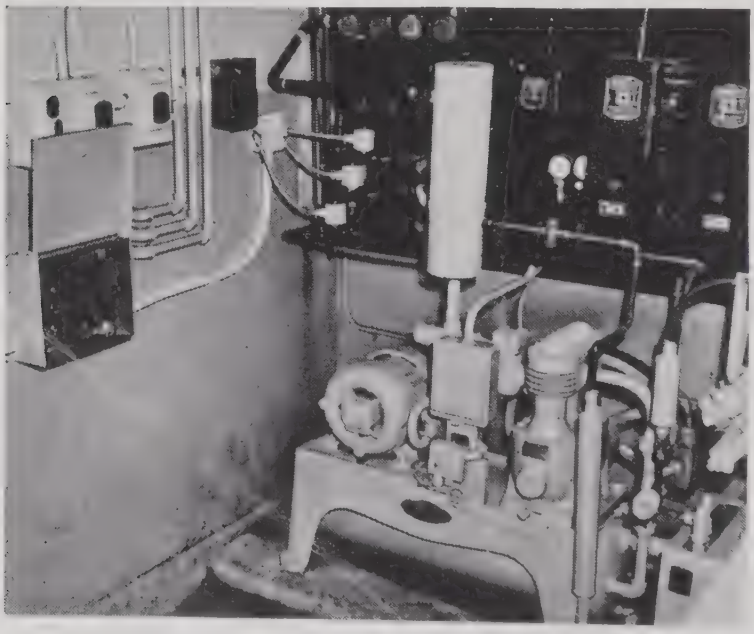
New York Office, 101 Park Avenue, Zone 17
Offices or Representatives in Other Principal Cities

Patterson Products

Freon Coolers
Water, Brine and other Coolers
Condensers
Slug Eliminators
Balance Loaders
Defrosting Systems

All of the above units are in the class of "heat transfer" equipment and it is in this class that The Patterson-Kelley Company has been specializing since 1880. Its products are carefully engineered and constructed of properly selected materials according to the most modern manufacturing methods. Literature describing any product will be sent on request.

If you have *any* requirement involving *any* type of heat transfer equipment, let us know. Our engineers will be glad to study your problem and recommend the proper type and size of unit.



Patterson Defrosting Unit installed in a 3-room refrigerating system, the temperatures of the rooms being 0°, 10° and 20° respectively. The Patterson Slug Eliminator, the vertical cylinder over the compressor, functions here as it does in all refrigeration systems: to prevent slugs of oil or refrigerant from returning to the compressor.

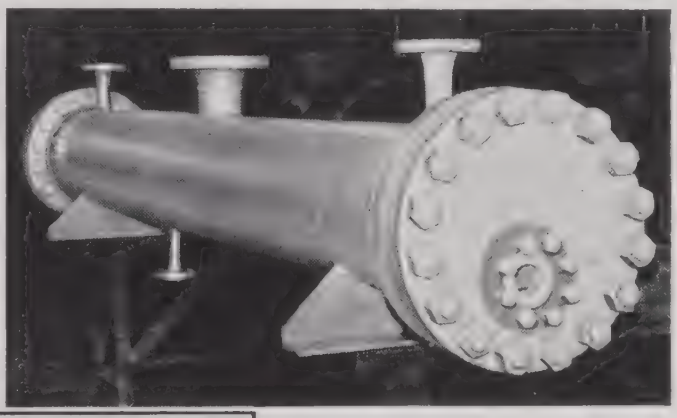
Downingtown Iron Works, Inc.

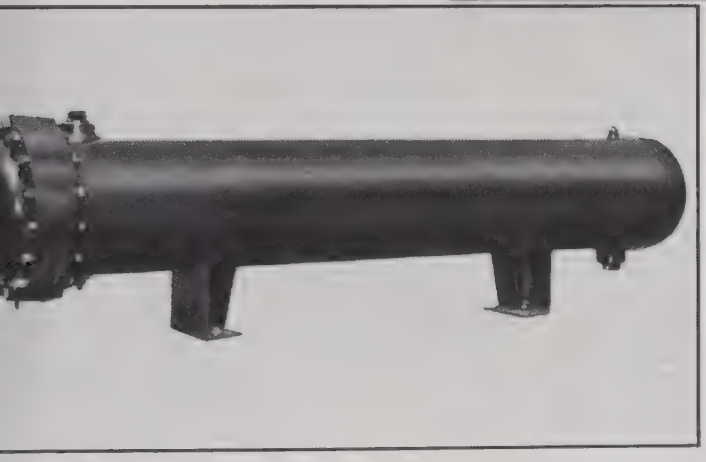
Heat Transfer Specialists

Downingtown, Pa.

DESIGNERS and BUILDERS of SHELL and TUBE CONDENSERS and COOLERS, Including:

- Freon and Ammonia Condensers
- Dry Expansion-Water and Brine Coolers
- Flooded-Water and Brine Coolers
- Chemical Process Liquid Coolers
- Oil Refinery Liquid Coolers





(Above) Mixed Gas Cooler—cools gases in tubes at 450 lb. pressure with ammonia boiling in shell. All steel.

(Left) Dry Expansion Chiller for Circulaing Cutting Oil System 12¾" od x 5'. "U" Tube Type.

DOWNINGTOWN . . . also DESIGNS AND BUILDS:

Many items outside the scope of the ordinary manufacturer such as Suction Line Heat Exchangers for Freon Systems of larger tonnage and Special Intercoolers and Coolers for oil, chemicals and liquids not ordinarily encountered in every day refrigerating practice.

SPECIAL COOLERS and CONDENSERS— BRONZE or STAINLESS STEEL

Downingtown specializes in coolers and condensers for applications requiring out of the ordinary experience in design and metallurgy.

Some examples: 1—Sea water condensers using everdur, aluminum brass and copper nickel.

2—Coolers for penicillin broth,—stainless steel.

3—Coolers for lube oil, gasoline or other refinery products.

4—Special liquid coolers, as cooling methyl alcohol with freon; cooling amyl acetate with ammonia.

Other features of our service cheerfully explained. Write us.

Downingtown is Approved by Hartford for A.S.M.E. U-69 Welding

ACME INDUSTRIES, INC.

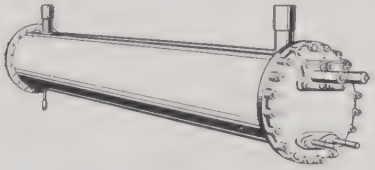
Jackson, Michigan

Representatives in Principal Cities

Continuously Serving The Refrigeration Industry Since 1919

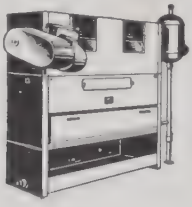
FREON CONDENSERS
AMMONIA CONDENSERS

Shell and Tube type for use with Ammonia, Freon or other Refrigerants. Standard or special designs to meet varying water temperatures available and condensing temperatures desired.



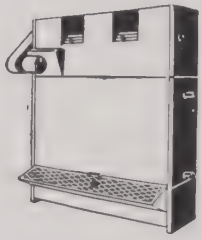
DRY-EX COOLERS

Refrigerant in Tubes, solution baffled through shell. For cooling water, brine, glycols or alcohols by direct expansion of refrigerant.



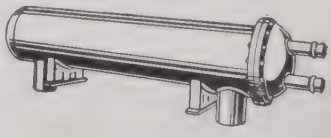
BLO-COLD INDUSTRIAL UNIT COOLERS

Blo-Cold Models are available for either medium temperature or low-temperature applications.



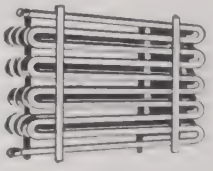
EVAPORATIVE CONDENSERS

All prime surface for Freon or Ammonia Refrigerants—Heavy gage sheet metal casings, especially processed for maximum resistance to rust and corrosion. Capacities to 100 tons.



HEAT INTERCHANGERS

Shell and Coil units for small capacities, shell and tube units for large installations. 16 standard models from one ton to 180 tons capacity.



PIPE COILS

Fabricated in all shapes and sizes from ½ in. IPS to 2 in. IPS., in accordance with customer's specifications.

Acme Industries, Inc. also manufacture Flooded Water and Brine Coolers, Oil Separators, Receivers and Fin Coils

WRITE FOR CATALOG ON ANY PRODUCT

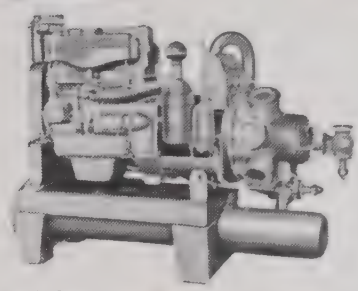
CONDENSERS, SHELL & TUBE (Continued)

Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 19, Va.
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Pufalo 13, N.Y.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.
Standard Heater & Oi]Equip Co., 245 Cornelison Ave., Jersey City 2, N.J.
Standard Refrigeration Co., 232 S. Hoyne Ave., Chicago 20, Ill.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Trane Co., La Crosse, Wis. *(p. 14)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7 Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walsh Holyoke Boiler Wks., Div. of Continental-United Industries Co., Inc., 110 Appleton St., Holyoke, Mass.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 163)*

CONDENSING UNITS, AIR COOLED, FREON & METHYL

Airtemp Div., Chrysler Corp., 1119 Leo St., Dayton 1, O. *(p. 9)*
American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Brunner Mfg. Co., Utica 1, N.Y. *(p. 63)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Copeland Refrigeration Corp., Sidney, O. *(p. 64)*
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chidago 7, Ill. *(p. 48)*
Crider Corp., 2001 Harding Highway, E., Lima, O. *(p. 68)*
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlen Ave., St. Louis 20, Mo. *(p. 53)*
Diceler Div., General Machine & Mfg. Co., Blair St. & Spring Garden Ave., Berwick, Pa.
Frick Co., Waynesboro, Pa. *(p. 47)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
International Harvester Co., 180 N. Michigan Ave., Chicago 1, Ill.
Jack & Heintz Precision Industries, Inc., Cleveland 1, O.
Lehigh Mfg. Co., Div. of Lehigh Foundries, Inc., 143 Fountain Ave., Lancaster, Pa. *(p. 61)*
Lynch Mfg. Corp., 3600 Summit St., Toledo 1, O.
Mayflower Products, Inc., 13 S. 5th St., Richmond, Ind.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. 62)*
Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Paley Mfg. Corp., 244 Herkimer St., Brooklyn 16, N.Y.
Reliance Refirgerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Servel, Inc., Evansville 20, Ind. *(p. 69)*
Tecumseh Products Co., Tecumseh, Mich. *(p. 70)*
Universal Cooler Div., International Detrola Corp., 299 Joseph St., Marion, O. *(p. 52)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 163)*

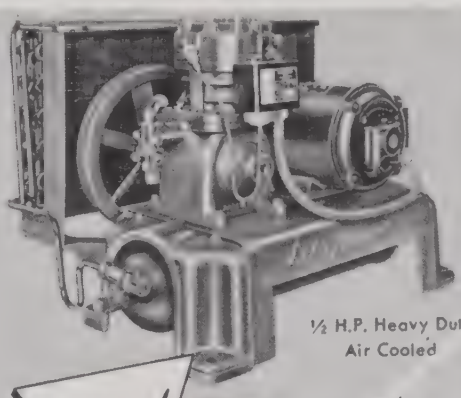
READY-POWER ENGINED REFRIGERATION
for Lowest Cost Air Conditioning and Refrigeration



Powered by International Engines Using Gasoline Natural Gas Butane or Diesel Fuel

For dependable air conditioning or refrigeration anywhere at lowest cost! Ideal for stationary, portable or marine operation.

THE READY-POWER CO.
11231 FREUD AVE., DETROIT 14, MICHIGAN.



½ H.P. Heavy Duty Air Cooled

Lehigh BLU-COLD COMMERCIAL REFRIGERATION

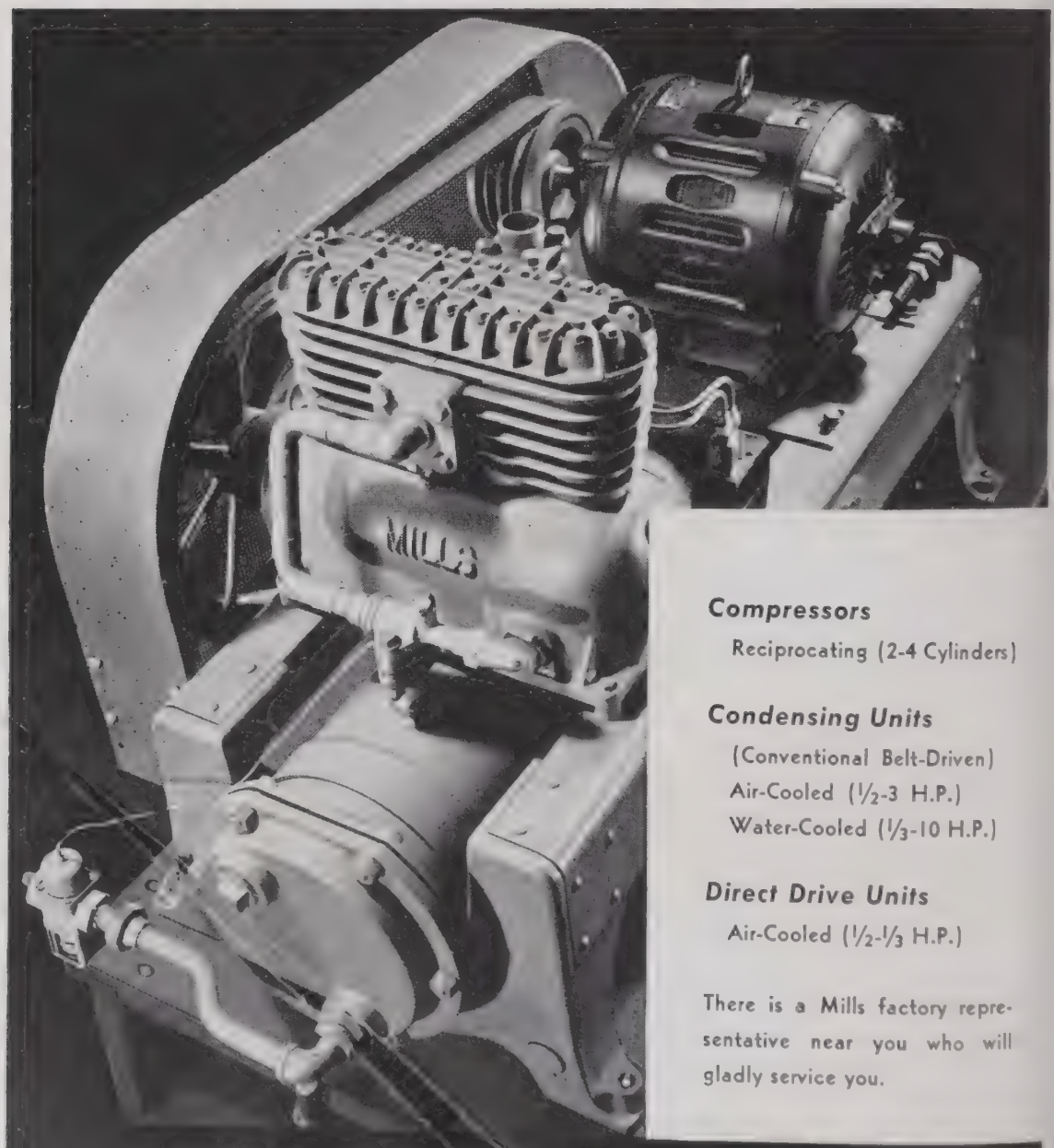
¼ H.P. to 5 H.P.
HIGH-MEDIUM-LOW TEMPERATURE
F. 12 or M.C.

Package AIR COOLED ★ Heavy Duty and Standard Duty AIR COOLED. ★ Heavy Duty WATER COOLED. ★ Combination AIR & WATER COOLED. General catalog on request.

Lehigh Mfg. Co. LANCASTER, PA.

MILLS CONDENSING UNITS

for air-conditioning...
for commercial and industrial refrigeration...



Compressors

Reciprocating (2-4 Cylinders)

Condensing Units

(Conventional Belt-Driven)
Air-Cooled (½-3 H.P.)
Water-Cooled (⅓-10 H.P.)

Direct Drive Units

Air-Cooled (½-⅓ H.P.)

There is a Mills factory representative near you who will gladly service you.

MILLS INDUSTRIES, INCORPORATED

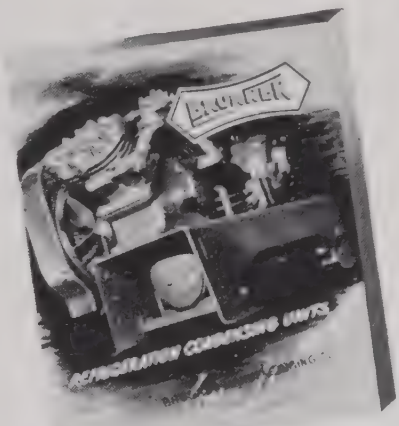
REFRIGERATION DIVISION: 4100 FULLERTON AVENUE, CHICAGO 39, ILL.

BRUNNER
SINCE 1906

Buy B.T.U.'s When You Select A Condensing Unit

Horse Power means nothing in performance. What you want is slow speed for long life and B.T.U.'s for performance.

When you select a Brunner Condensing Unit to do a job, you will find by comparison that you get lower cost of operation, longer life and trouble free performance.



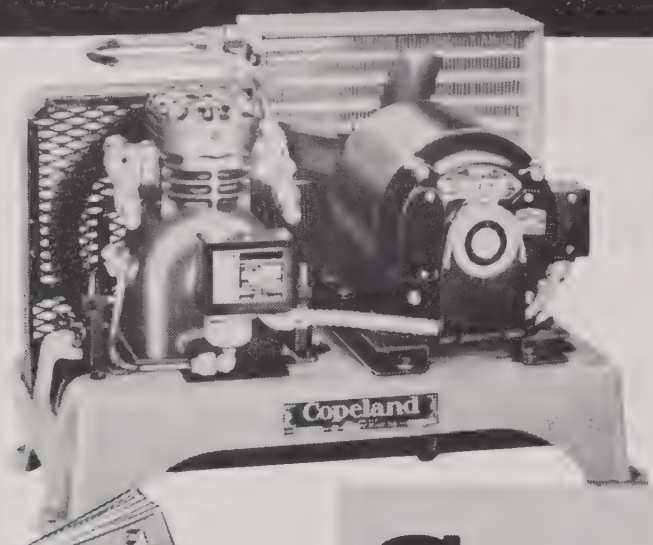
Brunner Condensing Units are manufactured in sizes 1/4 H.P. to 30 H.P. A separate plant and production line is maintained for the larger sizes . . . 5 H.P. to 30 H.P. . . . which insures prompt delivery.

Complete catalog on Request

BRUNNER MANUFACTURING CO.

UTICA, NEW YORK, U.S.A.

DEPENDABLE ELECTRIC REFRIGERATION SINCE 1918



For any refrigeration problem there is a Copeland unit in the correct size and design

Write for complete catalog giving entire specifications on each unit.

Copeland

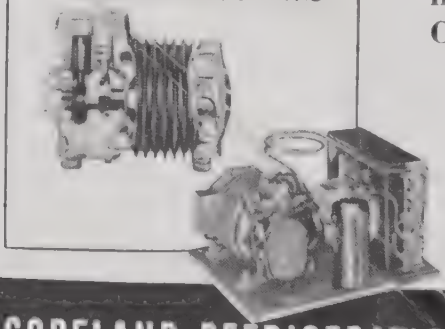
DEPENDABLE *Electric* REFRIGERATION

famous for long life, quietness, efficiency

Copeland Refrigerators and Water Coolers in standard sizes.

COPELAND units are sturdy, compact, Air-cooled remote models from 1/4 h.p. to 2 h.p. inclusive. Water-cooled remote models from 1/3 h.p. to 7½ h.p. Compact, belt-driven units for self-contained installations in sizes from 1/6 to 1/2 h.p. For today's best in design and construction achievements, COUNT ON COPELAND.

COPELAMETIC
the accessible hermetic



COPELAMETIC, the ACCESSIBLE hermetic, combines all the advantages of open-type units and welded-in hermetics. Eliminates belts, seals and manual oiling. Models from 1/20 H.P. to full 3/4 H.P.

COPELAND REFRIGERATION CORPORATION · SIDNEY · OHIO

General Electric Company

AIR CONDITIONING DEPARTMENT

5 LAWRENCE ST., BLOOMFIELD, NEW JERSEY

GENERAL ELECTRIC

DISTRICT AND LOCAL OFFICES

ATLANTA 3, GEORGIA
410 Red Rock Building

BOSTON 15, MASS.
700 Commonwealth Avenue

CHICAGO 54, ILLINOIS
The Merchandise Mart
Room 1127

CINCINNATI 2, OHIO
617 Vine Street, Room 1328

CLEVELAND 14, OHIO
925 Euclid Avenue
Room 524

DALLAS 2, TEXAS
1801 North Lamar Street

DETROIT 2, ILLINOIS
700 Antoinette Street

KANSAS CITY 6, MO.
Suite 2510
106 W. 14th Street

LOS ANGELES 54, CALIFORNIA
212 N. Vignes Street

MINNEAPOLIS 2, MINNESOTA
12 South 6th Street

NEW ORLEANS 12, LA.
Richard Bldg., Room 1004
837 Gravier Street

NEW YORK 22, NEW YORK
570 Lexington Avenue

PHILADELPHIA 22
PENNSYLVANIA
1405 Locust Street

PITTSBURGH 22
PENNSYLVANIA
535 Smithfield Street

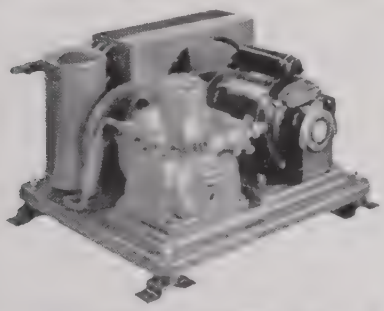
PORTLAND 7, OREGON
P.O. Box 909

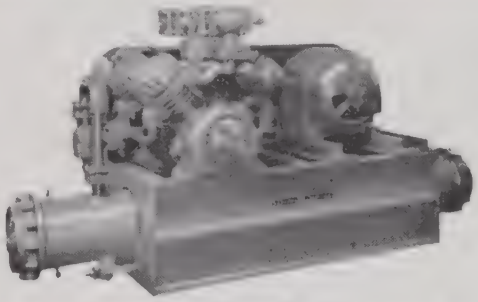
ST. LOUIS, MO.
Room 485, Arcade Building

SALT LAKE CITY 9, UTAH
200 South Main Street

SAN FRANCISCO 6
CALIFORNIA
235 Montgomery Street

WASHINGTON 5, D.C.
806 15th Street, N.W.





CONDENSING UNITS, 1/6 hp to 125 hp, reciprocating type, for "Freon" refrigerants. Air cooled models from 1/6 hp to 3 hp, water cooled models from 1/2 hp to 125 hp, motor compressor units for use with evaporative condensers or cooling towers from 3 hp to 125 hp. Hermetically sealed units also available to manufacturers for inclusion in refrigerated fixtures.

"CONDITIONED AIR" UNIT COOLERS, in a wide variety of styles and sizes, for preservation of quality, appearance, weight, and freshness of perishable foods.

PACKAGED REFRIGERATION EQUIPMENT

WATER COOLERS in bottle and foot-pedal-operated pressure bubbler models, available with open or hermetically-sealed condensing units. REACH-IN STORAGE REFRIGERATORS, five models including ice makers and forced air circulation types, in sizes ranging from 16 cu. ft. to 33 cu. ft. SECTIONAL WALK-IN ROOMS for standard and low temperature refrigeration. Assembly can be made in a variety of arrangements to fit individual requirements. FROZEN FOOD MERCHANDISERS, in sizes from 10 cu. ft. to 26 cu. ft. DRY BEVERAGE COOLERS holding from fifteen to thirty-one cases of 12-oz. bottles, including remote and self-contanied types. FOOD FREEZERS FOR FARM AND HOME, for quantity food freezing and frozen storage of foods, in 18 cu. ft. and 26 cu. ft. MILK COOLERS holding from three 10-gallon cans to eight 10-gallon cans. For rapid cooling an accessory circulator is available for all models. For extremely rapid cooling of milk, a cold shower accessory is also available.

PACKAGED AIR CONDITIONERS

PACKAGED AIR CONDITIONERS for direct or remote air conditioning for stores, offices, and office suites.

REMOTE AIR CONDITIONING EQUIPMENT

CENTRAL PLANT AIR CONDITIONERS, vertical and horizontal units for summer, winter, or year 'round air conditioning systems. FINNED HEAT TRANSFER SURFACE, cased to reduce field engineering requirements, for cooling with chilled water or direct expansion of "Freon," or heating with steam or hot water. REMOTE ROOM AIR CONDITIONERS for individual room control of air conditions in multi-room buildings.

GENERAL ELECTRIC *Air Conditioning · Commercial Refrigeration*

Worthington Pump and Machinery Corp.

Air Conditioning and Refrigeration Division

General Offices: Harrison, N.J.

District Offices and Representatives in Principal Cities

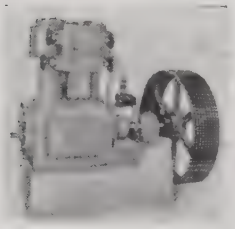
WORTHINGTON

A8-99

Refrigeration Systems for Air Conditioning Industrial Process and All Other Applications

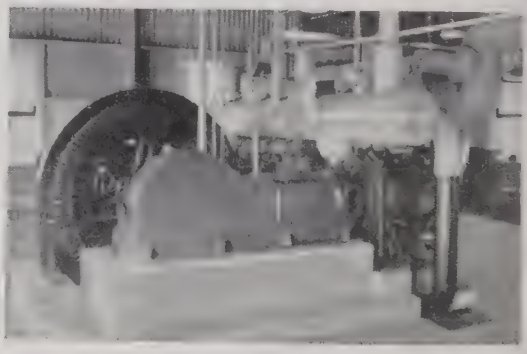
Complete refrigerating systems for use with Freon-11, Freon-12, Methyl Chloride, Ammonia or Carbon Dioxide, either direct expansion or water cooling applications. A complete line of refrigeration compressors, permitting impartial recommendations. A nation-wide organization of distributors in major cities to provide sales and engineering service and to plan complete air conditioning systems of the central or unit type. Architects, Engineers and contractors are invited to consult with us. Write to Harrison, N.J., or to anyone of our district offices for information on these products.

VERTICAL AMMONIA COMPRESSORS



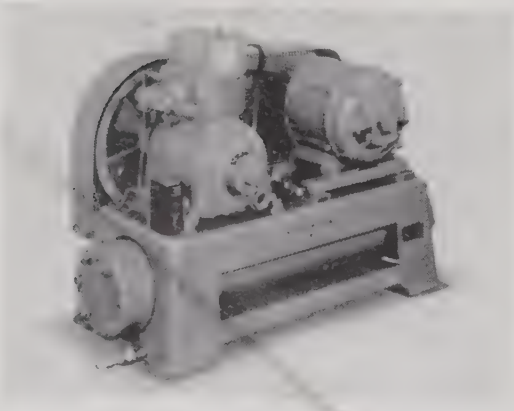
Pressure-lubricated, roller main bearings; safety heads; patented Feather Valves; belt drive or direct connected to electric motor, diesel or gas engine; sizes 3 x 3 to 10 x 10, 2-cylinder.

HORIZONTAL AMMONIA COMPRESSORS



Single and duplex; single-stage and two-stage; belt drive or direct connected to electric motor, diesel or steam engine; patented Feather valves; ratings from 60 to 1000. Automatic capacity control features are easily applied.

SELF-CONTAINED FREON-12 CONDENSING UNITS



Capacities from ¼ to 100 tons. Condensers are highly efficient multi-pass shell and straight tube type. Condenser heads readily removable, making tubes easily accessible for cleaning. Small sizes have condensers mounted in base; larger units have condenser mounted above compressor and motor. Minimum space required.

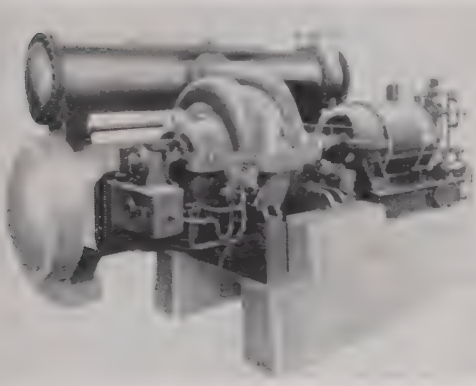
ICE PLANTS

Ice plant equipment of all types, including complete refrigeration machines with Worthington gas engine, diesel engine or steam drive in one unit. Also, motor or belt driven compressors. Ice tank with loose can or group lift. High transfer coils in brine race or brine coolers as best suited.

Ammonia Absorption Refrigeration Machinery
All Types of Auxiliary Equipment, Condensers, Coolers, Fittings, Etc.

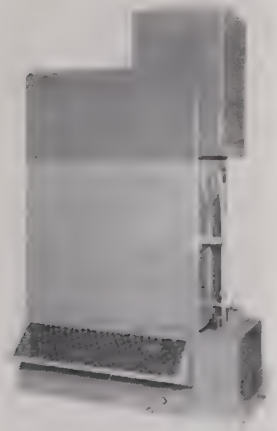
Worthington Pump and Machinery Corp.

CENTRIFUGAL REFRIGERATION WATER COOLING SYSTEMS



Freon-11 centrifugal compressor, water cooler and water-cooled condenser in compact unit assembly. Electric motor or steam turbine drive. 56 unit sizes . . . 150 to 1200 tons.

EVAPORATIVE CONDENSERS



Series ECZ Freon-12 illustrated. 10 to 50 tons refrigeration. Units of sectionalized construction; all parts easily accessible. Galvanized steel coils for ammonia. Bare copper coils for Freon-12.

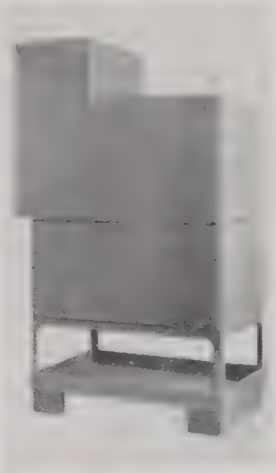
CONDENSERS AND BRINE COOLERS

Multi-pass, as illustrated for closed systems and space saving. Heads efficiently baffled to produce high liquid velocity. Vertical "Spira-Flo" types provided with circular water box and special water distributors. Highly efficient, easily accessible for cleaning without interfering with condenser operation.

BOOSTER REFRIGERATION COMPRESSORS

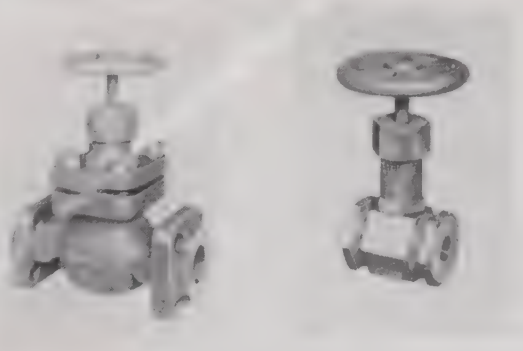
Multi-cylinder single-acting for capacities up to 400 cfm. Horizontal double-acting for larger capacity requirements. Designed to produce and maintain low evaporator pressures required for low temperature applications in conjunction with compression or absorption refrigeration systems.

PRODUCT COOLERS



Dry type units in many sizes and arrangements. Capacities from 2 to 10 tons. Air circulation rates from 2000 to 12000 cfm. Sectionalized construction with removable panels. Finned or bare pipe coils for use with Freon or Ammonia.

AMMONIA VALVES AND FITTINGS



Smaller sizes of all steel construction. Larger size valve bodies, ells and tees, close grained iron.

CONDENSING UNITS, AMMONIA

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 151)*
Creamery Package Mfg. Co. 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Cyclops Iron Wks., 837 Folsom St., San Francisco 7, Cal.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
Frick Co., Waynesboro, Pa. *(p. 47)*
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 163)*

CONDENSING UNITS, CENTRIFUGAL

Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 151)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

CONDENSING UNITS, GASOLINE DRIVEN

Brunner Mfg. Co., Utica 1, N.Y. *(p. 63)*
Ready-Power Co., 11231 Freud Ave., Detroit 14, Mich. *(p. 61)*
Servel, Inc., Evansville 20, Ind. *(p. 69)*
Sterling Mfg. Co., 2523 Farnam St., Omaha, Neb.
Universal Cooler Div., International Detrola Corp., 299 Joseph St., Marion, O. *(p. 52)*

CONDENSING UNITS, HERMETIC OR SEMI-HERMETIC, FREON & METHYL

Airtemp Div., Chrysler Corp., 1119 Leo St., Dayton 1, O. *(p. 9)*
American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
Copeland Refrigeration Corp., Sidney, O. *(p. 64)*
Crider Corp., 2001 Harding Highway, E., Lima, O. *(p. 68)*
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. 62)*
Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
Schaefer, Inc., 801 Washington Ave., N., Minneapolis 1, Minn.
Servel, Inc., Evansville 20, Ind. *(p. 69)*
Tecumseh Products Co., Tecumseh, Mich. *(p. 70)*
Universal Cooler Div., International Detrola Corp., 299 Joseph St., Marion, O. *(p. 52)*
York Corp., York, Pa. *(p. 163)*

CONDENSING UNITS, HOUSEHOLD

Copeland Refrigeration Corp., Sidney, O. *(p 64)*
Hotpoint, Inc., 5600 W. Taylor St., Chicago 44, Ill.
Servel, Inc., Evansville, Ind. *(p. 69)*
Tecumseh Products Co., Tecumseh, Mich. *(p. 70)*

THE CRIDER CORPORATION
LIMA OHIO

Refrigerating Systems

Universal Cooler Div., International Detrola Corp., 299 Joseph St., Marion, O. *(p. 52)*

CONDENSING UNITS, PROPANE

Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Propane Development Corp., 41 Murray St., N.Y.C. 7
Ready-Power Co., 11231 Freud Ave., Detroit 14, Mich. *(p. 61)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

CONDENSING UNITS, WATER COOLED, FREON & METHYL

Airtemp Div., Chrysler Corp., 1119 Leo St., Dayton 1, O. *(p. 9)*
American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Brunner Mfg. Co., Utica 1, N.Y. *(p. 63)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Copeland Refrigeration Corp., Sidney, O. *(p. 64)*
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Curtis Refrigerating Machine Div., Curtis Mfg. Co., 1949 Kienlen Ave., St. Louis 20, Mo. *(p. 53)*
Diceler Div., General Machine & Mfg. Co., Blair St., & Spring Garden Ave., Berwick, Pa.
Frick Co., Waynesboro, Pa. *(p. 47)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 46)*
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
International Harvester Co., 180 N. Michigan Ave., Chicago 1, Ill.
Lehigh Mfg. Co., Div. of Lehigh Foundries, Inc., 143 Fountain Ave., Lancaster, Pa. *(p. 61)*
Lummus Co., 420 Lexington Ave., N.Y.C.
Lynch Mfg. Corp., 3600 Summit St., Toledo 1, O.
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
Servel, Inc., Evansville 20, Ind. *(p. 69)*
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Ca.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Trane Co., La Crosse, Wis. *(p. 14)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
Universal Cooler Div., International Detrola Corp., 299 Joseph St., Marion, O. *(p. 52)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 163)*

CONDUIT, ELECTRICAL

Chicago Metal Hose Corp., Maywood, Ill.
General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
National Elec. Products Corp., Fulton Bldg., Pittsburgh 22, Pa. (Flexible)
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.
Youngstown Sheet & Tube Co., Youngstown, O.

CONDUIT SYSTEMS, INSULATED

Ric-Wil Co., Union Commerce Bldg., Cleveland 14, O.

CONNECTING RODS

American Crucible Products Co., Lorain, O.
Atlas Drop Forge Co., 209 W. Mt. Hope Ave., Lansing 2, Mich.
Saginaw Malleable Iron Div., Gen'l. Motors Corp., Saginaw, Mich.

Servel

ELECTRIC REFRIGERATION

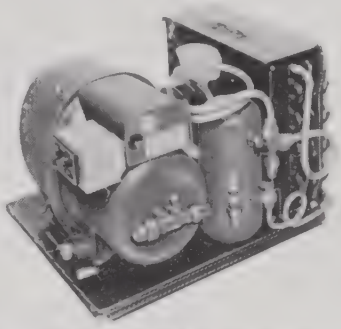
The Complete Line—

- Servel Supermetic Units
- Servel Belt-driven Units
- Servel Compressors

Servel's new line of commercial condensing units offers a wide range of medium and low temperature models from small fractional horsepower sizes through a series of larger units to 3 HP.

CHOICE OF 28 SEALED MODELS

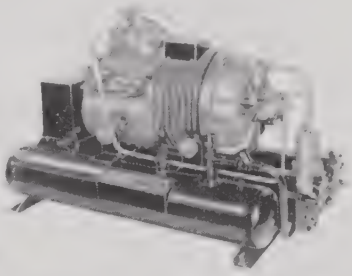
Servel Supermetic sealed type units are available in 28 different models. Fractional horsepower sizes include 10 compactly designed hermetically sealed models for both low and medium temperature applications. For larger installations (1 HP through 3 HP) four- and six-cylinder field serviceable Supermetics are available in air and water-cooled designs for low and medium temperature work. All units are fully connected, ready to run. Forced lubrication assures long and dependable refrigeration service.



Hermetically sealed units for self-contained work.

SERVEL BELT-DRIVEN UNITS

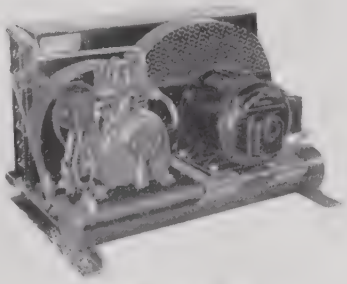
For applications where conventional air-cooled, belt-driven units are required, Servel offers 14 models for both low and medium temperature work in sizes from ¼ HP through 2 HP. These compactly-built units incorporate many new and improved design features, including forced lubrication to every vital part.



Compact, field serviceable Supermetics, 1 HP, 1½ HP, 2 HP and 3 HP.

COMPRESSORS—POWER UNITS

Complete condensing units are offered through franchised distributors and dealers, and to "Allied" fixture manufacturers. For special applications, manufacturers may also purchase "power units" (motor-compressor assemblies) or belt-driven type compressors, depending upon their particular requirements.

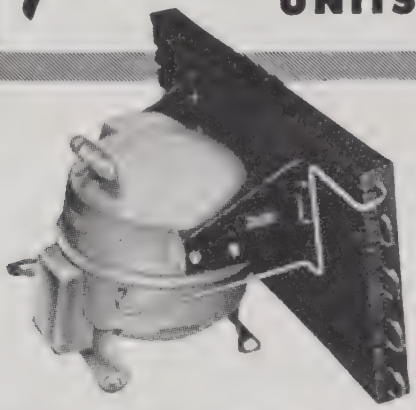


Belt-driven units, ¼ HP through 2 HP.

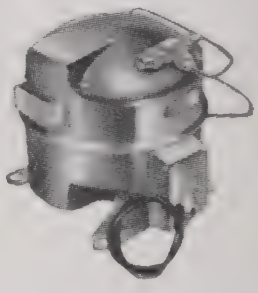
Complete Data Will Be Furnished On Request

Servel, Inc., Electric Refrigeration Div., Evansville 20, Indiana

Tecumseh HERMETIC
UNITS *and* COMPRESSORS



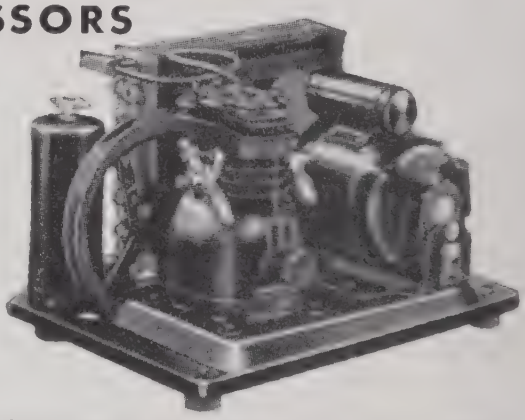
● Tecumseh Hermetic Units are available in seven basic fan-cooled and static condenser models in all sizes through ⅓ H.P. rating. Line production on single purpose machine tools plus triple inspection and careful selective fitting of all parts assure a high uniformity of quality. Illustrated at left: ⅛ H.P. Domestic Unit (Model S88LE) . . . noiseless in operation, particularly adaptable to pull-out construction, extensively used in the manufacture of domestic refrigerators.



● Tecumseh Hermetic Compressors are unsurpassed in the refrigeration industry. Twin cylinder models in ¼ and ⅓ H.P. sizes are available for commercial applications, single cylinder internal mounted models (illustrated) in 1/8, 1/6 and 1/5 H.P. sizes meet the requirements of domestic and light commercial uses. Spring mounted to reduce noise to a minimum.

Chieftain CONVENTIONAL
UNITS *and* COMPRESSORS

● Chieftain Conventional type refrigerating units have set high standards for the industry since 1935 . . . leaders in performance, quietness, low first cost, and freedom from field service. All parts are machined to close limits, triple inspected and selectively fitted to assure high precision, trouble-free operation. Illustrated at right: ¼ H.P. air-cooled Chieftain Condensing Units for competitively priced commercial equipment.



● Dependable Chieftain Compressors form the basis for the outstanding performance found in Chieftain Condensing Units. Available in single, two and four cylinder models, they have pioneered many of the major advances in compressor design and manufacture . . . advances such as precision machining methods, forced feed lubrication for reciprocating parts, automatic cylinder lubrication, Chieftainizing connecting rods to prevent scoring, etc. plus many improvements in preparation, testing, inspection and assembling. Illustrated at left: the Chieftain "4". Write today for complete information and descriptive literature.

TECUMSEH PRODUCTS CO.

TECUMSEH MICHIGAN

CONNECTIONS, FLEXIBLE (See FLEXIBLE CONNECTIONS)

CONNECTIONS, TUBE, SPECIAL

Brockway Co., 361 Church St., Naugatuck, Ct.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.

CONNECTORS, WIRE

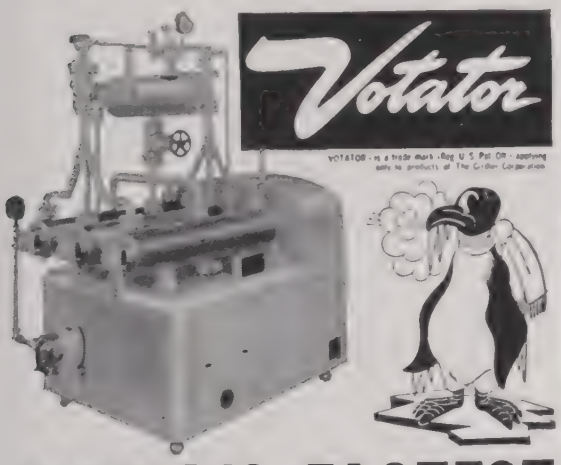
Burke Elec. Co., Erie, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Ideal Industries, Inc., Sycamore, Ill.

CONTACTS, ELECTRICAL

Fansteel Metallurgical Corp., N. Chicago, Ill.
General Plate Div., Metals & Controls Corp., Attleboro, Mass.
Gibson Elec. Co., 8350 Frankstown Ave., Pittsburgh 21, Pa.
Handy & Harman, 82 Fulton St., N.Y.C. 7
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls, Ind.
National Carbon Co., Inc., Unit. of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.

CONTINUOUS FREEZERS

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Burge Ice Machine Co., 218 N. Jefferson St., Chicago, Ill.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.



WORLD'S FASTEST FREEZE for juices, purees, pulps, liquid eggs, etc. New, continuous, closed method for slush freezing and processing simultaneously, in *seconds*—cuts tunnel freezer time and refrigeration costs. Write: *The Girdler Corporation, VOTATOR DIVISION, Louisville 1, Ky.* District Offices: 150 Broadway, New York City 7; 617 Johnston Bldg., Charlotte 2, N. C.; 2612 Russ Bldg., San Francisco 4.

A GIRDLER PRODUCT

Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
Refrigeration Corp. of America, Div. of Noma Elec. Corp., 55 W. 13th St., N.Y.C. 11
Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.
Votator Div., Girdler Corp., Louisville 1, Ky. *(p. 71)*
York Corp., York, Pa. *(p. 165)*

CONTROLS (See particular type following)

CONTROLS, AIR CONDITIONING SYSTEM

Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Barber-Colman Co., Rockford, Ill.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Sampsel Time Control, Inc., 600 N. Strong Ave., Spring Valley, Ill.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Warren Webster & Co., Camden, N.J.
Young Regulator Co., 5209 Euclid Ave., Cleveland 3, O

CONTROLS, DEFROSTING CYCLE

Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Flow Controls, Inc., 1821 W. North Ave., Chicago 22, Ill.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Miller Harris Instrument Co., 1434 W. Atkinson Ave., Milwaukee 6, Wis.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Paragon Elec. Co., 1600-12th St., Two Rivers, Wis.
Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
Perfex Corp., 500 W. Oklahoma Ave., Milwaukee 7, Wis.
Ranco Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 75)*
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon, N.Y.
White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.

CONTROLS, HIGH PRESSURE OR HIGH PRESSURE CUTOUT

(A—Ammonia; B—Other refrigerants)

(B) Allen-Bradley Co., Milwaukee 4, Wis.
(A,B) American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
(A,B) Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
(A,B) Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
(B) Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
(A,B) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*

(A,B) Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hays Corp., Michigan City, Ind.
(A,B) Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
(B) Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
(A,B) Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
(B) Perfex Corp., 500 W. Oklahoma Ave., Milwaukee 7, Wis.
(B) Ranco Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 75)*
(A,B) Refrigerating Specialities Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
(A,B) U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
(B) White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.

CONTROLS, HUMIDITY (See also HUMIDISTATS)

Bahnson Co., 1001 S. Marshall St., Winston-Salem 7, N.C.
Barber-Colman Co., Rockford, Ill.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
Standard Engrg. Wks., 289 Roosevelt Ave., Pawtucket, R.I.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.

CONTROLS, LIQUID LEVEL (See also FLOATS; also FLOAT SWITCHES; also HIGHSIDE FLOATS; also LOWSIDE FLOATS, etc.)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Automatic Control Co., 1005 University Ave., St. Paul 4, Minn.
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Hancock Valve Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Hays Corp., Michigan City, Ind.
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
Liquidometer Corp., 41-03-36th St., Long Island City, 1 N.Y.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Moore Products Co., H & Lycoming Sts., Phila. 24, Pa.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Photoswitch, Inc., 77 Broadway, Cambridge 42, Mass.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*

CONTROLS, MOTOR (See STARTERS)

CONTROLS, PROGRAM, SELECTIVE, SEQUENCE, etc.

Arrow-Hart & Hegman Elec. Co., 103 Hawthorn St., Hartford 6, Ct.
Automatic Control Co., 1005 University Ave., St. Paul 4, Minn.
Barber-Colman Co., Rockford, Ill.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Thomas A. Edison, Inc., W. Orange, N.J.
Leeds & Northrup Co., 4970 Stenton Ave., Phila. 44, Pa.
Miller Harris Instrument Co., 1434 W. Atkinson Ave., Milwaukee 6, Wis.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
National Time & Signal Corp., 21800 Wyoming Ave., Detroit 20, Mich.
Perfex Corp., 500 W. Oklahoma Ave., Milwaukee 7, Wis.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.J.
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon, N.Y.
Wheelco Instruments Co., Harrison & Peoria Sts., Chicago 7, Ill.

CONTROLS, SPECIAL

Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Cutler-Hammer, Inc., 215 N. 12th St., Milwaukee 1, Wis.
Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
Thomas A. Edison, Inc., W. Orange, N.J.
Fischer-Smith, Inc., 164 State St., West Englewood, N.J.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Miller Harris Instrument Co., 1434 W. Atkinson Ave. Milwaukee 6, Wis.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
Perfex Corp., 500 W. Oklahoma Ave., Milwaukee 7, Wis.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12. Ill.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
United Elec. Controls Co., 71 A St., S. Boston 27, Mass.
White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo

CONTROLS, SUCTION PRESSURE ACTUATED
(A—Ammonia; B—Other refrigerants)
(a—Close on rise; b—With H.P. cutout; c—Reverse acting; d—With overload protection; e—Multiple circuit)

(A:a,b,c) Allen Bradley Co., Milwaukee 4, Wis.
(A,B:a,b,c) American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
(A:a,c; B:a,b,c,d) Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
(A:a; B:a,b,c) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
(B:a,b,d) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(A,B:a,b,c,e) Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
(B:a,b,c,d,e) Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*

(Continued)

Penn Electric Switch Co.

Goshen, Indiana

Controls for Refrigeration, Air Conditioning, Heating, Engines, Pumps and Air Compressors

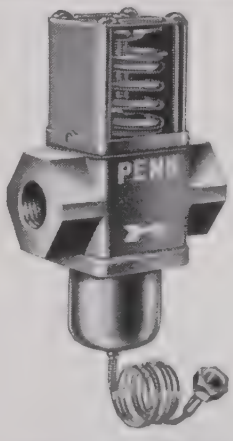
a Two-Pole Control For Refrigeration

Series 270 temperature or low side pressure controls. Dual controls with built-in high pressure cut-out also available.

The new Penn Series 270 refrigeration controls for pressure and temperature applications incorporate a load-carrying, 2-pole single throw contact structure . . . a feature which gives you two switches in one! Result? A wide variety of multiple applications formerly requiring two single pole conventional type controls. The Penn 270 provides the following typical application advantages:

1. Controls polyphase motors (where protection against single phasing is provided) *without the use of line starters.*
2. When installed with 1-Phase AC or DC motors, control is wired as a 2-pole switch to *break both lines.*
3. Incorporates two separate circuits necessary for the control of single phase multiple systems or polyphase systems when one pole is used as a pilot control. This one control can *start simultaneously the condensing unit and open magnetic valve.*
4. Controls *two separate load circuits* such as two single phase motors . . . controls "high-low" speed circulating fans in walk-in coolers, display cases, etc. . . . controls any other two function combinations within electrical specifications.

All Series 270 incorporate a visible calibrated scale which indicates *directly* both cut-in and cut-out settings. Its many design features assure freedom from contact trouble, better electrical performance and dependable, efficient operation. *Write for Bulletin 2652.*



Series 246 Water Regulating Valves

Penn Series 246 water regulators modulate the flow of water through condensers or compressors to meet cooling requirements. New revolutionary design eliminates sticking of seats . . . water hammer . . . drain plugs . . . rusting of range springs . . . need for lubrication . . . corrosion of and sedimentation on sliding parts. Available in threaded and flanged type models for commercial or marine service . . . from ⅜" to 2½" sizes.

Other Penn Refrigeration Controls

In addition to pressure and temperature controls and water regulating valves, Penn's line of refrigeration controls include cooling thermostats, humidistats, solenoid valves and low and line voltage relays.

FOR COMPLETE INFORMATION, WRITE FOR FREE CATALOG

The Mercoid Corporation

4201 BELMONT AVENUE, CHICAGO 41, ILLINOIS

MERCOID CONTROL FEATURES

Mercoid hermetically sealed mercury switches used exclusively. They are immune to dust, dirt or corrosion and insure better control performance and longer control life. The outside double adjustment and calibrated dial on all DA Pressure and Temperature Controls simplify setting the operating range—no guesswork.

No. 855 Line Voltage Thermostat

This instrument is recommended for such applications as cooler rooms, florists' cabinets or butcher cases, etc. It is equipped with a heavy duty capacity mercury contact switch that will operate directly on 10 amp. 115 volts or 5 amp. 230 volts. Various ranges available. This thermostat is dependable under severe operating conditions over a long period of years.

Type DA-61 Low and High Pressure Control

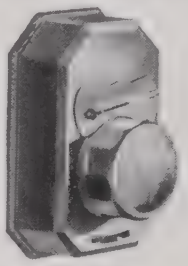
For close regulation of ammonia refrigeration equipment, from changes in suction pressure. By means of this control the regulation of individual box temperatures on a multiple system is simplified. Ranges available: 10 inches vac. to 75 lbs., or 0-300 lbs. Can be furnished with different circuit arrangements or with flanged case back or bottom connection.

Type DA-51 Low and High Pressure Control

This control provides very accurate and reliable regulation of refrigeration equipment using Freon, Methyl Chloride, Sulphur Dioxide or any other refrigerants not injurious to bronze. Available 25 inches vac. to 50 lbs., or 0-200 lbs. Can be furnished with plain case either bottom or back connection. See catalog No. 600 for further details.

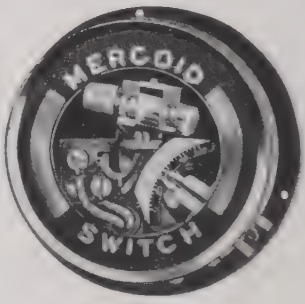
Type DA-55 Remote Stem Temperature Control

For refrigeration and air conditioning applications. Also suitable for the control of brine, water, or low air temperatures in freezing rooms. Available ranges: minus 30 to plus 60 degrees; minus 60 to plus 30 degrees; 0 to 75 degrees and 25 to 100 degrees. Can be furnished with special circuit arrangements and different types of air bulbs.



No. 855

Type DA-61



Type DA-51

Type DA-55

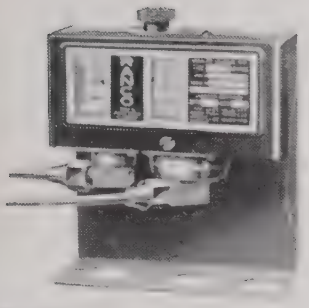
MERCOID

Industry's First Choice for Accuracy and Dependableness in Temperature or Pressure Controls

RANCO INC.
Columbus 1, Ohio, U.S.A.

Ranco Refrigeration Controls, designed and manufactured by refrigeration specialists, provide dependable, trouble-free service on the most exacting applications. More than 12,000,000 Ranco Controls have been supplied to the refrigeration industry, which uses more Ranco Controls than any other make. Check with Ranco Inc. first on your refrigeration control requirements.

Type "O" Refrigeration Controls
THE LEADING CONTROLS FOR FOOD PRESERVATION



O-1505

O-1505—DUAL PRESSURE COMMERCIAL CONTROL. High pressure cut-out independently adjustable. Single pole, snap acting switch closes circuit on increase and opens circuit on decrease in pressure.

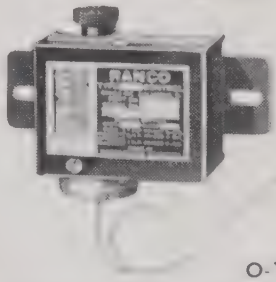
O-1401—LOW PRESSURE COMMERCIAL CONTROL. Graduated visible scale with calibrations for range and differential setting. Range screw changes cut-out and cut-in together. Differential screw changes cut-out only.

O-1401

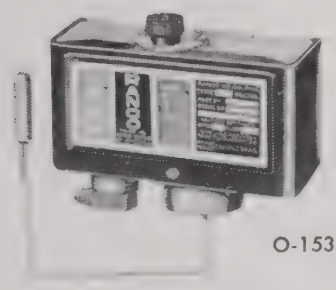
O-1402

O-1402—LOW PRESSURE COMMERCIAL CONTROL. Similar to O-1401, but has constant cut-in. Like all Ranco type O Controls, is compact, sturdily constructed and adaptable to individual installation requirements.

O-1419—TEMPERATURE COMMERCIAL CONTROL. Outside range and differential (cut-out) adjustments with graduated visible scales and calibrations. Lowest cut-out: F.; highest cut-in: 15° F.



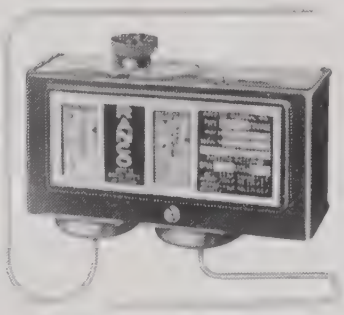
O-1419



O-1535

TYPE 91-O—EXCLUSIVE INTERLOCKING TWO-TEMPERATURE CONTROL. Assures uniform temperature, uniform high relative humidity and completely automatic defrosting of coil regardless of weather or load conditions or cold location of the compressor.

O-1535—DUAL TEMPERATURE COMMERCIAL CONTROL. Range screw changes temperature cut-out and cut-in together. High pressure cut-out independently adjustable. Differential screw changes temperature cut-out only.



91-O

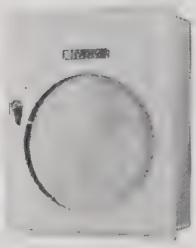
Ranco Inc. RANCO CONTROLS

Available Through Leading Wholesalers

WRITE FOR INFORMATION

See Honeywell FOR AUTOMATIC CONTROL THAT SERVES YOU BEST

Unit Air Conditioner Control Panel



Recording Locker Room Thermometer

Minneapolis-Honeywell, with more than 60 years of experience, is best prepared to fill your control needs. Manufacturers of recorders, indicators, controllers, and special control panels for commercial and industrial refrigeration.

MD 4747 Panel

For control of unit air conditioners up to 5 ton capacity. Incorporates dual pressure controller for high and low side of compressor, with temperature controller for return air. Temperature setting, reset switch, and 3-way selector switch for cooling or fan operation, accessible from outside of cabinet. Unit also contains starter relay for compressor motor up to 5 H.P. capacity. Special parts are available to meet the design requirements of any unit conditions.

Recording Locker Room Thermometer

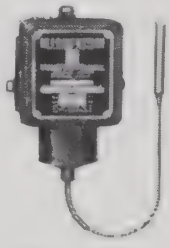
Model No. 645101; Chart No. 12777, Range —30°F. to +70°F., reverse. 12″ diameter, seven-day revolution with one-hour time divisions. Has hand-wound chart drive as standard installation. Electric clock drive optional—self-starting synchronous electric clock, 115/230 v., 20-50-60 cycles. Equipped with 5′ of bronze armored tubing and stainless steel bulb with 10″ extension neck.

Temperature Controllers

L480—Remote Bulb Temperature Controller. Standard features include tamperproof shield for external adjustment dial; cold control; adjustable differential; extra terminal for reverse action; accurately calibrated dial.

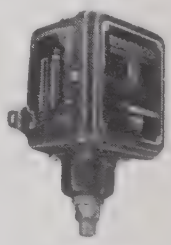
Air Switch

T491A—Heavy Duty wall mounted thermostat. Mercury Switch contacts are actuated by a bi-metal element which is designed to withstand extreme conditions of temperature and humidity.



Temperature Controllers

Heavy duty remote bulb temperature controllers in a variety of ranges, with external scale plate marked high-low and calibrated at mid-scale. T414A for 2-wire switching; T614A for single pole-double throw switching for control of two speed motors, valves, alarm circuits, etc.



Pressure Controllers

L414A—Heavy Duty Suction Pressure Controller for use with non-corrosive refrigerants only. Direct reading range and differential scales.

Write for complete information on these and our many additional Refrigeration Controllers and accessories

Minneapolis-Honeywell Regulator Company

Branch offices in all principal cities

FACTORIES: . . . Minneapolis, Minn., . . . Chicago, Ill., . . . Wabash, Ind., . . . Philadelphia, Pa.

Executive offices: 2818—Fourth Avenue, South . . . Minneapolis, Minn.

CONTROLS, SUCTION PRESSURE ACTUATED (Continued)

(A,B:a,b,c,d,e) Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
(B:a,c) Perfex Corp., 500 W. Oklahoma Ave., Milwaukee 7, Wis.
(B:a,b,c,d) Ranco Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 75)*
(A,B:a,d) Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
(A,B:a,b) U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
(B:a,b,c,e) White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.

CONTROLS, TEMPERATURE ACTUATED (See also THERMOSTATS)

Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Barber-Colman Co., Rockford, Ill.
Barnes & Jones, Inc., 128 Brookside Ave., Jamaica Plain 30, Mass.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
D'Este Div., American Chain & Cable Co., Inc., Reading, Pa.
Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Partlow Corp., 2 Campion Rd., New Hartford, N.Y.
Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
Perfex Corp., 500 W. Oklahoma Ave., Milwaukee 7, Wis.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Ranco Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 75)*
Red Spot Elec. Co., Tacoma, Wash.
Sampsel Time Control, Inc., 600 N. Strong Ave., Spring Valley, Ill.
Spencer Thermostat Co., Unit of Metals & Controls Corp. 34 Forest St., Attleboro, Mass.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon, N.Y.
United Elec. Controls Co., 71 A St., S. Boston 27, Mass.
White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.

CONTROLLERS, DIESEL ENGINE

Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*

CONTROLLERS, FLOW (See also REGULATORS' FLOW)

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill.
Fischer & Porter Co., Hatboro, Pa.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Hays Corp., Michigan City, Ind.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Refrigerating Specialities Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Trane Co., La Crosse, Wis. *(p. 14)*

CONTROLLERS, GAS PRESSURE (See also REGULATORS, PRESSURE)

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
Hays Corp., Michigan City, Ind.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
C. J. Tagliabue Mfg. Co., 500 Park Ave., Brooklyn 5, N.Y.

CONTROLLERS, STARTING LOAD

Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 159)*

CONTROLLERS, TEMPERATURE (See also REGULATORS TEMPERATURE)

American Schaeffer & Budenberg Instrument Div., Manning Maxwell & Moore, Inc., Bridgeport 2, Ct.
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Burling Instrument Co., 253 Springfield Ave., Newark 3, N.J.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
Leeds & Northrup Co., 4970 Stenton Ave., Phila. 44, Pa.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Moeller Instrument Co., Inc., 132nd St., & 89th Ave., Richmond Hill 18, N.Y.
Moore Products Co., H & Lycoming Sts., Phila. 24, Pa.
Nurnberg Thermometer Co., Inc., 124 Livingston Sts., Brooklyn 2, N.Y.
Partlow Corp., 2 Campion Rd., New Hartford, N.Y.
Pyrometer Instrument Co., 103 Lafayette St., N.Y.C. 13
Ranco Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 75)*
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 182)*
Spencer Thermostat Co., Unit of Metals & Controls Corp., 34 Forest St., Attleboro, Mass.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Wheelco Instruments Co., Harrison & Peoria Sts., Chicago 7, Ill.

CONVERTERS, ELECTRICAL (See RECTIFIERS)

CONVEYORS (See also particular type & ELEVATORS)

Ajax Flexible Coupling Co., Inc., Westfield, N.Y. (Vibrating)
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Baldwin Duckworth Div., Chain Belt Co., 360 Plainfield St., Springfield, Mass.
Bonded Scale Co., 2176 S. 3rd St., Columbus 7, O.

Brady Conveyors Corp., 20 W. Jackson Blvd., Chicago 4, Ill.
Cleveland Crane & Engrg. Co., Wickliffe, O. (Overhead)
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
W. J. Finnegan Co., 7402 Santa Monica Blvd., Los Angeles 46, Cal. *(p. 116)*
Gifford-Wood Co., Hill St., Hudson, N.Y.
Island Equip. Corp., 101 Park Ave., N.Y.C. 17
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Lamson Corp., Syracuse 1, N.Y.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

CONVEYORS, BELT

Alvey-Ferguson Co., Oakley Sta., Cin'ti., O.
Audubon Wire Cloth Corp., Richmond St. & Castor Ave., Phila. 34, Pa. (Wire Belt)
Beverage Engrg. & Equip. Co., 13301 Lakewood Hts. Blvd., Cleveland 7, O.
Bonded Scale Co., 2176 S. 3rd St., Columbus 7, O.
Brady Conveyors Corp., 20 W. Jackson Blvd., Chicago 4, Ill.
Conveyor Co., 3260 E. Slauson Ave., Los Angeles 11, Cal.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp., Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Gifford-Wood Co., Hill St., Hudson, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Island Equip. Corp., 101 Park Ave., N.Y.C. 17 (Wire Mech.)
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Lamson Corp., Syracuse 1, N.Y.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa. (Belting only)
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18 (Metal)

CONVEYORS, CHAIN

Alvey-Ferguson Co., Oakley Sta., Cin'ti., O.
Baldwin Duckworth Div., Chain Belt Co., 369 Plainfield St., Springfield, Mass.
Brady Conveyors Corp., 20 W. Jackson Blvd., Chicago 4, Ill.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Gifford-Wood Co., Hill St., Hudson, N.Y.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Lamson Corp., Syracuse 1, N.Y.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.

CONVEYORS, FOOD & SANITARY

Brady Conveyors Corp., 20 W. Jackson Blvd., Chicago 4, Ill.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Conveyor Co., 3260 E. Slauson Ave., Los Angeles 11, Cal.
Gifford-Wood Co., Hill St., Hudson, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.

CONVEYORS, MONORAIL, OVERHEAD & TROLLEY

American Monorail Co., 13107 Athens Ave., Cleveland 7, O.
Cleveland Crane & Engrg. Co., Wickliffe, O.
Island Equip. Corp., 101 Park Ave., N.Y.C. 17
Link-Belt Co., 2410 W. 18th St., Chicago 8, Ill.

CONVEYORS, ROLLER

Alvey-Ferguson Co., Oakley Sta., Cin'ti., O.
Beverage Engrg. & Equip. Co., 13301 Lakewood Hts., Blvd., Cleveland 7, O.
Bonded Scale Co., 2176 S. 3rd St., Columbus 7, O.
Brady Conveyors Corp., 20 W. Jackson Blvd., Chicago 4, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill
Gifford-Wood Co., Hill St., Hudson, N.Y.
Lamson Corp., Syracuse 1, N.Y.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.

CONVEYORS, SCREW & RIBBON

Brady Conveyors Corp., 20 W. Jackson Blvd., Chicago 4, Ill.
Conveyor Co., 3260 E. Slauson Ave., Los Angeles 11, Cal.
Gifford-Wood Co., Hill St., Hudson, N.Y.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Link-Belt Co., 2410 W. 18th St., Chicago 18, Ill.
D. J. Murray Mfg. Co., 1002-3rd St., Wausau, Wis.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill

CONVEYORS, VIBRATING

Ajax Flexible Coupling Co., Inc., Westfield, N.Y.
Bonded Scale Co., 2176 S. 3rd St., Columbus 7, O.

CONVEYOR COMPONENTS (See also CHAIN; also PULLEYS; also SPROCKETS, etc.)

Conveyor Co., 3260 E. Slauson Ave., Los Angeles 11, Cal.
Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp., Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Gifford-Wood Co., Hill St., Hudson, N.Y.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill. (Guards)

COOLANT COOLERS & COOLANT COOLING SYSTEMS (See also OIL COOLERS)

Airtemp Div., Chrysler Corp., 1119 Leo St., Dayton 1, O. *(p. 9)*
Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Frostrode Products, Inc., 19929 Exeter Rd., Detroit 3, Mich.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*

COOLERS (See particular type)

COOLING PONDS

Binks Mfg. Co., 3114 Carroll Ave., Chicago 12, Ill.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. *(p. 79)*
Jos. A. Martocello & Co., 229 N. 14th St., Phila. 7, Pa. *(p. 157)*
Phillips Cooling Tower Co., Inc., 114 Liberty St., N.Y.C 6
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Angeles 11, Cal. *(p. 80)*
Spray Engrg. Co., 114 Central St., Somerville 45, Mass.
Spraying Systems Co., 4021 W. Lake St., Chicago 24, Ill.
Thermal Industries, P.O. Box 725, Indio, Cal.
Water Cooling Corp., 71 Nassau, N.Y.C. 7
York Corp., York, Pa. *(p. 165)*

COOLING TOWERS (See also particular type)

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Binks Mfg. Co., 3114 Carroll Ave., Chicago 12, Ill.
Fluor Corp., Ltd., 2500 S. Atlantic Blvd., Los Angeles 22, Cal.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.

(Continued)

The Marley Company, Inc.

Fairfax and Marley Roads, Kansas City 15, Kansas

Representatives in All Principal Cities (Consult Classified Telephone Directory)

Water Cooling Towers of All Types and Capacities. Spray Nozzles

Marley Non-Clog Spray Nozzles

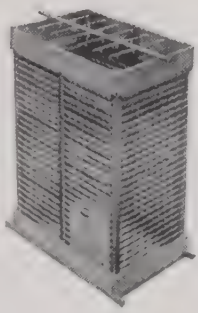
Low pressure tangential inlet with high speed whirl and uniform fine spray.

Two-Piece Nozzle for all cooling and spraying. Readily cleaned.

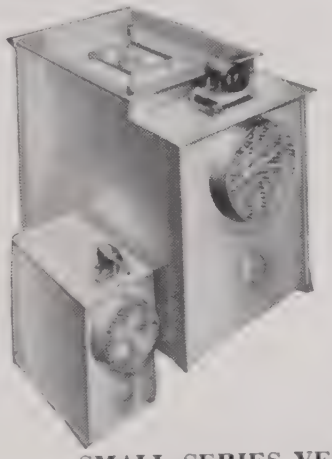
One-Piece Nozzle for cooling towers, spray ponds; large capacity.

Atomizing Nozzle for humidifying, vaporizing; a "mist-like" spray.

ATMOSPHERIC SPRAY TOWERS



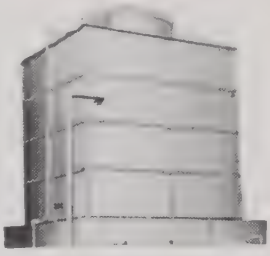
Marley Atmospheric Towers are designed for lowest cost, high efficiency installations on roof or ground where breeze is unobstructed. They are easily installed, simple and economical to operate. No fans, motors or other moving parts are employed. All are sturdily constructed from durable wood for long, trouble-free life. These towers owe their efficiency to the use of Marley low pressure spray nozzles and distribution systems. They are made in a wide range of closely graduated sizes and are shipped complete with necessary hardware, ready to erect.



MARLEY AQUATOWERS

Marley Aquatowers are built and shipped as complete units and are remarkable for simplicity of installation and operation. Their small space requirement makes them ideal for many applications either indoors or out. They are steel cased and are filled with nail-less redwood filling that retains correct alignment. Aquatowers are produced in two styles and seven sizes for specified refrigeration loads.

SMALL SERIES VERTICAL STEEL



These attractive induced draft steel cased towers are specifically designed for medium capacity refrigeration and air-conditioning service. Small Series Vertical towers are equipped with Marley Triple Effect eliminators, Marley low pressure spray nozzles and distribution system. They are prefabricated and piece-marked for simplified field assembly. V-belt fan drive is standard equipment for all models. Marley Geareducers are available for larger capacity towers.

MARLEY DOUBLE-FLOW TOWERS

For large capacity installations, Marley patented Double-Flow Towers are designed to provide maximum efficiency and flexibility.

COOLING TOWERS (Continued)

Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Harry Cooling Towers, Inc., West St., Doylestown, Pa. *(p. 80)*
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Lilie-Hoffmann Cooling Towers, Inc., 4239 Duncan Ave. St. Louis 10, Mo.
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. *(p. 79)*
Jos. A. Martocello & Co., 229 N. 14th St., Phila. 7, Pa. *(p. 157)*
New England Cooling Tower Co., 89 Broad St., Boston 10, Mass.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.

COOLING TOWERS

Atmospheric • Mechanical
Redwood • Steel

SANTA FE TANK & TOWER CO.

LOS ANGELES • SAN FRANCISCO
NEW YORK • TULSA • HOUSTON

Phillips Cooling Tower Co., Inc., 114 Liberty St., N.Y.C. 6
J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Kan. *(p. 81)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59
Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Angeles 11, Cal. *(p. 80)*
Thermal Industries, P.O. Box 725, Indio, Cal.
Trane Co., La Crosse, Wis. *(p. 14)*
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
C. H. Wheeler Mfg. Co., 1741 Sedgley Ave., Phila. 32, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

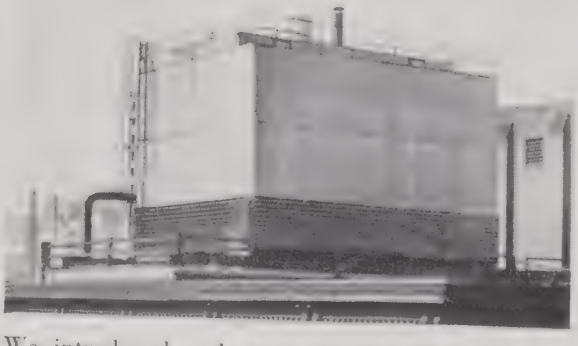
COOLING TOWERS, ATMOSPHERIC

Binks Mfg. Co., 3114 Carroll Ave., Chicago 12, Ill.
Fluor Corp., Ltd., 2500 S. Atlantic Blvd., Los Angeles 22, Cal.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Harry Cooling Towers, Inc., West St., Doylestown, Pa. *(p. 80)*
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Lilie-Hoffmann Cooling Towers, Inc., 4239 Duncan Ave., St. Louis 10, Mo.
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. *(p. 79)*
D. J. Murray Mfg. Co., 1002-3rd St., Wausau, Wis.
New England Cooling Tower Co., 89 Broad St., Boston 10, Mass.
ohn Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Phillips Cooling Tower Co., Inc., 114 Liberty St., N.Y.C. 6

(Continued)

HARRY-BURHORN

Water Cooling Equipment
Serving the industry since 1900



A complete line of
Atmospheric, Induced
Draft, Forced Draft,
Horizontal Flow Towers
and Evaporative Condensers

Large and Small
Outdoor and Indoor

We introduced and patented Slip Fit construction in our Towers over Fifteen years ago eliminating the use of nails in Harry Framing, Eliminators, and Decks. We also use Slip Fit joints in our standard Transite and Metal casing Mechanical Draft Towers.

Take advantage of our 48 Years Experience

HARRY COOLING TOWERS INC.

Works office P. O. Box 110, Doylestown, Pa. Phone 4458
Branches: 554 West 48th Street, New York 19, N.Y. Phone Columbus 5-6700
Professional Bldg., St. Petersburg, Fla. Phone 7-2795

J. F. Pritchard & Co.

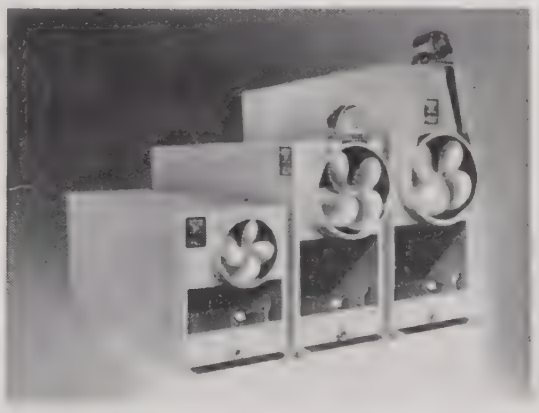
Equipment Division

908 Grand Avenue, Kansas City 6, Missouri

Amarillo, Atlanta, Baltimore, Chicago, Cincinnati, Dallas, Denver, Detroit, El Paso, Erie, Pa., Houston, Indianapolis, Little Rock, Los Angeles, Memphis, New Orleans, Newton, Iowa, New York, Oklahoma City, Omaha, Philadelphia, Pittsburgh, Portland, Ore., Richmond, Va., St. Louis, St. Paul, Salt Lake City, San Francisco, Spokane.

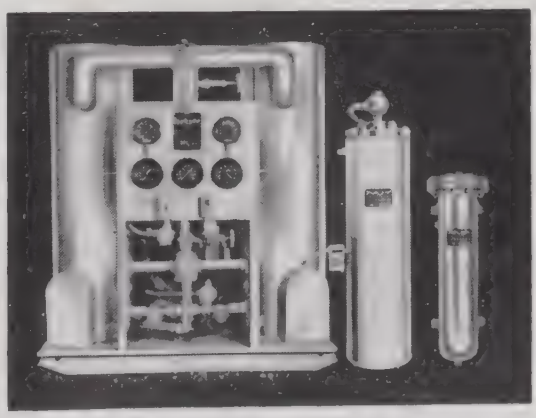
Cooling Towers—Heat Exchangers—Gas Equipment

This Pritchard Conventional Induced Draft Tower is representative of the many sizes, types, and designs which incorporate true counter-flow of air and water. All Pritchard towers are thoroughly engineered for longer life and trouble-free performance, adequately sized to perform at highest efficiency with lowest overall cost, according to the specific operating conditions and requirements. Pritchard cooling towers are available in material best suited to give completely satisfactory service.



Pritchard Series "P" package-type mechanical draft towers are available in several models, constructed of monel metal, aluminum, galvanized steel, or painted steel. For indoor or outdoor operation, the Series "P" tower is designed for operation in conjunction with refrigeration and air conditioning systems, air compressors, water jacketed oil or gas engines, and many other requirements that depend on low cost water.

The Pritchard AIRDFIN establishes new efficiency and economy in air cooled heat exchangers adaptable for jacket water cooling, steam condensing, and cooling vapors, gases, oils, acids, and other fluids. The AIRDFIN incorporates a flexibility for combination application, such as steam condensing and compressor jacket water and lube oil cooling. Special designs available in the "QUINTAIR" line of air cooled heat exchangers.



Pritchard HYDRYER models for gas or liquid dehydration by solid adsorption are available in standard or modified units for every specific application. The HYDRYER provides long efficient service for instrument air drying, prevention of equipment freeze-ups and corrosion, and industrial and laboratory process dew point control. Also available, the Pritchard PURIDRYER, using liquid desiccants for atmospheric air humidity control.

COOLING TOWERS (Continued)

J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Kan. *(p. 81)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Aneg-les 11, Cal. *(p. 80)*
Spraying Systems Co., 4021 W. Lake St., Chicago 24, Ill
Thermal Industries, P.O. Box 725, Indio, Cal.
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
Water Cooling Equip. Corp., Afton Sta., St. Louis 23, Mo

COOLING TOWERS, DECK TYPE

Binks Mfg. Co., 3114 Carroll Ave., Chicago 12, Ill.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Harry Cooling Towers, Inc., West St., Doylestown, Pa. *(p. 80)*
Lilie-Hoffmann Cooling Towers, Inc., 4239 Duncan Ave., St. Louis 10, Mo.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Phillips Cooling Tower Co., Inc., 114 Liberty St., N.Y.C. 6
J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Kan. *(p. 81)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Angeles 11, Cal. *(p. 80)*
Thermal Industries, P.O. Box 725, Indio, Cal.

COOLING TOWERS, INDOOR

Binks Mfg. Co., 3114 Carroll Ave., Chicago 12, Ill.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Harry Cooling Towers, Inc., West St., Doylestown, Pa. *(p. 80)*
Lilie-Hoffmann Cooling Towers, Inc., 4239 Duncan Ave., St. Louis 10, Mo.
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. *(p. 79)*
New England Cooling Tower Co., 89 Broad St., Boston 10 Mass.
Phillips Cooling Tower Co., Inc., 114 Liberty St., N.Y.C. 6
J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Kan. *(p. 81)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59
Thermal Industries, P.O. Box 725, Indio, Cal.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Water Cooling Corp., 71 Nassau St., N.Y.C. 7

COOLING TOWERS, MECHANICAL DRAFT

Binks Mfg. Co., 3114 Carroll Ave., Chicago 12, Ill.
Fluor Corp., Ltd., 2500 S. Atlantic Blvd., Los Angeles 22, Cal.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Harry Cooling Towers, Inc., West St., Doylestown, Pa. *(p. 80)*
Kennard Corp., 1819 S. Hanely Rd., St. Louis 17, Mo.
Lilie-Hoffmann Cooling Towers, Inc., 4239 Duncan Ave., St. Louis 10, Mo.
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. *(p. 79)*
New England Cooling Tower Co., Inc., 114 Liberty St., N.Y.C. 6
J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Kan. *(p. 81)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C 59
Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Angeles 11, Cal. *(p. 80)*
Thermal Industries, P.O. Box 725, Indio, Cal.
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
Water Cooling Equip. Corp., Afton Sta., St. Louis 23, Mo.
C. H. Wheeler Mfg. Co., 1741 Sedgley Ave., Phila. 32, Pa.

COPPER (See particular mill forms, i.e., BAR, SHEET, etc.)

COPPERSMITHING

Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.

CORDS, ELECTRIC (See WIRE, ELECTRIC; also HARNESSES, ELECTRICAL)

CORK (See INSULATION)

CORROSION INHIBITORS (See INHIBITORS, CORROSION)

COTTER PINS

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Hindley Mfg. Co., 53 John St., Valley Falls, R.I.
Lamson & Sessions Co., 1971 W. 8th St., Cleveland 2, O.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

COUNTING & COMPUTING DEVICES

Veeder-Root, Inc., Garden & Sargeant Sts., Hartford 2, Ct.

COUPLINGS, ELECTROMAGNETIC

Elliott Co., Jeanette, Pa.

COUPLINGS, PIPE (See FITTINGS, PIPE)

COUPLINGS, SHAFT (See SHAFT COUPLINGS)

COUPLINGS, MECHANICAL

Dresser Mfg. Div., Dresser Industries, Inc., 490 Fisher Ave., Bradford, Pa.
Gustin-Bacon Mfg. Co., Kansas City 7, Mo.
Hamilton Kent Mfg. Co., Kent, O. (Flexible)
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Link-Belt Co., 2410 W. 18th St., Chicago 8, Ill.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Phila. 20 Pa.
Ross Sprinkler Co., 34 Roberts St., Pasadena 3, Cal.
Victaulic Co. of America, 30 Rockefeller Plaza, N.Y.C. 20
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
J. A. Zurn Mfg. Co., Erie, Pa.

COUPLINGS, SELF-SEALING

Aeroquip Corp., 300 S. East Ave., Jackson, Mich. *(p. 83)*

COVERS, MANHOLE (See ACCESS DOORS)

CRADLES, MOTOR (See BASES)

CRANES & HOISTS

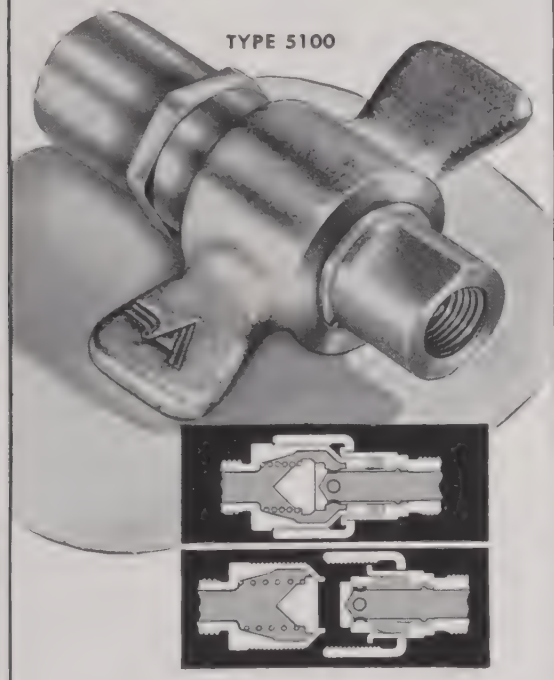
American Monorail Co., 13107 Athens Ave., Cleveland 7, O.
Euclid Crane & Hoist Co., 136 S. Chardon Rd., Euclid 17, O.
Frick Co., Waynesboro, Pa. *(p. 47)*
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Reading Chain & Block Corp., 2100 Adams St., Reading, Pa.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Shepard Niles Crane & Hoist Corp., Schuyler Ave., Montour Falls, N.Y. (Ice)
Union Mfg. Co., New Britain, Ct.

CRANKSHAFTS

Atlas Drop Forge Co., 209 W. Mt. Hope Ave., Lansing 2, Mich.
Interstate Drop Forge Co., 4035 N. 27th St., Milwaukee, Wis.
Modern Machine Wks., Inc., 5355 S. Kirkwood Ave., Cudahy, Wis. *(p. 195)*
Saginaw Malleable Iron Div., Gen'l. Motors Corp., Saginaw, Mich.

Aeroquip

SELF-SEALING COUPLINGS
FOR REFRIGERATION AND
AIR CONDITIONING UNITS



With the installation of Aeroquip Self-Sealing Couplings, fluid-carrying lines can be disconnected without loss of liquid or inclusion of air upon reconnection. Cooling units can thus be exchanged or serviced in less time.

Aeroquip Self-Sealing Couplings are suitable for lines carrying hydraulic fluids, oils, fuels, coolants or refrigeration gases. Write for further details.

Aeroquip for better performance, maintenance and service.

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JACKSON, MICHIGAN

303 Wareham Bldg., Hagerstown, Md.
327 M & M Building, Houston 2
1709 W. 8th St., Los Angeles 14
803 Pence Building, Minneapolis 3

CUBE MACHINES (See ICE CUBE CUTTERS; also ICE MANUFACTURING MACHINES, AUTOMATIC)

CUPS, GREASE & OIL (See LUBRICATORS)

CYLINDERS, COMPRESSED GAS, REFRIGERANT etc.

Harrisburg Steel Corp., P.O. Box 329, Harrisburg, Pa.
Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17 *(p. 197)*
Pressed Steel Tank Co., 1471 S. 66th St., Milwaukee 14, Wis. *(p. 194)*

DAIRY REFRIGERATORS (See DISPLAY CASES)

DAMPENERS, VIBRATION (See VIBRATION ABSORBER)

DAMPERS

Barber-Colman Co., Rockford, Ill.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
E. K. Campbell Heating Co., 1809 Manchester St., Kansas City 3, Mo.
Corbman Bros., 315 N. 7th St., Phila. 6, Pa.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Milcor Steel Co., Milwaukee, Wis.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
Young Regulator Co., 5209 Euclid Ave., Cleveland 3, O

DAMPER MOTORS

Barber-Colman Co., Rockford, Ill.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Sampsel Time Control, Inc., 600 N. Strong Ave., Spring Valley, Ill.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Young Regulator Co., 5209 Euclid Ave., Cleveland 3, O.

DAMPER REGULATORS

Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Brown Instrument Co., Div. Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
D'Este Div., American Chain & Cable Co., Inc., Reading, Pa.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
Sampsel Time Control, Inc., 600 N. Strong Ave., Spring Valley, Ill.
Schade Valve Mfg. Co., 2527 Bodine St., Phila. 33, Pa.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Tuttle & Bailey, Inc., New Britain, Ct.
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
Young Regulator Co., 5209 Euclid Ave., Cleveland 3, O.

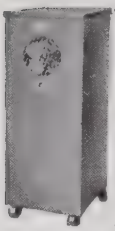
DECORATIONS, METAL (See also NAMEPLATES)

American Emblem Co., Inc., 9 Genesee St., New Hartford, Utica 1, N.Y.
Fox Co., Fox Lane, Cin'ti. 23, O.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich. *(p. 126)*

DEFROSTERS, AUTOMATIC (See CONTROLS, DEFROSTING CYCLE)

DEHUMIDIFIERS, CHEMICAL

Bryant Heater Co., 17825 St. Clair Ave., Cleveland 10, O.
Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
W. A. Hammond Drierite Co., 120 Dayton Ave., Xenia, O. *(p. 84).*



CHEMICAL DEHYDRATORS

Automatically controlled, for preventing dampness and condensation in basement rooms in summer and in unheated rooms during winter, also for radiant cooling systems.

THE H. J. KAUFMAN COMPANY
13215 Roselawn Ave., Detroit 4, Michigan

Florite Desiccant
Low Cost Granular
Drying Agent For
Air, Refrigerants, and
Other Gases and Liquids.
Performance Proved in
A Variety of Installations
FLORIDIN COMPANY
WARREN, PA.
Industrial Adsorbents since 1910

H. J. Kaufman Co., 13215 Roselawn Ave., Detroit 4, Mich. *(p. 84)*
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59

DEHUMIDIFYING AGENTS (See DEHYDRANTS)

DEHUMIDIFYING SYSTEMS & UNITS

Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
American Coils Co., 25 Lexington St., Newark 5, N.J.
Bryant Heater Co., 17825 St. Clair Ave., Cleveland 10, O.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Drying System, Inc., 1810¼ Foster Ave., Chicago 40, Ill.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
W. A. Hammond Drierite Co., 120 Dayton Ave., Xenia, O. *(p. 84)*
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
Pittsburgh Lectrodryer Corp., P.O. Box 1766, Pittsburgh 30, Pa.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Tamms Silica Co., 228 N. La Salle St., Chicago 1, Ill.
York Corp., York, Pa. *(p. 163)*

DEHYDRANTS

Aluminum Co. of America, Pittsburgh, Pa.
Davison Chemical Corp., Baltimore 3, Md. (Silica Gel) *(p. 85)*
Floridin Co., Warren, Pa. *(p. 84)*
W. A. Hammond Drierite Co., 120 Dayton Ave., Xenia, O. *(p. 84)*
Highside Chemicals Co., 195 Verona Ave., Newark 4, N.J.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Michigan Alkali Div., Wyandotte Chemicals Corp., Wyandotte, Mich.
Tamms Silica Co., 228 N. La Salle St., Chicago 1, Ill.

DEHYDRATORS (See DRIERS)

FOR DRYING SOLIDS—LIQUIDS—GASES

SOLIDS·LIQUIDS·GASES·SOLIDS·LIQUIDS·GASES·SOLIDS·LIQUIDS·GASES
DRIERITE

"The Versatile Desiccant"

DRIERITE is Neutral, Stable, Constant in Volume, Inert except toward water, Insoluble in Organic Liquids, Non-Disintegrating, Non-Wetting, Non-Poisonous, Non-Corrosive, Regenerative, Low in Cost.

DRIERITE dries all the common Gases—whether neutral, alkaline or acidic. Residual moisture amounts to 0.005 milligram per liter or 0.31 Lb. per million cubic feet.

DRIERITE dries the ordinary Organic Liquids and Solvents in either Liquid or Vapor Phase.

DRIERITE dries all the modern Refrigerants and is in wide use both by manufacturers of new equipment and by service engineers.

DRIERITE is available in all the standard granule forms and in the form of moulded pieces of various shapes and sizes.

Drierite Will Solve Your Drying Problem

W. A. HAMMOND DRIERITE COMPANY
120 DAYTON AVE., XENIA, OHIO

DAVISON PA100

Refrigeration Grade SILICA GEL

Davison again proves its leadership! Its Research Laboratories, working in close cooperation with the Refrigeration Industry, have developed a manufacturing process which *assures* dust-free performance of the preferred drying agent. Davison PA-100 meets every requirement . . . for maximum capacity, fast action, removal of acids and corrosive materials, freedom from caking, elimination of channeling, safe use, no attack on metals or alloys.

Your jobber stocks Davison PA-100 Refrigeration Grade Silica Gel in factory charged dehydrators or for refilling. Ask for PA-100 Silica Gel — in the can with the blue label.

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DELICATESSEN REFRIGERATORS (See DISPLAY CASES)

DEODORIZERS (See ODOR ELIMINATION SYSTEMS)

DESICCANTS (See DEHYDRANTS)

DESICCATORS (See DRIERS)

DIAPHRAGMS

Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*

Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

DIALS, GAUGE, etc. (See also NAMEPLATES)

Acromark Co., 5 Morrell St., Elizabeth 4, N.J.
Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.

DICHROMATE (See WATER TREATING MATERIALS)

DIE CASTINGS (See CASTINGS, DIE)

DIETHYLENE GLYCOL

Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30. E. 42nd St., N.Y.C. 17

DIFFUSERS, AIR (See AIR DIFFUSERS)

DISCS, COMPRESSOR VALVE (See VALVE FLAPPERS, REEDS, etc.)

Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.

DISCS, METAL, COMPOSITION, etc.

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. (Hard Rubber) *(p. 186)*
Jenkins Bros., 80 White St., N.Y.C. 13
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.

DISHES, REFRIGERATOR

Corning Glass Wks., Corning, N.Y.
Sneath Glass Co., Hartford City, Ind.

DISPENSING MACHINES, REFRIGERATED (See VENDING MACHINES)

DISPLAY CASES, REFRIGERATED

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A,B) American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
(A,B) American Refrigerator & Machine, Inc., 2700 University, N.E., Minneapolis, Minn.
(A) Artkraft Mfg. Corp., Kibby St., & D.T. & I. R.R., Lima, O.
(B) Bally Case & Cooler Co., Bally, Pa.
(A,B) Bemco Mfg. Corp., 1504 Minor at Pike, Seattle 1, Wash.
(A) Brunswick-Balke-Collender Co., 623 S. Wabash Ave., Chicago 5, Ill.
(A) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(A,B) Cleveland Refrigerator Co., 2901 E. 65th St., Cleveland 4, O.
(A,B) Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
(B) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
(A) Electro-Kold, Div. of E. S. Matthews, Inc., S. 151 Post St., Spokane 8, Wash.
(A,B) Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
(A,B,C) Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37 Pa.
(B) Fraser & Johnston Co., 725 Potrero Ave., San Francisco 10, Cal.
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 6, Tex.
(A,B,C) Gem Refrigerator Co., 2539 Germantown Ave., Phila. 33, Pa.
(A,B,C) General Refrigerator & Store Fixtures Co., 856 N. Broad St., Phila., Pa.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St. San Francisco 7, Cal.
(B) John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(A) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
(A,B,C) Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
(A,B) Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
(A,B,C) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
(C) McCall Refrigerator Corp., Hudson, N.Y.
(A,B) McCray Refrigerator Co., Kendallville, Ind.
(A,B) Masterfreeze Corp., Sister Bay, Wis.
(A,B,C) Minneapolis Show Case & Fixture Co., 1009 Washington Ave., S., Minneapolis, Minn.
(A,B,C) Modern Appliance Co., 111 S. Ellsworth, San Mateo, Cal.
(B) Morton Show Cases, Inc., Washington Courthouse, O.
(A,B) Nanticoke Refrigerator Manufacturers, Corner Hill & Slope Sts., Nanticoke, Pa.
(A,B,C) Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
(B) Paley Mfg. Corp., 244 Herkimer St., Brooklyn 16, N.Y.
(A) W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
(B) Royal Store Fixture Co., 847 N. Broad St., Phila. 23, Pa.
(B) St. Louis Butchers' Supply Co., 1543 . 15th St., St. Louis 6, Mo.
(A) Schaefer, Inc., 801 Washington Ave., N., Minneapolis 1, Minn.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
(B) Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.
(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
(A,B) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
(A,B) Charles Q. Sherman Corp., 149 Broadway, N.Y.C 6
(B) Spir-O.Freez Co., Inc., 1077 Castleton Ave., Staten Island 10, N.Y.
(B) Star Metal Mfg. Co., Trenton Ave. & Ann St., Phila. 34, Pa.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 178)*
(A,B) Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.
(C) Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
(A,B) Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.
(A,B) Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.
(A) Wells & Brunnell Mfg. Co., Inc., P.O. Box 1555, Tacoma 1, Wash. (Open Type)

DISPLAY CASE DOORS & FRAMES

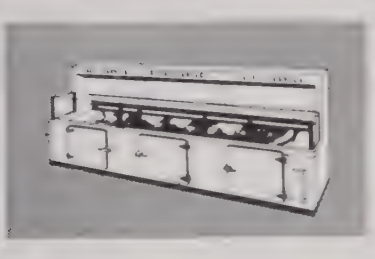
American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*

(*Continued*)

HUSSMANN

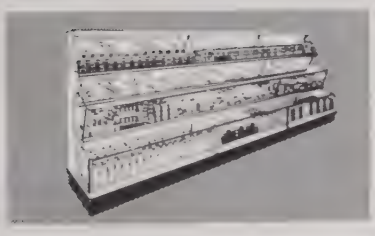
The Name That Stands for Completeness

IN FOOD REFRIGERATION EQUIPMENT



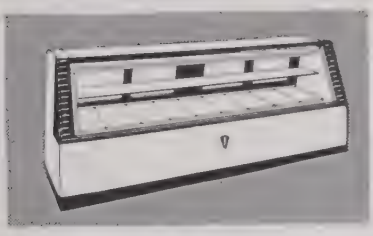
Self-Service Open Display Refrigerator for Dairy and Delicatessen Foods.

Self-Service Open Display Refrigerator for Produce.

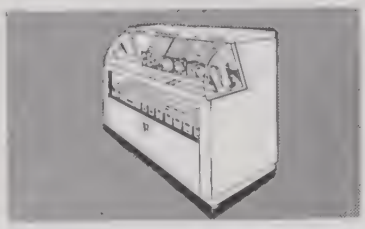


Triple-Deck Self-Service Open Display Refrigerator for Dairy Foods.

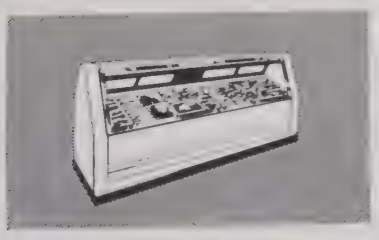
Market Coolers, in sizes to meet every requirement.



Double-Duty Case for the Service Meat Department.

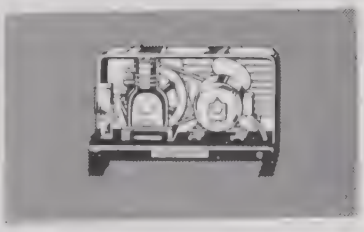


Open Self-Service Refrigerator for Frozen Foods.



Continuous Top Service Meat Case.

Condensing Units and Coils in every needed size.



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DISPLAY CASE DOORS & FRAMES (Continued)

Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
Luzerne Rubber Co., Trenton 9, N.J.
McCall Refrigerator Corp., Hudson, N.Y.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Modern Appliance Co., 111 S. Ellsworth, San Mateo, Cal.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.

DISTRIBUTORS, CONDENSER WATER (See CONDENSER WATER DISTRIBUTORS)

DISTRIBUTORS, REFRIGERANT

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.

DOOR & FRAME ASSEMBLIES (See DISPLAY CASE DOORS or COLD STORAGE DOORS)

DOORS, COLD STORAGE (See COLD STORAGE DOORS)

DOORS, DISPLAY CASE (See DISPLAY CASE DOORS)

DOORS, EVAPORATOR

Aluminum Goods Mfg. Co., Manitowoc, Wis.
Croname, Inc., 3701 N. Ravenswood, Chicago 13, Ill.
Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Sneath Glass Co., Hartford City, Ind.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.

DOORS, FREEZER (See COLD STORAGE DOORS)

DOOR GASKET

Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Ball Bros. Co., Muncie, Ind.
Brasco Mfg. Co., Harvey, Ill.
Butcher Boy Cold Storage Door Co., Div. of Western Fixture & Equip. Co., 170 N. Sangamon St., Chicago 7, Ill.
Chase Industrial Refrierator Equip. & Engrg. Co., 630 Reading Rd., Reading, Cin'ti. 15, O.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
W. J. Dennis & Co., 1732 N. Kolmar Ave., Chicago 39, Ill.
Dryden Rubber Co., 1014 S. Kildare Ave., Chicago 24, Ill.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Inland Mfg. Div., Gen'l. Motors Corp., Dayton 1, O.
Jamison Cold Storage Door Co., Hagerstown, Md. *(p. 45)*
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. *(p. 89)*
Kason Hardware Corp., 127 Wallabout St., Brooklyn 6, N.Y.
Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 16, Ill.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Sponge Rubber Products Co., 106 Derby Place, Shelton, Ct. *(p. 188)*
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
York Corp., York, Pa. *(p. 163)*

DOUBLE DUTY REFRIGERATORS (See DISPLAY CASES)

DOUGH RETARDERS

(A—Self contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A,B) American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
Baker-Perkins, Inc., Saginaw, Mich.
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(A,B) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
Duparquet-Dubois Refrigeration Div., Nathan Straus-Duparquet, Inc., 33 E. 17th St., N.Y.C. 3
(A,B) Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
(A,B) Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St. San Antonio 6, Tex.
(A,B,C) Gem Refrigerator Co., 2539 Germantown Ave., Phila. 33, Pa.
(A,B,C) General Refrigerator & Store Fixtures Co., 856 N. Broad St., Phila., Pa.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
(B) John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B,C) Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 179)*
(A,B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
(A) Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
(A,B) Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
(A,B) Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
(A,B,C) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
McCall Refrigerator Corp., Hudson, N.Y.
(A,B) McCray Refrigerator Co., Kendallville, Ind.
(A,B) Masterfreeze Corp., Sister Bay, Wis.
(A,B) Minneapolis Show Case & Fixture Co., 1009 Washington Ave., S., Minneapolis, Minn.
(A,B,C) Modern Appliance Co., 111 S. Ellsworth, San Mateo, Cal.
(A,B) Nanticoke Refrigerator Manufacturers, Corner Hill & Slope Sts., Nanticoke, Pa.
(A,B,C) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(C) J. P. Pfeiffer & Sons, Inc., 200 N. Paca St., Baltimore 1, Md.
Royal Store Fixture Co., 847 N. Broad St., Phila 23, Pa.
(B) St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
(B) Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.
(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
(B) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
(A) Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(A,B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 178)*
(B) Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.

DRAFT ARMS (See FAUCETS)

DRAFT BEVERAGE COOLERS (See BEER COOLERS, DRAFT)

DRAWN SHAPES (See SHAPES, DRAWN)

DRIERS

Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 159)*
Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*

(Continued)

DOOR GASKETS

FOR REFRIGERATORS & COOLING ROOMS

Jarrow Door Gaskets are the approved line for original installation and for replacement. Service men and manufacturers alike prefer them for their quality, economy, and long life. Ninety per cent of all refrigerators and cooling rooms can be fitted with stock Jarrow Gaskets. Your jobber has a complete assortment. Consult us on your special gasket needs

DOOR GASKETS FOR EVERY NEED

A Waterproof gaskets for low temperature work in sharp freezers, dry ice cabinets, and store rooms. Will not stick, freeze, or harden. Cushions 3/8" to 5/8" high.

B Sponge rubber gaskets with coated fabric covering either regular or grease-proof Cushions from 1/4" to 5/8" high.

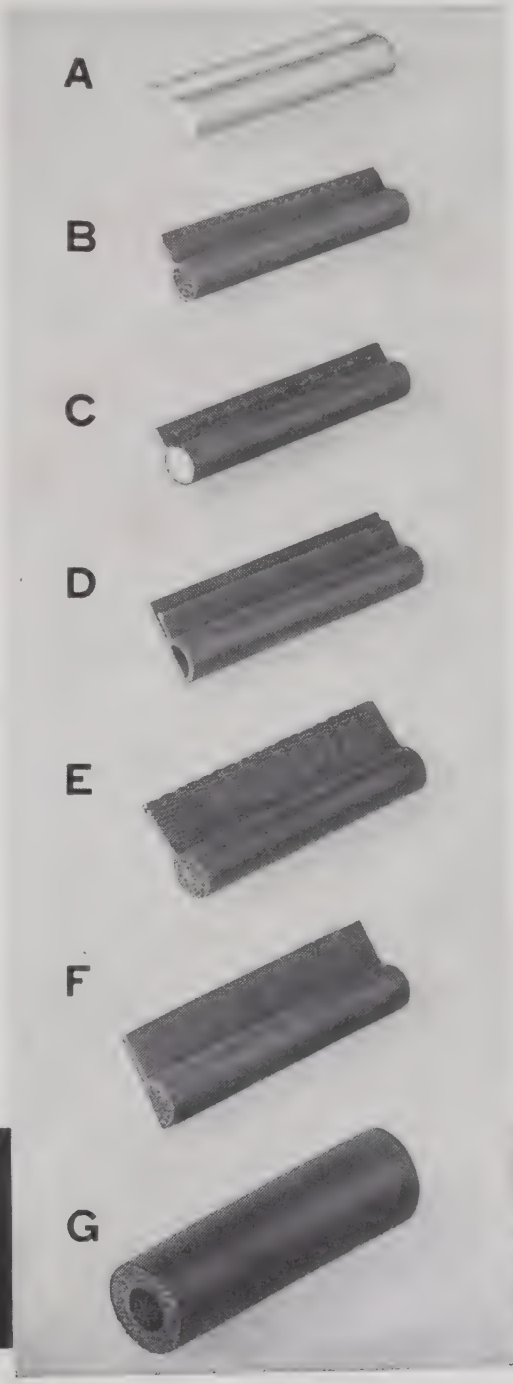
C Rubberized fabric door gaskets, with 1/8" to 1" cushion height, any size flanges, in black, white, or maroon with rubberized covering; also grease-proof in black only

D Extruded all-rubber gaskets, resilient, grease-resistant, odorless, and non-staining Long-lived black rubber. Any shape or size to fit any make of refrigerator; or to blueprint.

E Jarrow "Innerseal" gasket; sponge rubber cushion with rubber-covered woven wire flange Neoprene-dipped for grease-resistance Will withstand sub-zero temperatures. Cushions 1/4" to 5/8" high.

F Grease-proof rubber gaskets with sponge rubber cushions, sizes 1/4" to 1/2", with rubber on fabric flange. A fine grease-resistant gasket.

G Synthetic sponge rubber tubing, for insulating cold or return compressor lines. Thousands of tiny air cells prevent sweating and loss of efficiency Inside diameters 1/4" to 3/4"



Manufacturers:
Send us your gasket specifications. We make door gaskets and rubber insulation only and are best qualified to consult with you on your gasket problems.

JARROW PRODUCTS

General Office
420 NO. LA SALLE ST., CHICAGO 10
Factories: Chicago and Grand Rapids

DRIERS (Continued)

Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill. *(p. 55)*
Henry Valve Co., Melrose Park, Ill. *(p. 90)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Industrial Wire Cloth Products Corp., Wayne, Mich.
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 90)*
Mueller Brass Co., Port Huron, Mich.
Remco, Inc., 49th St. & A.V.R.R., Pittsburgh 1, Pa.
Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill.
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

DFN
DEHYDRATING SYSTEM

- Refillable type dehydrators with replaceable cartridges.
- Standard sealed type dehydrators, factory charged.
- Cartridges—all types of drying agents.

SPECIAL DEHYDRATING EQUIPMENT (low dew point)

For Refrigerants, oils, air, gases.

For dehydrating complete refrigeration systems and heat transfer assemblies in production.

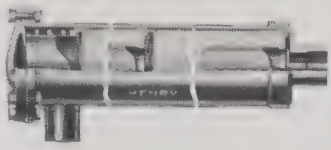
(Also full line of strainers and filters.)

McINTIRE CONNECTOR CO. • 252 Jefferson St., Newark, N. J.

HENRY

Cartridge Type Dehydrators

Types 756 and 757 with Dispersion Tube



Filled with Silica Gel. Cartridge easily removed and replaced without loosening end connections. Dispersion tube, which minimizes pressure drop and increases drying efficiency, is incorporated as an integral part of cartridge. By-passing of refrigerant between outside of cartridge and inner surface of dehydrator shell prevented by strong spring. Many other types incorporating "Abso-Dry" feature are also available.

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Melrose Park, Illinois, Suburb of Chicago.

APPROVED FOR USE BY ARMY • NAVY and MARITIME COMMISSION

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W. A. Hammond Drierite Co., 120 Dayton Ave., Xenia, O. *(p. 84)*
Pittsburgh Lectrodryer Corp., P.O. Box 1766, Pittsburgh 30, Pa.

DRINKING WATER BOTTLES, GLASS

Sneath Glass Co., Hartford City, Ind.

DRINKING WATER COOLERS, SELF-CONTAINED

Audiffren Refrigerating Co., Proctor, Vt.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Coolstream Corp., 240 Butler St.,Brooklyn 17, N.Y.
Copeland Refrigeration Corp., Sidney, O. *(p. 64)*
Cordley & Hayes, 541-4th Ave., N.Y.C. 16
Day & Night Mfg. Co., P.O. Box 150 Monrovia, Cal.
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Ebco Mfg. Co., 401 W. Town St., Columbus 8, O. *(p. 240)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
Glenwood-Inglewood Co., Glenwood Ave. at Thomas, Minneapolis 5, Minn.
Interstate Engrg. Corp., 2550 E. Imperial Highway, El Segundo, Cal.
Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
Perfecold, Inc., 1940 S. Main St., Los Angeles 7, Cal.
Puro Filter Corp. of America, 440 Lafayette St., N.Y.C 3
Sunroc Refrigeration Co., Glen Riddle, Pa. *(p. 91)*
Tal-Co Mfg. Co., 510 N. Dearborn St., Chicago 10, Ill. *(p. 90)*
Halsey W. Taylor Co., Warren, O.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
Westinghouse Elec. Corp., 653 Page Blvd., Springfield 2, Mass.

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Properly cooled water at its best with Kelvinator-refrigerated Electric Water Coolers. Sizes and models for every requirement. Write.

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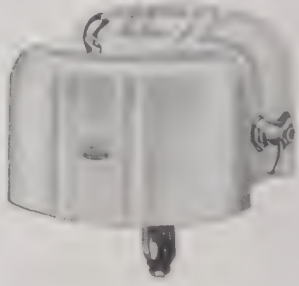

Tal-Co MANUFACTURING CO.
510 N. DEARBORN ST.
CHICAGO 10, ILL.

DRINKING WATER COOLERS, CAFETERIA

Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Cordley & Hayes, 541-4th Ave., N.Y.C. 16
Day & Night Mfg. Co., P.O. Box 150, Monrovia, Cal.
Ebco Mfg. Co., 401 W. Town St., Columbus 8, O. *(p. 240)*
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*
Interstate Engrg. Corp., 2250 E. Imperial Highway, El Segundo, Cal.
Perfecold, Inc., 1940 S. Main St., Los Angeles 7, Cal.
Sunroc Refrigeration Co., Glen Riddle, Pa. *(p. 91)*
Westinghouse Elec. Corp., 653 Page Blvd., Springfield 2, Mass.

DRINKING WATER COOLERS, ICE REFRIGERATED

Cordley & Hayes, 541-4th Ave., N.Y.C. 16
Gennett & Sons, Inc., 1 Main St., Richmond, Ind.
Star Metal Mfg. Co., Trenton Ave., & Ann St., Phila. 34, Pa.
Sunroc Refrigeration Co., Glen Riddle, Pa. *(p. 91)*

DRINKING WATER COOLER FITTINGS, FOUNTAINS, etc. (See also GLASS FILLERS)

James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Coolstream Corp., 240 Butler St., Brooklyn 17, N.Y.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Ebco Mfg. Co., 401 W. Town St., Columbus 8, O. *(p. 240)*
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
Puro Filter Corp. of America, 440 Lafayette St., N.Y.C. 3
Tal-Co Mfg. Co., 510 N. Daerborn Sts., Chicago 10, Ill. *(p. 90)*
Halsey W. Taylor Co., Warren, O.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
Watson-Stillman Co., Roselle, N.J. *(p. 113)*

DRINKING WATER COOLER LOWSIDES (See also WATER COOLERS)

Acme Industries, Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
Audiffren Refrigerating Co., Proctor, Vt.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Day & Night Mfg. Co., P.O. Box 150, Monrovia, Cal.
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*
Heat-X-Changer Co., Inc., 415 Lexington Ave., N.Y.C. 17 *(p. 240)*
Interstate Engrg. Corp., 2250 E. Imperial Highway, El Segundo, Cal.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Sunroc Refrigeration Co., Glen Riddle, Pa. *(p. 91)*
Tal-Co Mfg. Co., 510 N. Dearborn St., Chicago 10, Ill. *(p. 90)*
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*

DRIVES, CHAIN

Baldwin Duckworth Div., Chain Belt Co., 369 Plainfield St., Springfield, Mass.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
Morse Chain Co., Div. of Borg-Warner Corp., Ithaca, N.Y.
Ramsey Chain Co., Inc., 900 Broadway, Albany 1, N.Y.

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Spicer Mfg. Co., Div. of Dana Corp., 4100 Bennett Rd., Toledo 1, O

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Dayton Rubber Mfg. Co., 2342 W. Riverview Ave., Dayton 1, O.
R. & J. Dick Co., Inc., Passaic, N.J.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

DRIVES, VARIABLE SPEED (See also SPEED CHANGERS; also VARIABLE SPEED TRANSMISSIONS)

Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Browning Mfg. Co., Inc., Maysville, Ky.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Link-Belt Co., 2410 W. 18th St., Chicago 8, Ill.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Scientiae Corp., 101 Pine St., Dayton 2, O.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.

DRY ICE (See also CARBON DIOXIDE)

Air Reduction Sales Co., 60 E. 42nd St., N.Y.C. 17
Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
Mathieson Alkali Wks., Inc., 60 E. 42nd St., N.Y.C. 17
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.

DRY ICE MAKING MACHINERY

Harrisburg Steel Corp., P.O. Box 329, Harrisburg, Pa.
Lummus Co., 420 Lexington Ave., N.Y.C. 17
Mathieson Alkali Wks., Inc., 60 E. 42nd St., N.Y.C. 17
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

DRYERS (See DRIERS)

DRYING EQUIPMENT, AIR OR GASEOUS PHASE

Bowser, Inc., Terryville, Ct.
H. J. Kaufman Co., 13215 Roselawn Ave., Detroit 4, Mich. *(p. 84)*
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 90)*
Pittsburgh Lectrodryer Corp., P.O. Box 1766, Pittsburgh 30, Pa.
J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Kan. *(p. 81)*
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

DUCK (See also CANVAS)

U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Webb Mfg. Co., 4th & Cambria Sts., Phila. 33, Pa.

DUCT ACCESSORIES & FITTINGS (See also QUADRANTS)

Barber-Colman Co., Rockford, Ill.
Philip Carey Mfg. Co., Lockland, Cin'ti, 15, O.
Corbman Bros., 315 N. 75th St., Phila. 6, Pa.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
Haveg Corp., Marshallton, Del.
Milcor Steel Co., Milwaukee, Wis.
L. J. Mueller Furnace Co., 2005 W. Oklahoma Ave., Milwaukee 7, Wis.
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U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
Williamson Heater Co., 216 E. 6th St., Cin'ti. 2, O.

DUCT INSULATION & LINING (See also INSULATION)

Airtex Corp., 333 N. Michigan Ave., Chicago 1, Ill.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Baldwin-Hill Co., 500 Breuning Ave., Trenton 2, N.J.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Cellufoam Products Div., Masonite Corp., 111 W. Washington St., Chicago 2, Ill.
Cork Import Corp., 39 Park Place, Englewood, N.J.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Robert A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
Munn and Steele, Inc., 130 Lister Ave., Newark 5, N.J.
National Gypsum Co., 325 Delaware Ave., Buffalo 2, N.Y. *(p. 133)*
Refractory & Insulation Corp., 120 Wall St., N.Y.C. 5
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Wood Conversion Co., First Nat'l. Bank Bldg., St. Paul 1, Minn. *(p. 139)*

DUCTWORK PREFABRICATED

E. H. Allen Co., 22 Dorrance St., Boston 29, Mass.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Colonial Iron Wks. Co., 17643 St. Clair Ave., Cleveland 10, O.
Corbman Bros., 315 N. 7th St., Phila. 6, Pa.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Haveg Corp., Marshallton, Del.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
Williamson Heater Co., 216 E. 6th St., Cin'ti. 2, O.

DYNAMOTORS

Continental Elec. Co., Inc., 325 Ferry St., Newark 5, N.J.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Westinghouse Elec. Corp., Lima, O.

EJECTORS (See also STEAM JET)

Foster Wheeler Corp., 165 Broadway, N.Y.C.
Tranter Mfg. Co., 105 Water St., Pittsburgh 22, Pa.
Westinghouse Elec. Corp., P.O. Box 7348, S. Phila. 1, Pa.
C. H. Wheeler Mfg. Col, 1741 Sedgley Ave., Phila. 32, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

ELECTRIC GENERATING PLANTS, DIESEL OR GASOLINE DRIVEN (See also MOTOR GENERATORS)

Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Buda Co., 154th & Commercial Ave., Harvey, Ill.
Buffalo Gasolene Motor Co., 1280 Niagara St., Buffalo 13, N.Y.
Caterpillar Tractor Co., Peoria 8, Ill.
Crocker-Wheeler Div., Joshua Hendy Iron Wks., Ampere, N.J.
Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.
LeRoi Co., 1706 S. 68th St., Milwaukee 14, Wis.
Ready-Power Co., 11231 Freud Ave., Detroit 14, Mich. *(p. 95)*
Superior Engine Div., National Supply Co., 1401 Sheridan Ave., Springfield, O.
U. S. Motors Corp., 559 Nebrsaka St., Oshkosh, Wis.
Universal Motor Co., 430 Universal Drive, Oshkosh, Wis.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.
Witte Engine Wks., Div. of Oil Well Supply Co., 1600 Oakland Ave., Kansas City 3, Mo.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

ELECTRIC HEATER ELEMENTS (See HEATER ELEMENTS)

ELECTRICAL WIRING HARNESSES (See HARNESSES)

ELECTRODE COOLING SYSTEMS & COMPONENTS

Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Frostrode Products, Inc., 19929 Exeter Rd., Detroit 3, Mich.
Universal Welder Corp., 735 Carnegie Ave., Cleveland 15, O.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

ELECTRODES, WELDING (See WELDING ELECTRODES & RODS)

ELECTROPLATING SUPPLIES (See also ANODES)

General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
United Chromium, Inc., 51 E. 42nd St., N.Y.C. 17
U. S. Galvanizing & Plating Equip Corp.., 27 Heyward St., Brooklyn 11, N.Y.

ELEVATORS, BUCKET (See also CONVEYORS)

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Link-Belt Co., 2410 W. 18th St., Chicago 8, Ill.
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ELIMINATORS (See particular type, i.e., ODOR, SPRAY, VIBRATION ABSORBERS, etc.)

ELLS (See FITTINGS)

ELLS, WELDING

Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
John Simmons Co., Inc., 50 Church St., N.Y.C. 7

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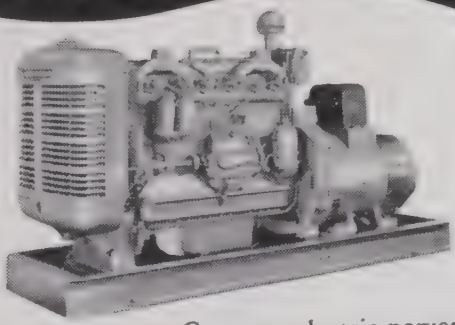
ENGINE GENERATORS
for Dependable, Low Cost, Independent Electric Power



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Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 113)*

EMBLEMS, METAL (See NAMEPLATES)

ENAMEL (See FINISHES)

ENAMEL, BITUMASTIC (See also COMPOUNDS, MASTIC)

Benjamin Foster Co., 4635 W. Girard Ave., Phila 31, Pa.
Alfred Hague & Co., 227-34th St., Brooklyn 32, N.Y.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.

ENAMELING, PORCELAIN (See also NAMEPLATES, etc.)

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Chicago Vitreous Enamel Product Co., 1407 S. 55th Court, Cicero 50, Ill.
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

ENCLOSURER, WIRE (See GUARDS)

ENGINES, BUTANE

Continental Motors Corp., Muskegon 82, Mich.
LeRoi Co., 1706 S. 68th St., Milwaukee 14, Wis.

ENGINES, DIESEL

American Locomotive Co., 30 Church St., N.Y.C.
Atlas Imperial Diesel Engine Co., Oakland, Cal. *(p. 94)*
Baldwin Locomotive Wks., Phila. 42, Pa.
Buckeye Machine Co., E. O'Connor Ave., Lima, O.
Buda Co., 154th & Commercial Ave., Harvey, Ill.
Caterpillar Tractor Co., Peoria 8, Ill.
Clark Bros. Co., Inc., Olean, N.Y.
Continental Motors Corp., Muskegon 82, Mich.
Cooper-Bessemer Corp., Mt. Vernon, O.
Cummins Engine Co., Inc., Columbus, Ind.
Detroit Diesel Engine Div., Gen'l. Motors Corp., 13400 W. Outer Dr., Detroit 23, Mich.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Lorimer Diesel Engine Co., 1530 Wood St., Oakland 7, Cal.
Niles Tool Wks. Div., General Machinery Corp., 545 N 3rd St., Hamilton, O.
Superior Engine Div., National Supply Co., 1401 Sheridan Ave., Springfield, O.
Union Diesel Engine Co., 2200 E. 7th St., Oakland 6, Cal
Waukesha Motor Co., Waukesha, Wis.
Witte Engine Wks., Div. of Oil Well Supply Co., 1600 Oakland Ave., Kansas City 3, Mo.
Wolverine Motor Wks., Inc., 35 Union Ave., Bridgeport 2, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

ENGINES, GAS

Atlas Imperial Diesel Engine Co., Oakland, Cal. *(p. 94)*
Bruce-Macbeth Engine Co., 2111 Center St., N.W., Cleveland 13, O.
Buda Co., 154th & Commercial Ave., Harvey, Ill.
Buffalo Gasolene Motor Co., 1280 Niagara St., Buffalo 13, N.Y.
Clark Bros. Co., Inc., Olean, N.Y.
Continental Motors Corp., Muskegon 82, Mich.
Cooper-Bessemer Corp., Mt. Vernon, O.
LeRoi Co., 1706 S. 68th St., Milwaukee 14, Wis.
Lorimer Diesel Engine Co., 1530 Wood St., Oakland 7, Cal.
McCulloch Motors Corp., 6101 W. Century Blvd., Los Angeles 45, Cal.
Waukesha Motor Co., Waukesha, Wis.
Western Engine Co., Div. of Vernon Tool Co., Ltd., 1101 Meridian Ave., Alhambra, Cal.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

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THE high thermal efficiency of the Diesel engine and the low cost of the fuel oil on which it operates, makes Diesel power the most economical motive power known for the operation of ice and refrigeration plants. The savings which can be effected by converting a plant from purchased electric power to Diesel drive will vary according to the local rates for electrical energy, price of fuel oil, and number of hours operation per year. Under average conditions the Diesel installation will earn the owner a dividend of 20% to 35% on the installed cost each year through savings of 50% to 75% in power cost.

Ask for Catalog No. 124 R.D.B.

Ice plants operating under the usual conditions of 20 pounds suction pressure, 185 pounds condensing pressure, and can water at 70 degrees, located in temperate climates, will use about 53 kilowatt hours of electricity per ton of ice produced. With electricity at 1½¢ per KWH, including demand charge, the power cost is 79½¢ per ton. There are in steady operation many Atlas Diesel engine driven plants as small as 20 tons daily rated capacity, which, under the same conditions, make a ton of ice for 29¢, including the following operating charges: 4.2 gallons fuel oil at 5½¢; lubricating oil at 50¢ per gallon (including allowance for renewing oil supply every 30 days) 2¢; sinking fund set aside for maintenance 4¢.

Modern vertical duplex ammonia compressors, up to and including 8″ x 8″, operate at 360 RPM, while the 9″ x 9″ and 10″ x 10″ machines turn at 300 RPM. Atlas Diesels are conservatively rated to operate at these same speeds, therefore the preferred arrangement is to direct connect the compressor to its engine, eliminating power and transmission losses and saving space, and utilizing a V-belt driven generator to supply electric power for the operation of auxiliary motors and lighting current.

The variable speed control permits the operation of Atlas Diesels at reduced speeds which makes them very flexible during periods when plants are operating at reduced capacity. This is impossible with electric motor driven plants because they are constant speed installations. The above subjects and many others are covered in detail in our Ice and Refrigeration Catalog, No. 124-RDB. Any ice maker who is interested in reducing his power costs should write for a copy of this catalog, which we will gladly mail without any obligation.

ENGINES, GASOLINE

Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Janos Asbestos Co., 39 Cortlandt St., N.Y.C. 7
LeRoi Co., 1706 S. 68th St., Milwaukee 14, Wis.
Waukesha Motor Co., Waukesha, Wis.

ENGINES, KEROSENE

Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Waukesha Motor Co., Waukesha, Wis.
Witte Engine Wks., Div. of Oil Well Supply Co., 1600 Oakland Ave., Kansas City 3, Mo.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

ENGINES, STEAM

Hooven, Owens, Rentschler Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Niles Tool Wks. Div., General Machinery Corp., 545 N. 3rd St., Hamilton, O.
Skinner Engine Co., Erie, Pa.
Troy Engine & Machine Co., 400 Railroad Ave., Troy, Pa.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
E. H. Wachs Co., 1525 Dayton St., Chicago 22, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

ENGINE JACKET-WATER COOLERS

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

ENGRAVING (See NAMEPLATES)

EQUALIZER TANKS (See SURGE TANKS; also ACCUMULATORS)

ESCUTCHEONS (See also NAMEPLATES)

American Emblem Co., Inc., 9 Genesee St., New Hartford Utica 1, N.Y.
Croname, Inc., 3701 N. Ravenswood, Chicago 13, Ill.
Fox Co., Fox Lane, Cin'ti. 23, O.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Scovill Mfg. Co., 99 Mill St.,Waterbury 91,Ct
Standard Molding Corp., 1517 E. 3rd St.,Dayton 1,O.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.
Stanley Wks., 195 Lake St., New Britain, Ct.

ETHYLENE GLYCOL

Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.

EVAPORATIVE CONDENSERS

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
Aerofin Corp., 410 Geddes St., Syracuse, N.Y. *(p. 11)*
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
American Coils Co., 25 Lexington St., Newark 5, N.J.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Baltimore Aircoil Co., Inc., 715 W. Pratt St., Baltimore 1, Md. *(p. 95)*
Betz Corp., 445 State St., Hammond, Ind.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y *(p. 51)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Frick Co., Waynesboro, Pa. *(p. 47)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
Harry Cooling Towers, Inc., West St., Doylestown, Pa. *(p. 80)*
Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 240)*
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
Larkin Coils, 519 S. Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
C. F. Moores Co., Inc., 1123 Ivy Hill Rd., Wyndmoor, Phila. 18, Pa.
Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal.

(*Continued*

ASRE Standard #20, "Methods of Rating and Testing Evaporative Condensers," gives complete and authoritative data on the method of rating, standard test methods, instruments, apparatus, test procedure, and definitions.

Copies can be obtained at 60¢ each from

The American Society of Refrigerating Engineers
40 West 40 Street
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B.A.C.
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715 West Pratt St.
Baltimore 1, Md.

Niagara Blower Company

General Sales Office: 405 Lexington Ave., New York 17, N.Y.

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Type "A"—Bulletin #107

NIAGARA AIR CONDITIONERS—
Type "C"—Bulletin #80

NIAGARA NO-FROST METHOD—
Bulletins #95 and #105

NIAGARA SPRAY COOLERS—
Bulletin #72, Part 3

NIAGARA DUAL COOLERS—
Bulletin #70

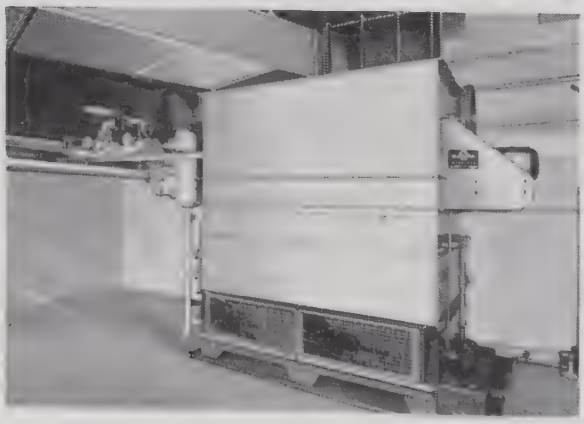
NIAGARA FAN COOLERS—
Bulletin #72, Part 2

NIAGARA AEROPASS CONDENSERS—
Bulletins #91 and #103

NIAGARA AERO HEAT EXCHANGER—
Bulletin #96

NIAGARA STANDARD COIL SECTIONS—Bulletin #92

NIAGARA MOTOR BLOWERS—
Bulletin #89, Part 1



NIAGARA AEROPASS CONDENSERS

Refrigeration plant operators, using the Niagara Aeropass Condenser, gain as much as 35% of power by reducing compressor head pressures. This gain is assured permanently because the patent Duo-Pass prevents scale depositing on Condenser tubes. Equipped with the Niagara "Oil-out," a system free from oil is obtained.

NIAGARA "No-Frost" METHOD

For refrigerated rooms below freezing, the Niagara "No-Frost" Method gives always full capacity because no ice ever forms on coils. It provides rapid, uniform cooling for products, trouble-free operation and lower operating expenses.

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Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Trane Co., La Crosse, Wis. *(p. 14)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*
Young Radiator Co., Racine, Wis. *(p. 16)*

EVAPORATIVE COOLERS (See ATMOSPHERIC COOLERS)

EVAPORATORS (See COILS, UNIT COOLERS, etc.)

EVAPORATORS, HOUSEHOLD REFRIGERATOR

Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Oakes North Chicago Div., Houdaille-Hershey Corp., 1900 Foss Park Ave., N. Chicago, Ill. *(p. 97)*
Standard Refrigeration Co., 232 S. Hoyne Ave., Chicago 20, Ill.
Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*

EXCHANGERS, HEAT (See HEAT EXCHANGERS)

EXPANDED METAL (See MESH, METAL)

EXPANSION JOINTS

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Chicago Metal Hose Corp., Maywood, Ill.
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
Dresser Mfg. Div., Dresser Industries, Inc., 490 Fisher Ave., Bradford, Pa.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill.
Reilly Tar & Chemical Corp., 1615 Merchants Bank Bldg. Indpls. 4, Ind.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 4
Zallea Bros. & Johnson, 9th St. & P.R.R., Wilmington 99, Del.
J. A. Zurn Mfg. Co., Erie, Pa.

EXPANSION VALVES, AUTOMATIC
(A—Ammonia; B—Methyl Chloride; C—Sulfur; D—Freon)

(A,B,C,D) Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
(B,C,D) Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
(A) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
(A,B,C,D) A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
(B,C,D) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
(B,C,D) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(A) General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
(A,B,C,D) Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
(B,C,D) Mayson Mfg. Co., Inc., 4332 Horatio St., Detroit 10, Mich.
(A,B,D) Refrigerating Specialities Co., 728 Sacramento Blvd., Chicago 12, Ill.
(A) Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.
(A,B,D) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,D) York Corp., York, Pa. *(p. 163)*

EXPANSION VALVES, HAND OR NEEDLE
(A—Ammonia; B—Other refrigerants)

(B) Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
(A) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
(A) Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(A,B) Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
(A,B) Henry Valve Co., Melrose Park, Ill. *(p. 232)*
(B) Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
(B) Mueller Brass Co., Port Huron, Mich.
(B) Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A) Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
(A,B) Watson-Stillman Co., Roselle, N.J. *(p. 113)*
(B) Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 163)*

EXPANSION VALVES, SPECIAL LOW TEMPERATURE
(A—Ammonia; B—Other refrigerants)

(A,B) Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
(B) Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
(B) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
(B) Flow Controls, Inc., 1821 W. North Ave., Chicago 22, Ill.
(A,B) Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.

(Continued)

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Step Plates
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Sharp Freezers
Liquid Coolers

Write for bulletin illustrating applications of BTC Evaporator Plates.

THE BREWER-TITCHENER CORP.
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Hydrogen Brazed • Internally Clean • Durable Finish • Efficient Circuits • Low Cost • Modern Production Lines • Years of Experience

All of these factors make this organization an outstanding source of supply for production quantities of household refrigerator evaporators and condensers.

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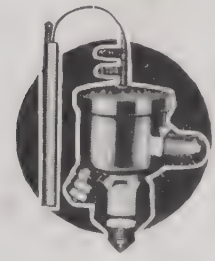
ALCO VALVE COMPANY

Engineered Refrigerant Controls
841 Kingsland Avenue, St. Louis 5, Missouri
New York Office: 122 East 42nd Street

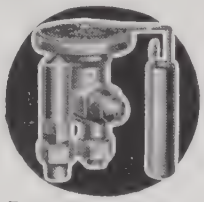
THE COMPLETE LINE OF REFRIGERANT CONTROLS

ALCO THERMO EXPANSION VALVES

for automatic control of liquid refrigerant on all types of refrigeration and air conditioning systems. *Capacities*—from fractional tonnage to 100 tons Methyl Chloride, 50 tons "Freon-12".

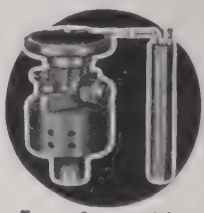


Thermo-Limit—with pressure limiting feature and capacity change.



Type TK—"3 valves in 1".

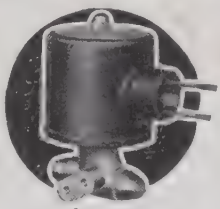
Type TCL



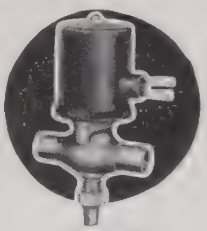
Type TR—Multi-Outlet

ALCO SOLENOID VALVES

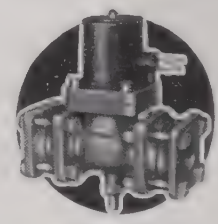
for all types of service. *For Liquid:* "Freon"—up to 75 tons. Methyl Chloride—up to 150 tons. *For Suction:* "Freon"—up to 8.8 tons. Methyl Chloride—up to 17 tons.



Type S1



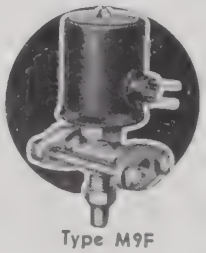
Type M3



Type R2

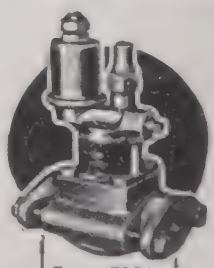
ALCO AMMONIA CONTROLS

Solenoid Liquid Valves—up to 172 tons. *Solenoid Suction Valves*—up to 28 tons. *Thermo Expansion Valves* — from fractional tonnage to 60 tons.



Type M9F

Type TG



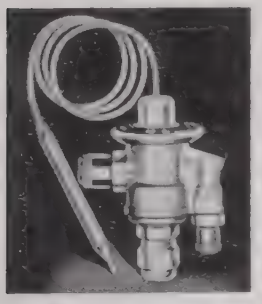
Type EPR13

ALCO ALSO MAKES: *Evaporator Pressure Regulators* for "Freon", Methyl Chloride and Ammonia with connection sizes up to 6"—*Solenoid Valves* for brine, water, gas, air and steam—*Float Switches*—*High Pressure Float Valves*—*Constant Pressure Expansion Valves*—*Liquid and Suction Line Strainers.*

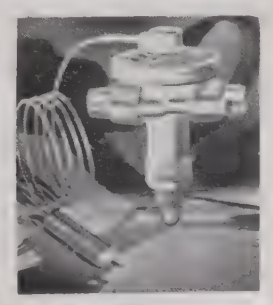
Automatic Products Company

2512 North Thirty Second Street
Milwaukee 10 Wisconsin

A-P Air Conditioning and Refrigeration Controls

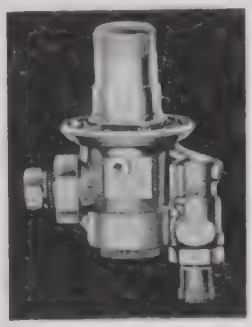


A-P Model No. 207
Maximum Capacity
½ ton Freon

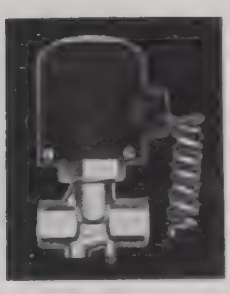


A-P Model No. 220-K
8 to 16 tons Freon

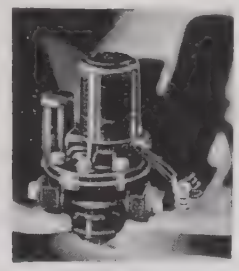
A-P Model 205, maximum capacity 2½ tons Freon



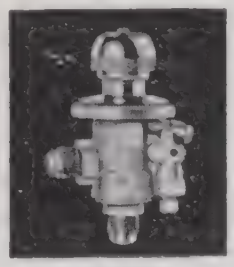
A-P Automatic Expansion Valves. Capacities up to 1 ton Freon



A-P Solenoid
No. 73RJ
Maximum Liquid Line Capacity, 5.4 tons Freon. Also available as a Water Solenoid (No. 71J).



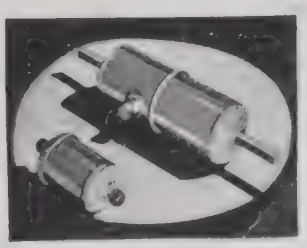
A-P Solenoid Water Valve. High capacity 2200 gals. per hour



A-P Constant Suction Pressure Regulator Valve No. 235-S. Sensitive, equipped with adjusting knob, with graduated pressure scale.

A-P Water Regulating Valve No. 68-B. Eliminates chatter and water hammer. Non-clogging, self cleaning valve stem gives longer, accurate control.

A-P "TRAP-IT"
Positive System-Protection against impurities and moisture.



A-P "TRAP-DRI"
Capacities up to 5 ton; some are rechargeable type. 100% Protection because it is a DRIER, STRAINER and FILTER.

A-P DEPENDABLE Refrigerant Valves widely used on refrigeration and air conditioning systems of all sizes and types, are well-known for their durability, long-life efficiency and accuracy in refrigerant control. Their reputation for DEPENDABILITY is based on long experience and knowledge in designing and engineering the kind of valves that help make every installation completely satisfactory in operating efficiency.

DEPENDABLE
REFRIGERANT VALVES

Detroit Lubricator Company

5900 Trumbull Ave., Detroit 8, Michigan, U.S.A.

Division of AMERICAN RADIATOR & Standard Sanitary CORPORATION

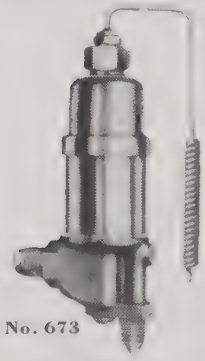
New York 18, N.Y., 40 West 40th Street **Chicago 5, Ill., 816 S. Michigan Avenue**

Export Department, Box 218, Ridgefield, New Jersey

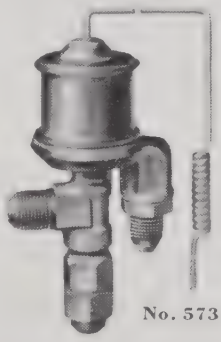
Canadian Representatives: Railway and Engineering Specialties Limited, Montreal, Toronto, Winnipeg

"DETROIT" THERMOSTATIC EXPANSION VALVES

"Detroit" Thermostatic Expansion Valves are designed to keep the evaporator in a refrigerating system completely refrigerated. Power elements are "gas charged" to a definite pressure for motor overload protection, providing quicker response and more sensitive control. Capacities ⅓ to 20 tons Freon-12. Needles and seats are of Delubaloy, a very hard, corrosion resistant alloy, which insures long trouble-free service.

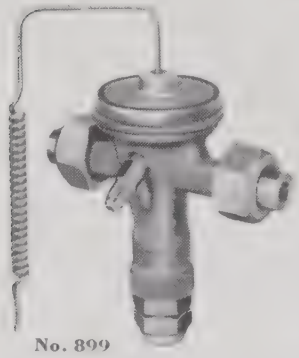


No. 673



No. 573

No. 673, the "standard of the industry," has a long record of dependability in a wide variety of installations. Designed for average size commercial and air conditioning installations, it has capacity up to 3½ tons Freon-12. Write for Bulletin No. 82.



No. 899

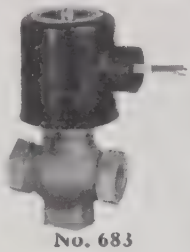
No. 573 provides the same dependability as No. 673 in smaller capacities. Its double diaphragm construction with gas charged power element permits close superheat control at low suction pressures and provides motor overload protection in its simplest, most effective form, using only one power element.

Other "Detroit" Thermostatic Expansion Valves carry capacities up to 20 tons Freon-12. All have made exceptional records of dependability and satisfactory service.

Special "Detroit" Valve No. 793 is the industry standby for operations at temperatures below minus 30°F.

Write for catalog No. 200B.

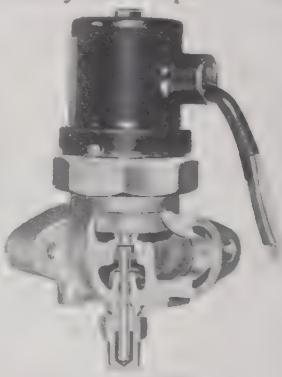
REFRIGERATION SOLENOID VALVES



No. 683

Solenoid Valves incorporate many outstanding features. The design is such that all objectionable A.C. hum is eliminated, thus making it adaptable for use in quiet locations. Conservative coil ratings insure positive opening against high pressures, and a strong kick-off spring assures positive, tight closing. Plungers and valve seats are made of non-magnetic alloy, eliminating any possibility of minute particles of metal, which might interfere with tight closure, accumulating on plunger or seat.

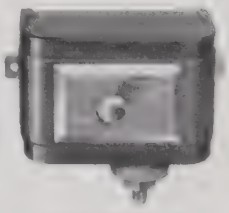
These valves require little space, are easily mounted, and may be readily taken apart in the field for cleaning, without the necessity of breaking the refrigerant line. "DL" Solenoid Valves are made in several sizes and capacities and for all standard voltages to handle the smallest or largest jobs.



No. 686

The 681 and 686 Solenoid Valves are pilot-operated. All Solenoid Valves draw 15 watts when energized. Write for Bulletin 197.

NO. 450 REFRIGERATION CONTROLS



No. 450FB-3

Every need for refrigeration temperature and pressure controls is satisfied by the "DL" line of #450 Controls. Included are instruments having a constant "cut-in" point or constant "cut-out" point. The range and differential adjustments are easily made by means of large knurled knobs inside the control case.

These controls are flexible. All power elements, switch assemblies, etc., are interchangeable in the field, thus making it possible to change the controls' characteristics—or to service them in the shortest possible time.

Attractively finished in black enamel with red cover plate. Conservatively rated to carry up to 1 H.P. load, 110 to 550 volts, A.C. Special switches available for D.C. Write for Bulletin 181

EXPANSION VALVES (Continued)

(B) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(A,B) Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
(B) Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17 Mo.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(B) Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
(A) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

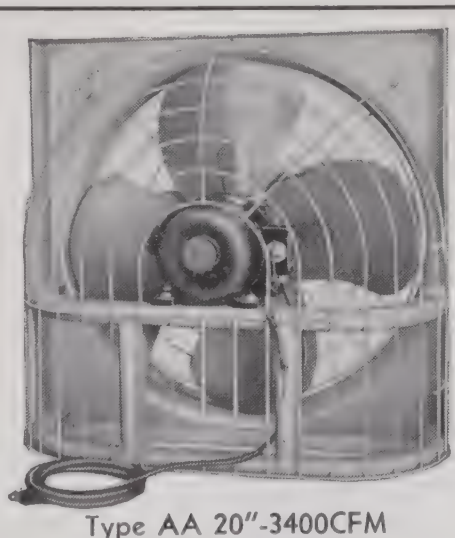
EXPANSION VALVES, THERMOSTATIC

(A—Ammonia; B—Methyl Chloride; C—Sulfur; D—Freon)

(A,B,C,D) Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
(B,C,D) Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
(B,C,D) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
(B,C,D) Flow Controls, Inc., 1821 W. North Ave., Chicago 22, Ill.
(B,C,D) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(B,C,D) General Controls Co., 801 Allen Ave., Glendale 1, Cal. *(p. 199)*
(B,C,D) Mayson Mfg. Co., Inc., 4332 Horatio St., Detroit 10, Mich.
(B,C,D) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(A,B,C,D) Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.
(B,D) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B,D) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

FANS (See also particular type & BLOWERS)

Aerovent Fan Co., 710 E. Ash St., Piqua, O.
Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
E. H. Allen Co., 22 Dorrance St., Boston 29, Mass.
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
Arex Co., 333 N. Michigan Ave., Chicago 1, Ill.
Bayley Blower Co., 66th & Burnham Sts., Milwaukee 14, Wis.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Burt Mfg. Co., 926 S. High St., Akron 11, O
C & H Air Conditioning Fan Co., Inc., 1603 DeKalb Ave., N.E., Atlanta, Ga.
Chelsea Fan & Blower Co., Inc., 1206 S. Grove St., Irvington 11, N.J. *(p. 101)*
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
De Bothezat Ventilating Equip., Div. of American Machine & Metals, Inc., E. Moline, Ill.
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
Duriron Co., Inc., Dayton 1, O.
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7, Ill.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
Garden City Fan Co , 332 S. Michigan Ave., Chicago 4, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y
Hartzell Propeller Fan Co., Div. of Castle Hills Corp., 910 S. Downing St., Piqua, O.
Haveg Corp., Marshallton, Del.
Holcomb & Hoke Mfg. Co., Inc., 1545 Van Buren St., Indpls. 7, Ind.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
King Co., 902 N. Cedar St., Owatonna, Minn
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., 128 Linden St., Allentown, Pa.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Moore Co., 544 Westport Rd., Kansas City 2, Mo.
Myers Elec. Co., 209-1st Ave., Pittsburgh 22, Pa.
National Engrg. & Mfg. Co., 213 W. 19th St., Kansas City 8, Mo.
Herman Nelson Corp., 1824-3rd Ave., Moline, Ill.
Peerless Elec. Co., 2000 W. Market St., Warren, O.
J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Kan. *(p. 81)*
Reed Unit-Fans, Inc., 1001 St. Charles Ave., New Orleans 8, La.
Robinson Ventilating Co., Zelienople, Pa.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Angeles 11, Cal. *(p. 80)*
Schwitzer-Cummins Co., Indpls. 7, Ind.
F. A. Smith Mfg. Co., Inc., Union & Augusta Sts., Rochester 2, N.Y.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Torrington Mfg. Co., 70 Franklin St., Torrington, Ct. *(p. 103)*
Trane Co., La Crosse, Wis. *(p. 14)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
U. S Thermo Control Co., 44 S. 12th St., Minneapolis 4, Minn.
Viking Air Conditioning Corp., 5600 Walworth Ave. Cleveland 2, O.
Water Cooling Equip. Corp., Afton Sta., St. Louis 23, Mo
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11



Type AA 20"-3400CFM
Type AA-24"-4800CFM

Totally Enclosed
Ball Bearing Motors

Will operate in any position
FOR QUICK FREEZING

CHELSEA FAN & BLOWER CO., INC.
1206 GROVE ST., IRVINGTON 11, N.J.

FANS, CIRCULATING

Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
C & H Air Conditioning Fan Co., Inc., 1603 DeKalb Ave., N.E., Atlanta, Ga.
Chelsea Fan & Blower Co., Inc., 1206 Grove St., Irvington 11, N.J. *(p. 101)*
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7, Ill.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hartzell Propeller Fan Co., Div. of Castle Hills Corp., 910 S. Downing St., Piqua, O.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Marathon Elec. Mfg. Corp., Wausau, Wis.
Myers Elec. Co., 209-1st Ave., Pittsburgh 22, Pa
Propellair, Inc., Springfield, O.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Robinson Ventilating Co., Zelienople, Pa.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Schwitzer-Cummins Co., Indpls. 7, Ind.
Signal Elec. Mfg Co., Menominee, Mich.
F. A. Smith Mfg. Co., Inc., Union & Augusta Sts., Rochester 2, N.Y.
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Trane Co., La Crosse, Wis. *(p. 14)*
Victor Elec. Products, Inc., 2950 Robertson Rd., Cin'ti. 9, O.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

FANS, EXHAUST & VENTILATING

Aerovent Fan Co., 710 E. Ash St., Piqua, O.
Air Controls, Inc., Div. of Cleveland Heater Co., 2310 Superior Ave., Cleveland 14, O.
Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
E. H. Allen Co., 22 Dorrance St., Boston 29, Mass.
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
Arex Co., 333 N. Michigan Ave., Chicago 1, Ill.
Bayley Blower Co., 66th & Burnham Sts., Milwaukee 14, Wis.
Thomas Beckett & Co., Inc., 2118 Griffin St., Dallas 2, Tex.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Burt Mfg. Co., 926 S. High St., Akron 11, O.
C & H Air Conditioning Fan Co., Inc., 1603 DeKalb Ave., N.E., Atlanta, Ga.
Chelsea Fan & Blower Co., Inc., 1206 Grove St., Irvington 11, N.J. *(p. 101)*
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
DeVilbiss Co., 300 Phillips Ave., Toledo 1, O.
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7, Ill.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
Garden City Fan Co., 332 S. Michigan Ave., Chicago 4, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hartzell Propeller Fan Co., Div. of Castle Hills Corp., 910 S. Downing St., Piqua, O.
Hirschman-Pohle Co., Inc., 21 Lent Ave., LeRoy, N.Y.
Holcomb & Hoke Mfg. Co., Inc., 1545 Van Buren St., Indpls. 7, Ind.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 11, Ill.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Marathon Elec. Mfg. Corp., Wausau, Wis.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Moore Co., 544 Westport Rd., Kansas City 2, Mo.
Myers Elec. Co., 209-1st Ave., Pittsburgh 22, Pa.
National Engrg. & Mfg. Co., 213 W. 19th St., Kansas City 8, Mo.
Herman Nelson Corp., 1824-3rd Ave., Moline, Ill.
Northern Blower Co., W. 65th St., South of Denison, Cleveland 2, O.
Peerless Elec. Co., 2000 W. Market St., Warren, O.
Reed Unit-Fans, Inc., 1001 St. Charles Ave., New Orleans 8, La.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Robinson Ventilating Co., Zelienople, Pa.
Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C 59
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Schwitzer-Cummins Co., Indpls. 7, Ind.
Signal Elec. Mfg. Co., Menominee, Mich.
F. A. Smith Mfg. Co., Inc., Union & Augusta Sts., Rochester 2, N.Y.
H. J. Somers, Inc., 6063 Wabash Ave., Detroit 8, Mich.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Trane Co., La Crosse, Wis. *(p. 14)*
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
Victor Elec. Products, Inc, 2950 Robertson Rd., Cin'ti. 9, O.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

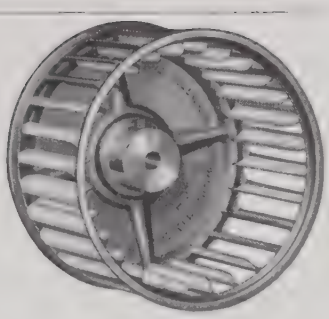
FANS, HIGH STATIC PRESSURE

Aladdin Heating Corp., 2222 San Pablo Ave., Oakland 12, Cal.
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
Bayley Blower Co., 66th & Burnham Sts., Milwaukee 14, Wis.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
Hartzell Propeller Fan Co., Div. of Castle Hills Corp., 910 S. Downing St., Piqua, O.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
Herman Nelson Corp., 1824-3rd Ave., Moline, Ill.
Robinson Ventilating Co., Zelienople, Pa.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
Trane Co., LaCrosse, Wis. *(p. 14)*
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

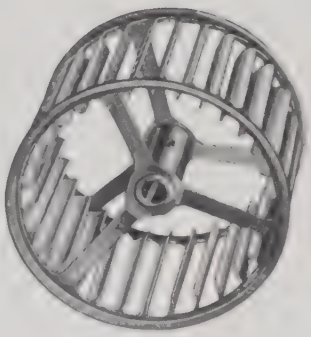
FAN BLADES & FAN PARTS (See also BLOWER WHEELS)

Thomas Beckett & Co., Inc., 2118 Griffin St., Dallas 2, Tex.
C & H Air conditioning Fan Co., Inc., 1603 DeKalb Ave., N.E., Atlanta, Ga.
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Mayne Products Co., 324 Harries Bldg., Dayton 2, O.
Myers Elec. Co., 209-1st Ave., Pittsburgh 22, Pa. (12" to 108")
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
Victor Elec. Products, Inc., 2950 Robertson Rd., Cin'ti. 9, O.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.

Vairified* AIR IMPELLERS

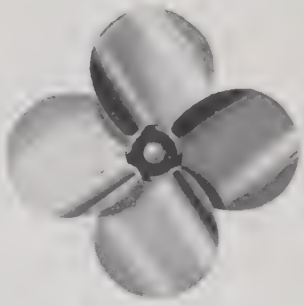


SHOULD BE SELECTED DURING EARLY STAGES OF UNIT DESIGN

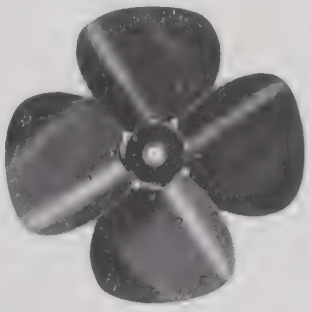


By consulting Torrington during the early stages of product design, manufacturers planning the inclusion of an air impeller in a new unit will secure the greatest good from our long experience in the application of forced air.

The annoying, sometimes embarrassing delays and expense of design changes can be avoided with expert help at the right time, which, we repeat, is *in the beginning*.



From among the wide range of sizes, pitches and styles of Airistocrat propeller type fan blades, or sizes and widths of Airotor blower wheels, the correct air impeller for your needs can usually be supplied. Write or phone for information.



*Vairified . . . aerodynamically designed for maximum operating efficiency with quietness; guaranteed to perform as rated under NAFM and NEMA Test Code conditions; hand gauged for uniform contour, alignment and balance. The Torrington Manufacturing Company.

TORRINGTON

MANUFACTURING COMPANY, TORRINGTON, CONN.

FARM FREEZERS (See HOME & FARM FREEZERS)

FASTENERS & FASTENING DEVICES

Central Screw Co., 3501 Shields Ave., Chicago 9, Ill
Dzus Fastener Co., Inc., Babylon, N.Y.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Shakeproof, Inc., 2501 N. Keller Ave., Chicago 39, Ill.
Stronghold Screw Products, Inc., 216 W. Hubabrd St., Chicago 10, Ill.
Tinnerman Products, Inc., 2038 Fulton Rd., Cleveland 13, O.
United-Carr Fastener Corp., 31 Ames St., Cambridge 42, Mass.
U. S.Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20,

FAUCETS, BEER, CARBONATED WATER, etc.

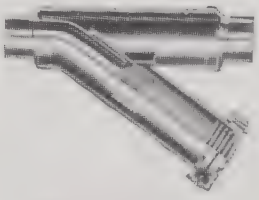
Hudson Products Co., Inc., 440 St. Aubin, Detroit 7, Mich.
North Penn Co., 72-5th Ave., N.Y.C. 11
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*

FAUCETS, DRINKING WATER (See DRINKING WATER COOLER FITTINGS)

FELT (See also INSULATION)

American Hair & Felt Co., 222 N. Bank Dr., Chicago 54 Ill.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Robt. A. Keasbey Co., 139 E. 19th St., N.Y.C 11 *(p. 138)*
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.

HENRY



"Y" Strainers
Type 895

Brass plated welded construction with forged brass connections (except F.P.T. sizes). Negligible pressure drop. Screen can be removed for cleaning without removing strainer from line. Large screen area. Light weight. Baffle construction prevents heavy particles from injuring screen, which is reinforced to prevent distortion. Furnished with 100 mesh screen. Complete line shown in Catalog No. 98.

HENRY VALVE CO.
Melrose Park, Illinois, Suburb of Chicago

APPROVED FOR USE BY
ARMY • NAVY
and MARITIME COMMISSION

FERMENTATION ROOMS, BAKERY

Baker-Perkins, Inc., Saginaw, Mich.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Fred D. Pfening Co., 1075 W. 5th Ave., Columbus 8, O.
Union Steel Products Co., 448 Pine St., Albion, Mich.

FIBRE, VULCANIZED, SHEET; ROD; TUBE, etc.

Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
Haveg Corp., Marshallton, Del.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Wilmington Fiber Specialty Co., P.O. Drawer 1028, Wilmington 99, Del.
Wolverine Fabricating & Mfg. Co., Inc., Princess St. & M.C.R.R., Inkster, Mich.

FILTERS (See particular type; also STRAINERS; also SCALE TRAPS)

FILTERS, AIR LINE

Air-Maze Corp., 5200 Harvard Ave., Cleveland 5, O.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Gas & Oil Industry Labs., Inc., 4 Paine Ave., Irvington 11, N.J.
Moore Products Co., H & Lycoming Sts., Phila. 24, Pa.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*

FILTERS, LIQUID

Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Liquid Conditioning Corp., 114 E. Price St., Linden, N.J.
Wm. W. Nugent & Co., Inc., 410 N. Hertitage Ave., Chicago 22, Ill.
Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill.

FILTERS, MAGNETIC

Stearns Magnetic Mfg. Co., 667 S. 28 St., Milwaukee 4, Wis.

FILTERS, OIL

Air-Maze Corp., 5200 Harvard Ave., Cleveland 5, O.
Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
Bowser, Inc., 1302 E. Creighton Ave., Ft. Wayne 2, Ind.
Burt Mfg. Co., 926 S. High St., Akron 11, O.
Cochrane Corp., 17th St., below Allegheny Ave., Phila. 32, Pa.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Honan-Crane Corp., Div. of Houdaille-Hershey Corp., 64 Indiana Ave., Lebanon, Ind.
Wm. W. Nugent & Co., Inc., 410 N. Hermitage Ave., Chicago 22, Ill.
Rega Mfg. Co., 79 Mt. Hope Ave., Rochester, N.Y.

FILTERS, REFRIGERANT (See also STRAINERS)

Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. [illegible])*
Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Henry Valve Co., Melrose Park, Ill. *(p. 104)*

Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 189)*
Michigan Wire Cloth Co., 2098 Howard St., Detroit 16, Mich.
Mueller Brass Co., Port Huron, Mich.
Refrigerating Specialities Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Remco, Inc., 49th St. & A.V.R.R., Pittsburgh 1, Pa.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill.

FILTERS, WATER

American Water Softener Co., S.E. Corner 4th & Lehigh Ave., Phila. 33, Pa.
E. W. Bacharach & Co., 800 Rialto Bldg., Kansas City 6, Mo.
Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Buhring Water Purifying Co., P.O. Box 155, Atlantic Highlands, N.J.
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Dollinger Corp., 1 Centre Park, Rochester 3, N Y.
Elgin Softener Corp., Elgin, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Infilco, Inc., 325 W. 25th St., Chicago 16, Ill.
Liquid Conditioning Corp., 114 E. Price St., Linden, N.J.
Loomis-Manning Filter Co., Schuylkill Ave., below Reed St., Phila. 46, Pa.
Wm. W. Nugent & Co., Inc., 410 N. Hermitage Ave., Chicago 22, Ill.
Oshkosh Filter & Softener Co., 51 Ceape St., Oshkosh, Wis.
Puro Filter Corp. of America, 440 Lafayette St., N.Y.C. 3
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Scottdale Ozone Co., Scottdale, Pa.
Sunroc Refrigeration Co., Glen Riddle, Pa. *(p. 91)*
Waters Filter & Cooler Co., 10 W. 29th St., N.Y.C. 1

FINISHES

American Chemical Paint Co., Ambler, Pa. *(p. 41)*
Arco Co., 7301 Bessemer Ave., Cleveland 4, O.
Baer Bros., 438 W. 37th St., N.Y.C. 18
Brooklyn Varnish Mfg. Co., Inc., 50 Jay St., Brooklyn 1, N.Y.
Chicago Vitreous Enamel Product Co., 1407 S. 55th Court, Cicero 50, Ill.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Ernecke & Salmstein Co., 1611 N. Sheffield Ave., Chicago 14, Ill.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Grand Rapids Varnish Corp., 1350 Steele Ave., S.W., Grand Rapids 2, Mich.
Alfred Hague & Co., 227-34th St., Brooklyn 32, N.Y.
A. C. Horn Co., Inc., 43-36-10th St., Long Island City, N.Y.
Inertol Co., Inc., 470 Frelinghuysen Ave., Newark 5, N.J.
Interchemical Corp., 57 State St., Newark, N.J.
Jones-Dabney Co., Div. of Devoe & Raynolds, 1481 S. 11th St., Louisville 8, Ky.
Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
National Lead Co., 111 Broadway, N.Y.C. 6
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O
O'Neil Duro Co., P.O. Box 1166, Milwaukee 1, Wis.
Paint-Point Products Co., Inc., 99 S. 6th St., Brooklyn 11, N.Y
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Reilly Tar & Chemical Corp., 1615 Merchants Bank Bldg. Indpls. 4, Ind.
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
L. Sonneborn Sons, Inc., 88 Lexington Ave., N.Y.C. 16
Standard Varnish Wks., 2600 Richmond Terrace, Staten Island 3, N.Y.
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
United Chromium, Inc., 51 E. 42nd St., N.Y.C. 17
U. S. Gutta Percha Paint Co., Dudley & Eddy Sts., Providence 1, R.I.
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J
Zapon Div., Atlas Powder Co., Stamford, Ct.

FINISHES, INSULATION

Arco Co., 7301 Bessemer Ave., Cleveland 4, O.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Eagle-Picher Sales Co, American Bldg., Cin'ti. 1, O.
Jones-Dabney Co., Div. of Devoe & Raynolds, 1481 S 11th St., Louisville 8, Ky.
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Standard Varnish Wks., 2600 Richmond Terrace, Staten Island 3, N.Y.
United Chromium, Inc., 51 E. 42nd St., N.Y.C. 17
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.

FINNED COILS

(A—Ammonia; B—Other refrigerants)

(A,B) Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
(A,B) Aerofin Corp., 410 Geddes St., Syracuse, N.Y. *(p. 11)*
(B) American Coils Co., 25 Lexington St., Newark 5, N.J.
(A,B) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Brown Fintube Co., Elyria, O.
(A,B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
(A,B) California Steel Products Co., Barrett & "A" St., Richmond, Cal.
(A) Cyclops Iron Wks., 837 Folsom St., San Francisco 7, Cal.
(B) **Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J.** *(p. 239)*
(B) Fedders-Quigan Corp., 57 Tonawanda St., Buffalo 7, N.Y.
(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(A) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(B) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
(A,B) General Refrigeration Div., Yates-American Machine Co., Beloit, Wis.
(B) Hastings Air Conditioning Co., Inc., Hastings, Neb.
Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 224)*
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
(A,B) Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
(A,B) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
Larkin Coils 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
(B) Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
(B) McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
(B) Melco Mfg. Co., Inc., Grand Ave., Ridgefield, N.J.
(A,B) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(B) Reese & Long Refrigeration Products, Inc., 408 E. 25th St., N.Y.C. 10
(A,B) Refrigeration Appliances, Inc., 917 W. Lake St Chicago 7, Ill.

(A,B) **Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O.** *(p. 228)*
Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
(A,B) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(B) Rogers Air Conditioning Div., 724 Garrison Ave., N.Y.C. 59
(B) Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
(B) Standard Refrigeration Co., 232 S. Hoyne Ave., Chicago 20, Ill.
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona Cal.
(B) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(A,B) Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*

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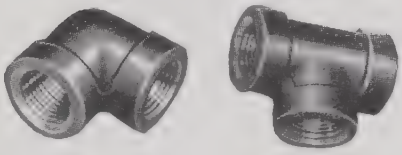

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(B) Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 163)*

FINNED TUBING

Aerofin Corp., 410 Geddes St., Syracuse N.Y. *(p. 11)*
Amercan Coils Co., 25 Lexington St., Newark 5, N.J.
Brown Fintube Co. Elyria, O.
Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
Carpenter Steel Co., Reading, Pa.
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
G & O Mfg. Co., 138 Winchester Ave., New Haven 8, Ct.
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
David E. Kennedy, Inc., 58-2nd Ave., Brooklyn 15, N.Y.
McCord Corp., 2587 E. Grand Blvd., Detroit, 11, Mich. *(p. 107)*
Melco Mfg. Co., Inc., Grand Ave., Ridgefield, N.J.
Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1411 Central Ave., Detroit 9, Mich.

FISH REFRIGERATORS (See DISPLAY CASES)

FITTINGS (See particular type)

FITTINGS, AMMONIA

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Henry Valve Co., Melrose Park, Ill. *(p. 106)*
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Vibraseal Corp., 2832 E. Grand Blvd., Detroit 11, Mich.
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

FITTINGS, BRAZING & SOLDERING

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Grabler Mfg. Co., 6565 Broadway, Cleveland 5, O.

(Continued)

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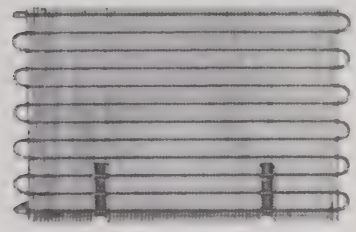

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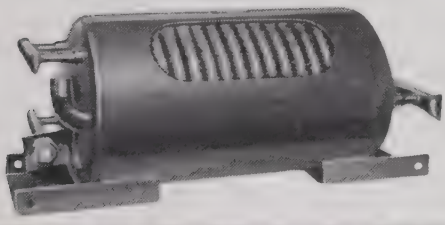
Round continuous tube, plate or fin type condensers for domestic refrigeration.



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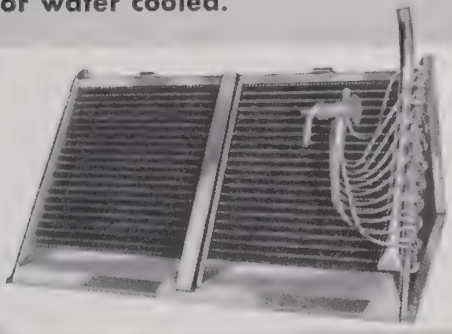
Condensers for commercial refrigeration and air conditioning, air or water cooled.



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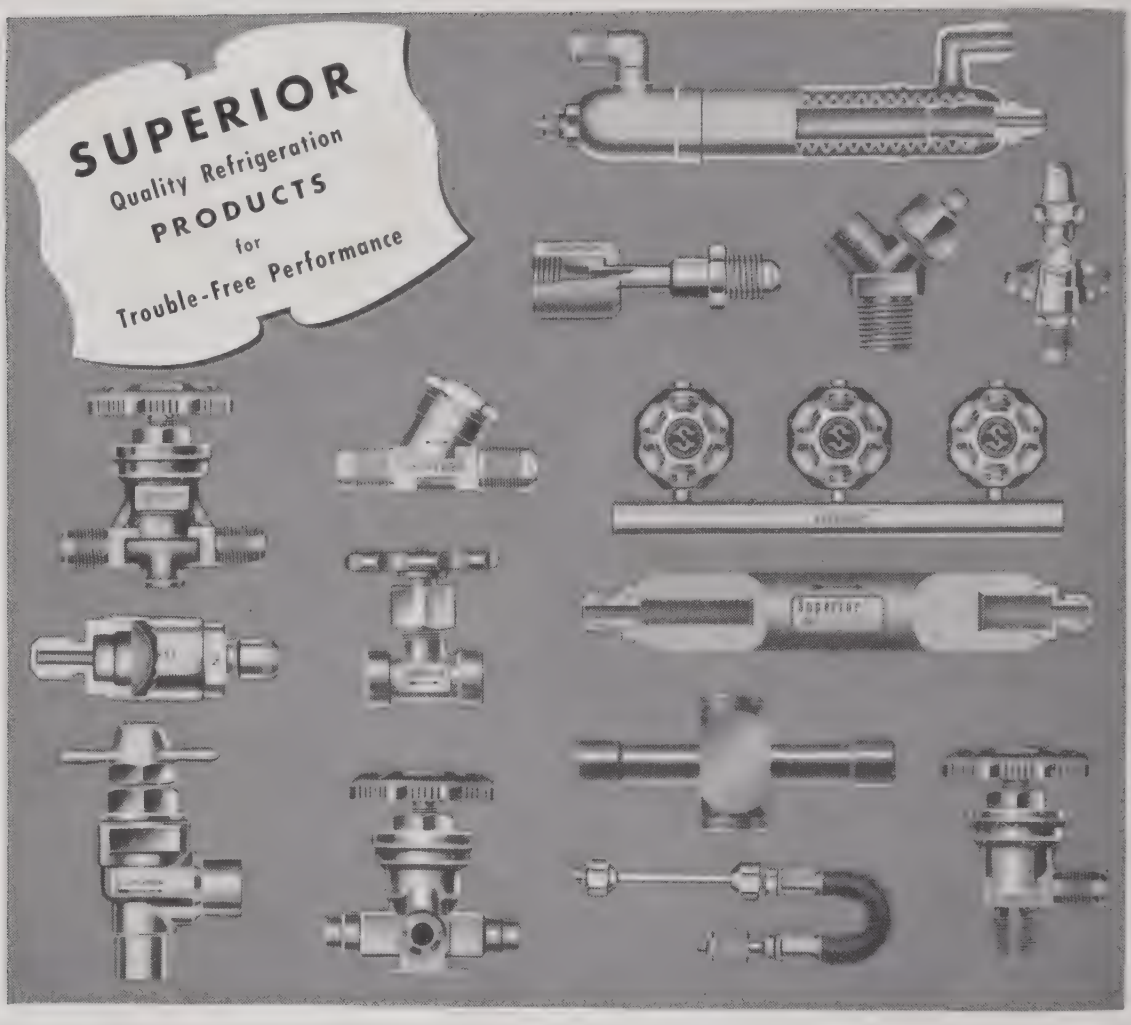

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In all refrigeration and air-conditioning systems, the SUPERIOR line includes the famous Diaphragm Packless Line valves; globe, charging, purging or drain valves; hand expansion valves; pressure cup (wing cap) globe valves; packed line, angle, compressor and cylinder valves; check valves; liquid indicators and sight glasses; quick couplers; gauge manifolds; economizers (heat-exchangers) and manifolds; dehydrators and filters; rapid-chargers (refrigerant transfer systems); charging hose; and SAE flare fittings.

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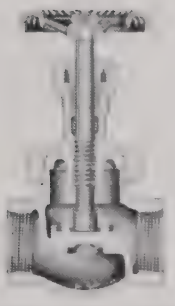
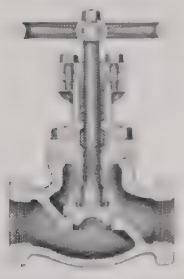


Brass Globe Valves with Plug Type Disc



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Forged Steel Socket-Welding Fittings

FITTINGS, BRAZING & SOLDERING (Continued)

Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Charles W. Krieg Co., 48 Dickerson St., Newark 4, N.J.
Mueller Brass Co., Port Huron, Mich.
National Lead Co., 111 Broadway, N.Y.C. 6
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 113)*

FITTINGS, CAPILLARY TUBE

Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill.

FITTINGS FOR CARBON DIOXIDE

Frick Co., Waynesboro, Pa. *(p. 47)*
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Vibraseal Corp., 2832 E. Grand Blvd., Detroit 11, Mich.
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

FITTINGS, CAST IRON

James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Grabler Mfg. Co., 6565 Broadway, Cleveland 5, O.
Jarecki Mfg. Co., 1345 W. 12th St., Erie, Pa.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Ravena Iron Co., 100 Main St., Ravena, N.Y.
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 4
J. A. Zurn Mfg. Co., Erie, Pa.

FITTINGS, COMPRESSION (See also FITTINGS, FLARE)

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Commonwealth Brass Corp., 5835 Commonwealth Ave., Detroit 8, Mich.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Wm. W. Nugent & Co., Inc., 410 N. Hermitage Ave., Chicago 22, Ill.
Packless Metal Products Corp., 31 Winthrop Ave., New Rochelle, N.Y.
Vibraseal Corp., 2832 E. Grand Blvd., Detroit 11, Mich.

FITTINGS, DUCT (See DUCT ACCESSORIES)

FITTINGS, ELECTRIC CONDUIT (See also CLAMPS, CONDUIT & CABLE)

General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Minerallac Elec. Co., 25 N. Peoria St., Chicago 7, Ill.
National Elec. Products Corp., Fulton Bldg., Pittsburgh 22, Pa.
Pyle-Nat'l. Co., 1371 W. 37th St., Chicago 9, Ill. *(p. 17)*

FITTINGS, FLARE, REFRIGERANT

Brockway Co., 361 Church St., Naugatuck, Ct.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Commonwealth Brass Corp., 5835 Commonwealth Ave., Detroit 8, Mich.
Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 51, Ill. *(p. 55)*
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Imperial Brass Mfg. Co., 437 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Mueller Brass Co., Port Huron, Mich.
Packless Metal Products Corp., 31 Winthrop Ave., New Rochelle, N.Y.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

FITTINGS, FLARE INVERTED

Brockway Co., 361 Church St., Naugatuck, Ct.
Commonwealth Brass Corp., 5835 Commonwealth Ave., Detroit 8, Mich.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Imperial Brass Mfg. Co., 537 S. Racine Ave. Chicago 7, Ill. *(p. 111)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

FITTINGS, FLANGED

James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. *(p. 47)*
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Jarecki Mfg. Co., 1345 W. 12th St., Erie, Pa.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 4
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
York Corp., York, Pa. *(p. 163)*

FITTINGS, FORGED

Commonwealth Brass Corp., 5835 Commonwealth Ave., Detroit 8, Mich.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Henry Valve Co., Melrose Park, Ill. *(p. 106)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
York Corp., York, Pa. *(p. 163)*

FITTINGS, HARD RUBBER

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Luzerne Rubber Co., Trenton 9, N.J.
National Motor Bearing Co., Inc., Redwood City, Cal.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

FAN COIL UNITS (See AIR CONDITIONING FAN COIL UNITS)

FITTINGS, HOSE (See also HOSE CLAMPS)

Aeroquip Corp., 300 S. East Ave., Jackson, Mich. *(p. 83)*
Aircraft Standard Parts Co., Inc., 1711-19th Ave., Rockford, Ill.
Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.

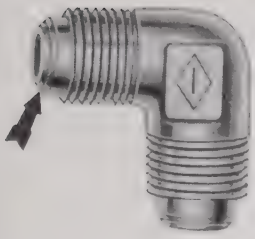
(Continued)

The Imperial Brass Mfg. Company

537 So. Racine Ave., Chicago 7, Ill.

Fittings • Valves • Driers • Filters • Floats • Charging Lines
Tools for: Cutting • Flaring • Bending • Pinch-off • Swedging

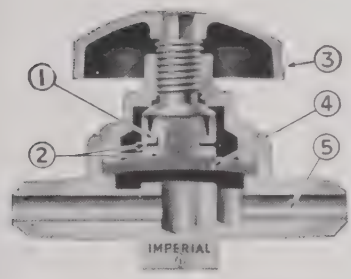
TRIPLE-SEAL FLARED TUBE FITTINGS



Imperial Fittings give extra protection against leakage because a groove in the seat of sizes 3/8" and larger provides a self-seating joint that will remain leak-proof even though the face of the seat may be nicked or marred.

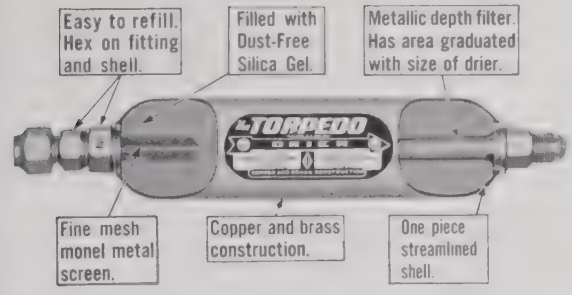
Nuts, tees and elbows are made from brass forgings. This assures a fine grain structure, freedom from internal stress, high tensile strength and the elimination of seepage, season cracking or splitting. Elbows and tees have flats for wrench grip. Extra length pipe threads.

DIASEAL VALVE



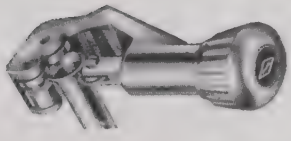
A better, more dependable diaphragm valve for refrigeration and air conditioning use. Offers many outstanding advantages: (1) No Springs—direct lift provides positive control with flow in either direction. (2) Only Two Moving Parts—simple construction assures greater dependability. (3) Easy Finger-Tip Action—quick, sure opening and closing with less than two turns of handle. (4) Long Life Diaphragm is impervious to all common refrigerants. Has withstood over 1,000,000 openings and closings in tests under refrigerant pressure. (5) Inlet and Outlet Ports in line. Simplifies installation.

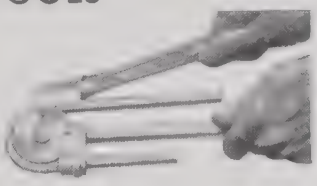
TORPEDO DRIER



Designed for greater efficiency and faster drying action. Note these features: (1) One piece, streamlined copper shell which provides greater strength, lighter weight, fewer joints and less chance for leakage; (2) charged with new Dust-Free Silica Gel; (3) greater filtering area on all sizes provided by finger type metallic depth filter; (4) interchangeable tubing connections on larger sizes; (5) easy to refill.

IMPERIAL TUBE WORKING TOOLS





Good tools are essential to making first class tubing connections. Imperial tube working tools have proven their quality and ease of operation.

The line includes tube cutters, flaring tools, tube benders, reamers, and swedging tools for working with copper, brass, aluminum and thin-wall steel tubing.

Illustrated above are 274-F Tube Cutter for 1/8" to 1" O.D. tubing; 195-F Flaring Tool for 1/4" to 5/8" O.D. Tubing; and 364-F Tube Bender for 1/8" to 3/4" O.D. Tubing. (Individual bender for each size.) Many others also available.

FITTINGS, HOSE (Continued)

DeVilbiss Co., 300 Phillips Ave., Toledo 1, O.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Marman Products Co., Inc., 940 Redondo, Inglewood Cal.
Tinnerman Products, Inc., 2038 Fulton Rd., Cleveland 31, O.
John M. Watt's Sons, 112 Walnut St., Phila., Pa.

FITTINGS, NON-FERROUS

Brockway Co., 361 Church St., Naugatuck, Ct.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Kennedy Valve Mfg. Co., Elmira, N.Y.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
York Corp., York, Pa. *(p. 163)*

FITTINGS, PIPE (See also particular fitting)

Alloy Steel Products Co., Inc., 1508 W. Elizabeth Ave., Linden, N.J.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Brockway Co., 361 Church St., Naugatuck, Ct.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Commonwealth Brass Corp., 5835 Commonwealth Ave., Detroit 8, Mich. (Brass only)
Corley Co., Inc., 629 Grove St., Jersey City 2, N.J.
Crane Co., 836 Michigan Ave., Chicago 5, Ill *(p. 109)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
Dresser Mfg. Div., Dresser Industries, Inc., 490 Fisher Ave., Bradford, Pa.
Duriron Co., Inc., Dayton 1, O.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. *(p. 47)*
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Grabler Mfg. Co., 6565 Broadway, Cleveland 5, O.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Gustin-Bacon Mfg. Co., Kansas City 7, Mo.
Hasco Valve & Machinery Co., 1819 W. St. Paul Ave., Milwaukee 3, Wis.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Henry Valve Co., Melrose Park, Ill. *(p. 106)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Jarecki Mfg. Co., 1345 W. 12th St., Erie, Pa.
Jefferson Union Co., Inc., 71 Gooding St., Lockport, N.Y.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Charles W. Krieg Co., 48 Dickerson St., Newark 4, N.J.
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
Minerallac Elec. Co., 25 N. Peoria St., Chicago 7, Ill.
A. B. Murray Co., Inc , 604 Green Lane, Elizabeth, N.J.
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Vibraseal Corp., 2832 E. Grand Blvd., Detroit 11, Mich.
Victaulic Co. of America, 30 Rockefeller Plaza, N Y.C. 20
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 4
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*
J. A. Zurn Mfg. Co., Erie, Pa.

FITTINGS, SHEET METAL (See DUCT ACCESSORIES; also DUCTWORK, PREFABRICATED

FITTINGS, WELDING (See also particular fitting)

Alloy Products Corp., 1045 Perkins Ave., Waukesha, Wis.
Bonney Forge & Tool Wks., Allentown, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Frick Co., Waynesboro, Pa. *(p. 47)*
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 113)*

FLAKE ICE MACHINERY (See ICE MAKING MACHINES)

FLANGE ADAPTERS (See ADAPTER FLANGES)

FLANGES, PIPE (See also FITTINGS)

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Harrisburg Steel Corp., P.O. Box 329, Harrisburg, Pa.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Ravena Iron Co., 100 Main St., Ravena, N.Y.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Ross Sprinkler Co., 34 Roberts St., Pasadena 3, Cal.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 3
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
York Corp., York, Pa. *(p. 163)*

FLEXIBLE CONNECTIONS (See also CHARGING LINES; also TUBING, FLEXIBLE; also SWING JOINTS)

American Brass Co., Waterbury 88, Ct.
Brockway Co., 361 Church St., Naugatuck, Ct.

(Continued)

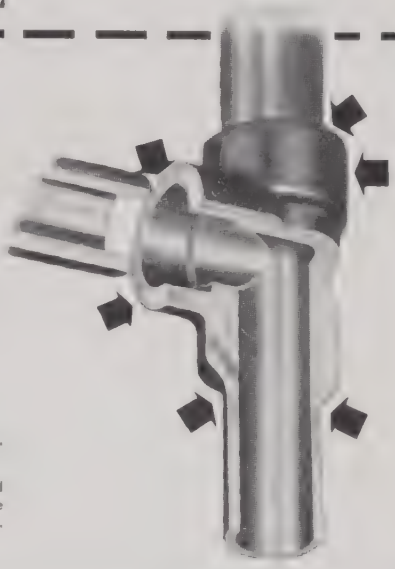
WATSON W-S STILLMAN

a complete line of forged steel pipe fittings for GENERAL REFRIGERATION SERVICE

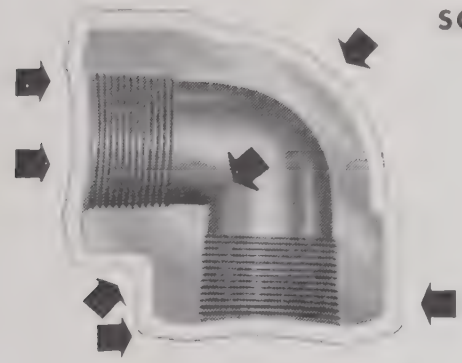
• Both types of Watson-Stillman fittings—screw-end and socket-welding—cut maintenance costs because they're built right to fit tight *permanently*. Watson-Stillman offers a full line of these trouble-free fittings machined from solid drop forgings in either Carbon, Carbon-Molybdenum, Chromium-Molybdenum or various types of Stainless Steel.

SOCKET WELDING FITTINGS ARE EASY TO INSTALL, LONG-LASTING

- Just slip fitting over end of pipe and weld.
- Deep sockets support and align pipes—eliminate tack welding, the use of special fixtures, accurate measuring and cutting of pipe. Ample come-and-go.
- Position of weld prevents "icicles". Wall sections 1¼ times nominal pipe thickness permit proper heat penetration, correctly proportioned fillet weld.
- Inside diameter of fitting matches inside of pipe for smooth flow.
- No weak sections—long bands extend well beyond bottom of socket, provide reinforcement at points of strain.
- Joints are extra-strong and leak-proof, unaffected by vibration, distortion or shock. You can put these fittings in and forget them.



All are available for use with Schedule 40—Standard Weight Pipe, Schedule 80—Extra Strong Pipe, Schedule 160 and Double Extra Strong Pipe. Sizes ⅛" to 4" I. P. S.

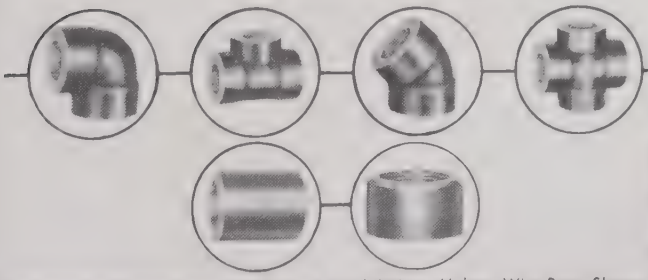


SCREW-END TYPES FOR ALL PRESSURE AND TEMPERATURE SERVICES

- Threads are long, accurately cut.
- Designed for perfect alignment.
- Uniform wall thickness.
- Unrestricted flow.
- Controlled chemical and physical properties.
- Long low bands afford better wrench grip.
- Rugged—bored from solid drop forgings.

W-S Screw End Fittings are available in three classes—2000, 3000 and 6000 pound. Sizes ⅛" to 4" I. P. S.

Sold Through Leading Distributors

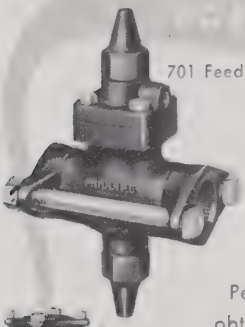


WATSON-STILLMAN
DISTRIBUTOR PRODUCTS DIVISION

1848 Centennial 1948

Designers and Manufacturers of Forged Steel Fittings, Valves, Wire Rope Shears, Hand Pumps, Jacks, Pipe Benders and Hydraulic Equipment

PHILLIPS FLOAT CONTROLS

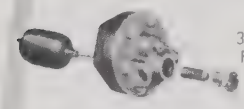


701 Feed Valve

Capacities: Up to 2000 Tons.

The No. 701 feed valve operates by means of the varying pressure in a power cylinder above the piston. Perfect modulated action obtained on flooded systems when used in conjunction with either No. 300 HP or No. 100 VP float controls. No. 701 Feed Valve may also be used with thermal expansion or magnetic solenoid.

100 VP Pilot Float Valve
Welded Steel Chamber



300 HP Pilot Float Valve

The No. 101 Remote Feed Valve on the Right permits location of float and chamber outside of evaporator. Adjustable liquid level.

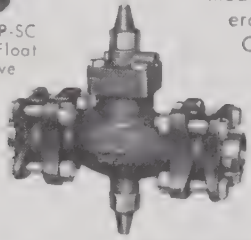
101 Remote Feed Valve With Float Chamber

250 VP-SC Pilot Float Valve

FOR "FREON"

The No. 701 Feed Valve is a pilot controlled piston valve actuated by pressures in the power cylinder. Because of its quick and responsive action, the 205 VP-SC Pilot is recommended for ideal modulated action of refrigerant.

Capacities: Up to 750 Tons.



701 "Freon" Feed Valve

Write for catalog

H. A. PHILLIPS & CO.
3255 WEST CARROLL AVENUE
CHICAGO 24, ILLINOIS

FLEXIBLE CONNECTIONS (Continued)

Chicago Metal Hose Corp., Maywood, Ill.
Eclipse Aviation Metal Hose Dept., Div. of Bendix Aviation Corp., Phila. 44, Pa.
Henry Valve Co., Melrose Park, Ill. *(p. 106)*
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
Packless Metal Products Corp., 31 Winthrop Ave., New Rochelle, N.Y.
Seamlex Co., Inc., 4123-24th St., Long Island City 1, N.Y. *(p. 218)*
York Corp., York, Pa. *(p. 163)*
Zallea Bros. & Johnson, 9th St. & P.R.R., Wilmington 99, Del.

FLEXIBLE COUPLINGS (See also SHAFT COUPLINGS)

Baldwin Duckworth Div., Chain Belt. Co., 369 Plainfield St., Springfield, Mass.
Brockway Co., 361 Church St., Naugatuck, Ct.
Certified Flexible Couplings, 369 Lexington Ave., N.Y.C. 17
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind. (Shaft)
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa. (For power Transmission)
Marman Products Co., Inc., 940 Redondo, Inglewood, Cal.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Phila. 20, Pa.
Ramsey Chain Co., Inc., 900 Broadway, Albany 1, N.Y.
Water Cooling Corp., 71 Nassau St., N.Y.C. 7 (Pipe)

FLOAT BALLS

Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Arthur Harris & Co., 210 N. Aberdeen St., Chicago 7, Ill.
Industrial Wire Cloth Products Corp., Wayne, Mich.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)

FLOATS, LIQUID LEVEL CONTROL (See also FLOAT SWITCHES; also HIGHSIDE FLOATS; also LOWSIDE FLOATS; also CONTROLS, LIQUID LEVEL)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Armstrong Machine Wks., 846 Maple St., Three Rivers, Mich.
Ernst Water Column & Gage Co., 250 S. Livingston, Livingston, N.J.
Frick Co., Waynesboro, Pa. *(p. 47)*
Henry Valve Co., Melrose Park, Ill. *(p. 106)*
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p.114)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Reichert Float & Mfg. Co., 2238 Smead Ave., Toledo 6, O.
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
York Corp., York, Pa. *(p. 163)*

FLOAT VALVES, REFRIGERANT (See HIGHSIDE FLOATS; also LOWSIDE FLOATS)

FLOAT VALVES, WATER

Thomas Beckett & Co., Inc., 2118 Griffin St., Dallas 2, Tex.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.

Robert Mfg. Co., 3417 Crenshaw Blvd., Los Angeles 16, Cal.
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.
Trane Co., La Crosse, Wis. *(p. 14)*
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.

FLOAT SWITCHES (See also HIGHSIDE FLOATS; also LOWSIDE FLOATS)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Allen-Bradley Co., Milwaukee 4, Wis.
Automatic Control Co., 1005 University Ave., St. Paul 4, Minn.
Fischer & Porter Co., Hatboro, Pa. (Flow Alarm)
Frick Co., Waynesboro, Pa. *(p. 47)*
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hancock Valve Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Henry Valve Co., Melrose Park, Ill. *(p. 106)*
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Soreng Mfg. Corp., 1901 Clybourn Ave., Chicago 14, Ill.

FLORISTS REFRIGERATORS (See REFRIGERATORS, FLORISTS)

FLOW CONTROLS (See CONTROLLERS, FLOW)

FLOW RATE INDICATORS

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Barton Instrument Co., 3500 Union Pacific Ave., Los Angeles 23, Cal.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Fischer & Porter Co., Hatboro, Pa.
Gas & Oil Industry Labs., Inc., 4 Paine Ave., Irvington 11 N.J.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I
Hays Corp., Michigan City, Ind.
Infilco, Inc., 325 W. 25th St., Chicago 16, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Wm. W. Nugent & Co., Inc., 410 N. Hermitage Ave., Chicago 22, Ill.
Taylor Instrument Cos. 95 Ames St., Rochester 1, N.Y. *(p. 183)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.

FLUX, BRAZING, SOLDERING, WELDING, etc.

Alpha Metals, Inc., 363 Hudson Ave., Brooklyn 1, N.Y.
American Chemical Paint Co., Ambler, Pa. *(p. 41)*
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
M. W. Dunton Co., 670 Eddy St., Providence 3, R.I.
Handy & Harman, 82 Fulton St., N.Y.C. 7
Charles W. Krieg Co., 48 Dickerson St., Newark 4, N.J.
Marquette Mfg. Co., Inc., 307 E. Hennepin, Minneapolis 14, Minn.
National Lead Co., 111 Broadway, N.Y.C. 6
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Pennsylvania Salt Mfg. Co., 1000 Widener Bldg., Phila. 7, Pa. (Aluminum)
Ruby Chemical Co., 68 McDowell St., Columbus 8, O.
Superior Flux Co., 913 Public Square Bldg., Cleveland 13, O.

FLYWHEELS

Linderme Machine & Tool Co., 12241 Coyle Ave., Detroit 27, Mich.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.

FOAM RUBBER (See RUBBER PRODUCTS, CELLULAR)

FOIL, METALLIC

Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
Alpha Metals, Inc., 363 Hudson Ave., Brooklyn 1, N.Y (Lead)
Aluminum Co. of America, Pittsburgh 19, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Johnston Tin Foil & Metal Co., 6106 S. Broadway, St. Louis 11, Mo.

FOOT VALVES

James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Deming Co., 884 S. Broadway, Salem, O.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
Jarecki Mfg. Co., 1345 W. 12th St., Erie, Pa.
Jenkins Bros., 80 White St., N.Y.C. 13
Ravena Iron Co., 100 Main St., Ravena, N.Y.
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Swaby Mfg. Co., 2330 W. Cermak Rd., Chicago 8, Ill.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
White Flomatic Corp., P.O. Box 267, Hoosick Falls, N.Y.
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 12, Pa.

FORECOOLERS (See WATER COOLERS)

FORGINGS

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Atlas Drop Forge Co., 209 W. Mt. Hope Ave., Lansing 2, Mich.
Bethlehem Steel Co., Bethlehem, Pa.
Bonney Forge & Tool Wks., Allentown, Pa.
Cleveland Hardware & Forging Co., 3270 E. 79th St., Cleveland 4, O.
Columbus Bolt Wks. Co., 291 Marconi Blvd., Columbus 16, O.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Duff-Norton Mfg. Co., Box 1889, Pittsburgh 30, Pa.
Endicott Forging & Mfg. Co., Inc., 1905 North St., Endicott, N.Y.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Harrisburg Steel Corp., P.O. Box 329, Harrisburg, Pa.
Interstate Drop Forge Co., 4035 N. 27th St., Milwaukee, Wis.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O
McInness Steel Co., Corry, Pa.
Ohio Forge & Machine Corp., 3010 Woodhill Rd., Cleveland 4, O.
Pratt & Letchworth Co, 189 Tonawanda St., Buffalo 7, N.Y. (Steel)
Transue & Williams Steel Forging Corp., Alliance, O.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
York Corp., York, Pa. *(p. 163)*

FORGINGS, NON-FERROUS

Aluminum Co. of America, Pittsburgh 19, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Endicott Forging & Mfg. Co., Inc., 1905 North St., Endicott, N.Y.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Mueller Brass Co., Port Huron, Mich. (Aluminum)
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Titan Metal Mfg. Co., Bellefonte, Pa.

FOUL GAS EXTRACTORS (See PURGERS)

FRAMES, DISPLAY CASE, etc. (See DISPLAY CASE DOORS)

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FREEZER DOORS (See COLD STORAGE DOORS)

FREON REFRIGERANTS

Ansul Chemical Co., Marinette, Wis. (Sales Agents) *(p. 175)*
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del. (Sales Agents) *(p. 153)*
Eston Chemicas, Inc., 3100 E. 26th St., Los Angeles 23, Cal. (Sales Agents) *(p. 152)*
Kinetic Chemicals, Inc., 10th & Market Sts., Wilmington, Del. *(p. 117)*
Virginia Smelting Co., W. Norfolk, Va. (Sales Agents) *(p. 177)*

FRICTION TAPE

Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Johns-Manville, 22. E 40th St., N.Y.C. 16
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc, 7800 S. Woodlawn Ave., Chicago 19, Ill.

FROZEN FOOD CABINETS (See HOME & FARM FREEZERS)

FUSES

Bussmann Mfg. Co., University & Jefferson Sts., St. Louis 7, Mo.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Jefferson Elec. Co., 25 Ave. & Madison St., Bellwood, Ill.
Littlefuse, Inc., 4757 Ravenswood Ave., Chicago 40, Ill.

FUSIBLE PLUGS

Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Mueller Brass Co., Port Huron, Mich.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
Standard Valve Mfg. Co., 817 Albany St., Boston 19, Mass.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

GALVANIZING

W. Ames & Co., Jersey City, N.J.
Joseph P. Cattie & Bros., Gaul & Letterly Sts., Phila. 25, Pa.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo. (Bolts, etc., & Customers' property)

GAS COOLERS

Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

GAS MASKS

Acme Protection & Equip. Co., 3616 Liberty Ave., Pittsburgh, Pa.
Chicago Eye Shield Co., 2300 Warren Blvd., Chicago 12, Ill.
General Scientific Equip. Co., 27th & Huntingdon St, Phila. 32, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Mine Safety Appliances Co., Braddock, Thomas & Meade Sts., Pittsburgh, Pa.
Pulmosan Safety Equip. Corp., 176 Johnson St., Brooklyn 1, N.Y.
York Corp., York, Pa. *(p. 163)*

GASES, RARE

Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17 *(p. 197)*
Matheson Co., Inc., P.O. Box 85, E. Rutherford, N.J.

GASKET, DOOR (See DOOR GASKET)

GASKETS & GASKET MATERIAL

Akron Metallic Gasket Co., P.O. Box 911, Akron, O.
Anchor Packing Co., 401 N Broad St., Phila. 8, Pa.
Armstrong Cork Co., Lancaster, Pa. *(p. 136)*
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Belmont Packing & Rubber Co., Butler & Sepviva Sts., Phila. 7, Pa.
Buffalo Weaving & Belting Co., 262 Chandler St., Buffalo 7, N.Y.
Chamberlin Co. of America, 1254 LaBrosse St., Detroit 26, Mich.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Chicago-Wilcox Mfg. Co., 7701 Avalon Ave., Chicago 19, Ill.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Endura Mfg. Corp., 45 N. 4th St., Quakertown, Pa.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Excelsior Leather Washer Mfg. Co., Inc., Rockford, Ill. (Leather)
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
Goetze Branch, Johns-Manville Products Corp., Allen Ave., New Brunswick, N.J.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
W. F. Houghton & Co.; 303 W. Lehigh Ave., Phila. 33, Pa.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Inland Mfg. Div., Gen'l. Motors Corp., Dayton 1, O.
Janos Asbestos Co., 39 Cortlandt St., N.Y.C. 7
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Johnston Tin Foil & Metal Co., 6106 S. Broadway, St Louis 11, Mo.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Klingerit, Inc., 16 Hudson St., N.Y.C. 13

(Continued)

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GASKETS & GASKET MATERIALS (Continued)

Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
McCord Corp., 2857 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
Metallo Gasket Co., 10 Bethany St., New Brunswick, N.J.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Phoenix Specialty Mfg., Co. Inc., 80 Albany Ave., Freeport, Long Island, N.Y.
Horace G. Preston Co., 2581 Beecher Ave., Detroit 16, Mich.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Rhopac, Inc., 168 N. Clinton St., Chicago 6, Ill.
Smooth-On Mfg. Co., 572 Communipaw Ave., Jersey City 4, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.
Wilmington Fiber Specialty Co., P.O. Drawer 1028, Wilmington 99, Del.
Wolverine Fabricating & Mfg. Co., Inc., Princess St. & M.C.R.R., Inkster, Mich.
York Corp., York, Pa. *(p. 163)*

GAUGE GLASSES

Semon Bache & Co., 636 Greenwich St., N.Y.C. 14
L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.
Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Corning Glass Wks., Corning, N.Y.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Fischer & Porter Co., Hatboro, Pa.
Meade Goodloe, 406 S. Main St., Los Angeles, Cal.
Henry Valve Co., 3260 W. Grand Ave., Chicago 51, Ill. *(p. 119)*
Paul B. Huyette Co., Inc., 401 N. Broad St., Phila. 8, Pa.
Jerguson Gage & Valve, 87 Fellsway, Comerville, Mass.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nathan Mfg. Co., 416 E. 106th St., N.Y.C. 29
Oil Rite Corp., 3466 S. 13th St., Milwaukee 7, Wis.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y.
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 12, Pa.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

GAUGE PULSATION ABSORBERS

Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
Meriam Instrument Co., 10920 Madison Ave., Cleveland 2, O.
Mueller Brass Co., Port Huron, Mich.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.

GAUGES (See particular type)

GAUGES, AIR FILTER

Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Hays Corp., Michigan City, Ind.
Owens-Corning Fiberglas Corp., 2012 Nicholas Bldg., Toledo 1, O. *(p. 18)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.

GAUGES, ALARM

Ashcroft Gauge Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Ashton Valve Co., 161-1st St., Cambridge, 42, Mass *(p. 168)*
Automatic Temperature Control Co., Inc , 34 E. Logan St., Phila., Pa.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Marshalltown Mfg. Co., Marshalltown, Ia.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.

GAUGES, AMMONIA

Ashcroft Gauge Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Jerguson Gage & Valve, 87 Fellsway, Somerville, Mass.
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
James P. Marsh Corp., 2073 Southport Ave., Chicago 14, Ill.
Marshalltown Mfg. Co., Marshalltown, Ia.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

GAUGES, COMPOUND

Ashcroft Gauge Div., Manning, Maxwell & Moore, Inc. Bridgeport 2, Ct.
Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
James P. Marsh Corp., 2073 Southport Ave., Chicago 14 Ill.
Marshalltown Mfg. Co., Marshalltown, Ia.
H. O Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

GAUGES, DIFFERENTIAL

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Barton Instrument Co., 3500 Union Pacific Ave., Los Angeles 23, Cal.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., Div. Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Hays Corp., Michigan City, Ind.
Meriam Instrument Co., 10920 Madison Ave., Cleveland 2, O.
Moeller Instrument Co., Inc., 132nd St., & 89th Ave., Richmond Hill 18, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Vacuum Equip. Div., Distillation Products, Inc., 755 Ridge Rd., W., Rochester 13, N.Y.

GAUGES, DRAFT

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Hays Corp., Michigan City, Ind.
Meriam Instrument Co., 10920 Madison Ave , Cleveland 2, O.
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
U. S Gauge, Div. of American Machine & Metals, Sellersville, Pa.

GAUGES, DUCT PRESSURE

Bailey Meter Co., 1050 Ivanhoe Rd.,Cleveland 10, O.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Hays Corp., Michigan City, Ind.
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd. Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.

GAUGES, FLOW RATE (See FLOW RATE INDICATORS)

GAUGES, FREON

Ashcroft Gauge Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Jerguson Gage & Valve, 87 Fellsway, Somerville, Mass.
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
James P. Marsh Corp., 2073 Southport Ave., Chicago 14, Ill.
Marshalltown Mfg. Co., Marshalltown, Ia.
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

GAUGES, LIQUID LEVEL, TANK, etc. (See also GAUGE GLASSES)

AC Spark Plug Div., Gen'l. Motors Corp., Flint 2 Mich.
Alloy Steel Products Co., Inc., 1508 W. Elizabeth Ave., Linden, N.J. (Stainless Steel)
Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Barton Instrument Co., 3500 Union Pacific Ave., Los Angeles 23, Cal.
L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.
Boston Auto Gage Co., 70 West St., Pittsfield, Mass.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
Fischer & Porter Co., Hatboro, Pa.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Green Bay Paper Products Co., Div. of Bearlee Products Co., 1255 Monroe Ave., Green Bay, Wis. (Also barrel gauges)
Hays Corp. Michigan City, Ind.
Henry Valve Co., Melrose Park, Ill. *(p. 119)*
Paul B. Huyette Co., Inc., 401 N. Broad St., Phila. 8, Pa.
Jerguson Gage & Valve, 87 Fellsway, Somerville, Mass.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Liquidometer Corp., 41-03-36th St., Long Island City 1, N.Y.
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Nathan Mfg. Co., 416 E. 106th St., N.Y.C. 29
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Oceco Div., Johnston & Jennings Co., Addison Rd., Cleveland 4, O.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.
Yarnall-Waring Co., Chestnut Hill, Phila. 18, Pa.

GAUGES, PRESSURE

Ashcroft Gauge Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo
DeVilbiss Co., 300 Phillips Ave., Toledo 1, O.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Hays Corp., Michigan City, Inc.
Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
James P. Marsh Corp., 2073 Southport Ave., Chicago 14 Ill.

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Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

GAUGES, PRESSURE RECORDING (See RECORDERS, PRESSURE)

GAUGES, TEMPERATURE (See also THERMOMETERS, DIAL)

AC Spark Plug Div., Gen'l Motors Corp., Flint 2, Mich.
Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh 8, Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Hays Corp., Michigan City, Ind.
Liquidometer Corp., 41-03-36th St., Long Island City 1, N.Y.
James P. Marsh Corp., 2073 Southport Ave., Chicago 14, Ill.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Pyrometer Instrument Co., 103 Lafayette St., N.Y.C. 13
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

GAUGES, VACUUM

Ashcroft Gauge Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Boston Auto Gauge Co., 70 West St., Pittsfield, Mass.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
General Elec. Co., 1 River Md., Schenectady 5,NY.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Hays Corp., Michigan City, Ind.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
James P. Marsh Corp., 2073 Southport Avenue, Chicago 14, Ill.
Marshalltown Mfg. Co., Marshalltown, Ia.
Meriam Instrument Co., 10920 Madison Ave., Cleveland 2, O.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
F. J. Stokes Machine Co., Tabor Rd., Phila. 20, Pa.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C 12

GEAR MOTORS (See MOTORS, GEARHEAD)

GEARS

Brown & Sharpe Mfg. Co., 235 Promenade St., Providence 1, R.I.
Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Gleason Wks., 1000UniversityAve., Rochester 3, N.Y.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Ohio Forge & Machine Corp., 3010 Woodhill Rd., Cleveland 4, O.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Phila. 20, Pa.
Ramsey Chain Co., Inc., 900 Broadway, Albany 1, N.Y.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

GENERATORS (See ELECTRIC GENERATING PLANTS; also MOTOR GENERATORS)

GLASS

American Window Glass Co., Farmers Bank Bldg., Pittsburgh 22, Pa.
Semon Bache & Co., 636 Greenwich St., N.Y.C. 14
Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
Libbey-Owens-Ford Glass Co., Nicholas Bldg., Toledo 3, O.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.

GLASS BLOCKS

Owens-Illinois Glass Co., Duraglas Bldg., Toledo 1, O.
Pittsburgh Corning Corp., 632 Duquesne Way, Pittsburgh 22, Pa.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Refractory & Insulation Corp., 120 Wall St., N.Y.C. 5

GLASS, INSULATING OR HEAT DEFLECTING (See also GLASS BLOCKS)

Pittsburgh Corning Corp., 632 Duquesne Way, Pittsburgh 22, Pa.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Thermopane Div., Libbey-Owens-Ford Glass Co., Nicholas Bldg., Toledo 3, O.

GLASS, GAUGE (See GAUGE GLASSES)

GLASS FILLERS (See also DRINKING WATER COOLER FITTINGS)

Ebco Mfg. Co., 401 W. Town St., Columbus 8, O. *(p. 90)*
Halsey W. Taylor Co., Warren, O.

GLASS SPECIALTIES

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Semon Bache & Co., 636 Greenwich St., N.Y.C. 14
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila 31, Pa.
Corning Glass Wks., Corning, N.Y.
Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
Fischer & Porter Co., Hatboro, Pa.

GLYCOLS (See DIETHYLENE, ETHYLENE, & PROPYLENE GLYCOL)

GOVERNORS

Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.

GRAPHITE PRODUCTS (See also CARBON PRODUCTS)

Acheson Colloids Corp., Port Huron, Mich.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Graphite Metallizing Corp., 1050 Nepperhan Ave., Yonkers 3, N.Y.
Speer Carbon Co., St. Marys, Pa.
U. S. Graphite Co., Saginaw, Mich.

GREASE CUPS (See also LUBRICATORS)

Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y.

GREASES

Arkansas Fuel Oil Co., Div. of Cities Service Oil Co., Slattery Bldg., Shreveport, La.
Black Bear Co., Inc., 44-23rd St., Long Island City 1, N.Y.
Borne Scrymser Co., P.O. Box 256, Elizabeth, N.J.
Cities Service Oil Co. (Del.), 919 N. Michigan Ave., Chicago, Ill.
Cities Service Oil Co. (Pa.), 60 Wall Tower, N.Y.C. 5
Gulf Oil Corp., Gulf Bldg., Pittsburgh, Pa.
W. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
E. H. Kellogg & Co., 93 Water St., N.Y.C. 5
Lubriplate Div., Fiske Bros. Refining Co., 129 Lockwood St., Newark, N.J
Macmillan Petroleum Corp., 530 W. 6th St., Los Angeles 14, Cal.
New York & New Jersey Lubricant Co., 292 Madison Ave., N.Y.C. 17
Ohio Grease Co., N. Spring St., Loudonville, O.
Shell Oil Co., Inc., 50 W. 50th St., N.Y.C. 20
Standard Oil Co. of California, 225 Bush St., San Francisco 20, Cal.
Sun Oil Co., 1608 Walnut St., Phila. 3, Pa. *(p. 161)*

GRILLES (See also AIR DIFFUSERS; also REGISTERS)

A-J Mfg. Co., 2119 Washington, Kansas City 8, Mo.
Alton Mfg. Co., 1112 Ross Ave., Dallas 2, Tex.
Auer Register Co., 3608 Payne Ave., Cleveland, O.
Barber-Colman Co., Rockford, Ill.
E. K. Campbell Heating Co., 1809 Manchester St., Kansas City 3, Mo.
Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp. Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Diamond Mfg. Co., Wyoming, Pa.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
Lockjoint Wood Products Co., 1721 Mildred Ave., Wichita 7, Kan.
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
Manhattan Perforated Metal Co., Inc., 43-17-37th St., Long Island City 1, N.Y.
Newman Bros., Inc., 666 W. 4th St., Cin'ti. 3, O.
Pyle Nat'l. Co., 1371 W. 37th St., Chicago 9, Ill. *(p. 17)*
Radiant Steel Products Co., 200 Walnut St., Williamsport 28, Pa.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.
Titus Mfg. Corp., Waterloo, Ia.
Tuttle & Bailey, Inc., New Britain, Ct.
U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
United Steel & Wire Co., Battle Creek, Mich.
Wheeling Corrugating Co., Wheeling, W. Va.
Young Regulator Co., 5209 Euclid Ave., Cleveland 3, O.

GROMMETS (See also WASHERS)

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Dryden Rubber Co., 1014 S. Kildare Ave., Chicago 24, Ill.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Sponge Rubber Products Co., 106 Derby Place, Shelton, Ct. *(p. 188)*
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

GUARDS, FAN, MACHINERY, etc.

E. H. Allen Co., 22 Dorrance St., Boston 29, Mass.
Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp., Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Kentucky Metal Products Co., Preston St. & Audubon Park, Louisville 4, Ky.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Union Steel Products Co., 448 Pine St., Albion, Mich.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
United Welding Div., General Machinery Corp., 545 N 3rd St., Hamilton, O.

HAIR FOR INSTRUMENTS

Henry J. Green, 1191 Bedford Ave., Brooklyn 16, N.Y.

HAIR FELT (See INSULATION, HAIR PRODUCTS; also FELT)

HANDLES, VALVE, etc.

Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Walworth Co., 60 E. 42nd St., N.Y.C. 17

HANGERS, PIPE (See also CLAMPS, PIPE)

Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Chicago Expansion Bolt Co., 1338 W. Concord Place, Chicago 22, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Grabler Mfg. Co., 6565 Broadway, Cleveland 5, O.
Minerallac Elec. Co., 25 N. Peoria St., Chicago 7, Ill.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Walworth Co., 60 E. 42nd St., N.Y.C. 17

HARDENING ROOMS (See ICE CREAM HARDENING ROOMS)

HARDWARE, COLD STORAGE DOOR

Arcade Mfg. Div., Rockwell Mfg. Co., 1212 E. Shawnee St., Freeport, Ill.
Butcher Boy Cold Storage Door Co., Div. of Western Fixture & Equip. Co., 170 N. Sangamon St., Chicago 7, Ill.
Chase Industrial Refrigerator Equip. & Engrg. Co., 630 Reading Rd., Reading, Cin'ti. 15, O.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17 *(p 142)*
Fox Co., Fox Lane, Cin'ti. 23, O.
Jamison Cold Storage Door Co., Hagerstown, Md. *(p. 45)*
Kason Hardware Corp., 127 Wallabout St., Brooklyn 6 N.Y.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 16, Ill.
Standard-Keil Hardware Mfg. Co., Inc., 2413 Atlantic Ave., Brooklyn 33, N.Y. *(p. 123)*
Streator Products Corp., 508 N. 8th St., Fairfield, Ia.
York Corp., York, Pa. *(p. 163)*

HARDWARE, REFRIGERATOR

Arcade Mfg. Div., Rockwell Mfg. Co., 1212 E. Shawnee St., Freeport, Ill.
Chase Industrial Refrigerator Equip. & Engrg. Co., 630 Reading Rd., Reading, Cin'ti. 15, O.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich. *(p. 126)*
Kason Hardware Corp., 127 Wallabout St., Brooklyn 6, N.Y.

(Continued)

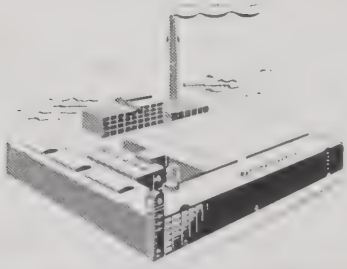
Headquarters—

Refrigerator Hardware

For domestic, commercial and low temperature cabinets

Standard and custom built designs.

SURFACE TYPE HARDWARE
"THRU THE DOOR" HARDWARE
EDGE MOUNTED HARDWARE
PLASTICS
DIE CASTING
SCREWS AND BOLTS



NATIONAL LOCK COMPANY
Refrigerator Hardware Division
ROCKFORD, ILLINOIS

HARDWARE, REFRIGERATOR (Continued)

Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
National Lock Co., 7th St. and 18th Ave., Rockford, Ill. *(p. 122)*
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 16, Ill.
Standard-Keil Hardware Mfg. Co., Inc., 2413 Atlantic Ave., Brooklyn 33, N.Y. *(p. 123)*

HARDWARE SPECIALTIES (See also specific item, such as HOOKS, HINGES, etc.)

Arcade Mfg. Div., Rockwell Mfg. Co., 1212 E. Shawnee St., Freeport, Ill.
Chase Industrial Refrigerator Equip. & Engrg. Co., 630 Reading Rd., Reading, Cin'ti. 15, O.
Croname, Inc., 3701 N. Ravenswood, Chicago 13, Ill.
Fox Co., Fox Lane, Cin'ti. 23, O.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich. *(p.126)*
Kason Hardware Corp., 127 Wallabout St., Brooklyn 6, N.Y.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

HARNESSES, ELECTRICAL

Interstate Mfg. Corp., 133 Sussex Ave., Newark 4, N.J. *(p. 246)*
Riverside Mfg. Co., 200 S. River St., Ypsilanti, Mich. *(p. 247)*

HEADS, TANK, etc. (See TANK HEADS)

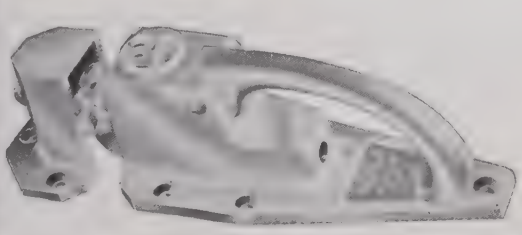
HEAT EXCHANGERS

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
Aerofin Corp., 410 Geddes St., Syracuse, N.Y. *(p. 11)*
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
American Locomotive Co., 30 Church St., N.Y.C.
Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Betz Corp., 445 State St., Hammond, Ind.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Cochrane Corp., 17th St., below Allegheny Ave., Phila. 32, Pa.
Colonial Iron Wks. Co., 17643 St. Clair Ave., Cleveland 10, O.
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill. *(p. 167)*
Downingtown Iron Wks., Downingtown, Pa. *(p. 59)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
Duriron Co., Inc., Dayton 1, O.
W. J. Finnegan Co., 7402 Santa Monica Blvd., Los Angeles 46, Cal. *(p. 116)*
Fluor Corp., Ltd., 2500 S. Atlantic Blvd., Los Angeles 22, Cal.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Heat-X-Changer Co., Inc., 415 Lexington Ave., N.Y.C. 17 *(p. 240)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Kirsch Co., Sturgis, Mich. (Hydrogen brazed) *(p. 54)*
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
Larkin Coils, 519 Memorial Dr , S.E., Atlanta 1, Ga. *(p. 222)*
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Lummus Co., 420 Lexington Ave., N.Y.C.
Mueller Brass Co., Port Huron, Mich.
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Kan. *(p. 81)*
Quaker City Iron Wks., Aramingo Ave. & E. Tioga St., Phila., Pa.
Remco, Inc., 49th St. & A.V.R.R., Pittsburgh 1, Pa.
Rome-Turney Radiator Co., Rome, N.Y. *(p. 43)*
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Angeles 11, Cal. *(p. 80)*
Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
David Stout & Sons, 7016 Manchester Blvd., St. Louis 17, Mo.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
Trane Co., La Crosse, Wis. *(p. 14)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*

"QUALITY SINCE 1876"
REFRIGERATOR HARDWARE

NEW KEIL YORK

"KEIL IS STANDARD"



- A complete line of locks and hinges.
- Safety "patented built-in" cylinder locks.
- Modern design non-cylinder locks.
- Easy action hinges.

STANDARD-KEIL HDWRE. MFG. CO., INC.
2413-23 Atlantic Ave.
Brooklyn 33, N.Y.

Build your
REFRIGERATION SYSTEMS
Right!!

Specify
Acklin
HYDROGEN-BRAZED
LIQUID RECEIVERS
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TOLEDO 7, OHIO

OFFICES IN
New York, Indianapolis, Detroit, Chicago, Buffalo, Cincinnati, Cleveland, Lansing, Milwaukee

Votator Div., Girdler Corp., Louisville 1, Ky. (Mechanical) *(p. 71)*
Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill. (Capillary Tube)
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*
Young Radiator Co., Racine, Wis. *(p. 16)*

HEAT EXCHANGER TEES

Mueller Brass Co., Port Huron, Mich.

HEAT PUMPS (See CONDENSING UNITS)

HEAT TREATING

Atlas Drop Forge Co., 209 W. Mt. Hope Ave., Lansing 2, Mich.
Duff-Norton Mfg. Co., Box 1889, Pittsburgh 30, Pa.

HEATERS, AIR, ELECTRIC

American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Dravo Corp., 300 Penn Ave., Pittsburgh 22, Pa.
Falcon Equip. Co., 13th St., New Brighton, Pa.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Red Spot Elec. Co., Tacoma, Wash.
Wesix Elec. Heater Co., 390-1st St., San Francisco 5, Cal.
Westinghouse Elec. Corp., Meadville, Pa.
Edwin L. Wiegand Co., 7506 Thomas Blvd., Pittsburgh 8, Pa.

HEATERS, AIR, GAS FIRED

Bryant Heater Co., 17825 St. Clair Ave., Cleveland 10, O.

HEATER ELEMENTS, ELECTRIC

American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.
Clarostat Mfg. Co., Inc., 130 Clinton St., Brooklyn 2, N.Y.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Red Spot Elec. Co., Tacoma, Wash.
Wesix Elec. Heater Co., 390-1st St., San Francisco 5, Cal.
Westinghouse Elec. Corp., Meadville, Pa.
Edwin L. Wiegand Co., 7506 Thomas Blvd., Pittsburgh 8, Pa.

HEATING COILS (See AIR CONDITIONING COILS)

HERMETIC HOUSINGS

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 124)*

HERMETIC TERMINALS (See TERMINALS, HERMETIC)

HIGHSIDE FLOATS (See also FLOAT SWITCHES)
(A—Ammonia; B—Other refrigerants)

(A,B) Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 93)*
(B) Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 159)*
(A,B) Armstrong Machine Wks., 846 Maple St., Three Rivers, Mich.
(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(B) Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill *(p. 111)*
(A,B) King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 158)*
(A,B) H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 114)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A,B) York Corp., York, Pa. *(p. 163)*

RICHARD M. ARMSTRONG CO.

BOX 188
WEST CHESTER, PA.

HEAT TRANSFER DESIGNERS AND BUILDERS

LIQUID—SUCTION HEAT EXCHANGERS
5 to 100 Tons Capacity

New Design gives practically no pressure loss
High Superheat (65F outlet temp.)
Low Cost per ton of capacity

These exchangers are most useful on low temperature jobs and water cooling jobs. Avoid sweatback by boiling liquor out of suction vapor, thereby also helping to reduce oil pumping by compressor.

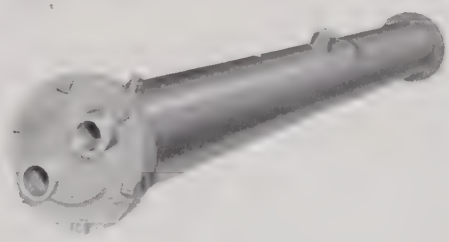
FREON CONDENSERS—1 TO 100 TONS

LAND

Copper Finned Tubes—rolled into steel tube sheet

Shell—steel

Heads—cast iron



MARINE

Tubes—Cupronickel finned

Tube Sheets—Cupronickel

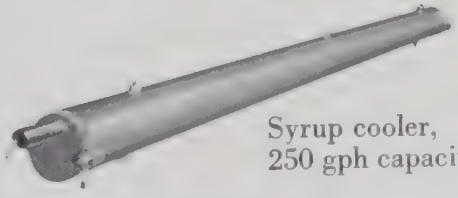
Shell—steel

Heads—cast iron

SHELL AND TUBE
A.S.M.E. Stamped When Required

LIQUID COOLERS—SHELL AND TUBE

Dry Expansion Water Chillers
Flooded Water Chillers
Vegetable Oil Chillers
Stainless Steel Coolers



Syrup cooler,
250 gph capacity

ALL DESIGNS BACKED BY MANY YEARS OF HEAT TRANSFER EXPERIENCE

MISCELLANEOUS SHELL AND TUBE HEAT EXCHANGERS

For engineering and installation data on commercial locks and hinges:

1. Consult the Grand Rapids Brass Company catalog. This provides full scale dimensional drawings, with screw hole centers, screw sizes, offsets, and other useful data on standard exterior-mounted commercial refrigerator hardware. May we send you a copy?

2. Similar information, with reduced scale drawings, is available in our 4-page installation manual.

3. When you're faced with a special problem, don't hesitate to call or write the home office. Our experience and cooperation are yours to command!

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it GRIPS

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GRAND RAPIDS, MICHIGAN

AMERICA'S *Quality* LINE OF
COMMERCIAL REFRIGERATOR HARDWARE

HINGES, REFRIGERATOR (See also HARDWARE)

Chase Industrial Refrigerator Equip. & Engrg. Co., 630 Reading Rd., Reading, Cin'ti. 15. O.
Grand Rapids Brass Co., 60 Scribner Ave., N.W. Grand Rapids 1, Mich. *(p. 126)*
Kason Hardware Corp., 127 Wallabout St., Brooklyn 6, N.J.
Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 16, Ill.
Standard-Keil Hardware Mfg. Co., Inc., 2413 Atlantic Ave., Brooklyn 33, N.Y. *(p. 123)*
Stanley Wks., 195 Lake St., New Britain, Ct.
Veeder-Root, Inc., Garden & Sargeant Sts., Hartford 2, Ct.
E. R. Wagner Mfg. Co., 4001 N. 32nd St., Milwaukee, Wis.

HOISTS (See CRANES)

HOME & FARM FREEZERS; FROZEN FOOD CABINETS

Ace Cabinet Corp., New Bedford, Mass.
Admiral Corp., 3800 Cortland St., Chicago 47, Ill.
Amana Society, Amana, Ia.
American Central Mfg. Corp., Div. of Aviation Corp., 18th & Columbia Sts., Connersville, Ind.
American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
American Refrigerator & Machine, Inc., 2700 University, N.E., Minneapolis, Minn.
Artkraft Mfg. Corp., Kibby St. & D.T.&I.R.R., Lima, O
Authorized Cabinet Co., Grand Haven, Mich.
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Ben-Hur Mfg. Co., 634 E. Keefe Ave., Milwaukee 12, Wis.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Central Supply Co., 210 S. Capitol Ave., Indpls., Ind.
Chapman Refrigerator Sales, Corvallis, Ore.
Coolerator Co., 50 Ave W., and Wadena St., Duluth 1, Minn.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Crosley Div., Aviation Corp., Cin'ti. 25, O.
Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
Deepfreeze Div., Motor Products Corp., 2301 Davis St., N. Chicago, Ill.
Econofreze Mfg. Co, Inc., 300 Polson Bldg., Seattle 4, Wash.
Electro-Kold, Div. of E. S. Matthews, Inc., S. 151 Post St., Spokane 8, Wash.
Esco Cabinet Co., West Chester, Pa.
Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
W. J. Finnegan Co., 7402 Santa Monica Blvd., Los Angeles 46, Cal. *(p. 116)*
Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Fraser & Johnston Co., 725 Potrero Ave., San Francisco 10, Cal.
Frez-O-Mat Corp., 4800 S. Richmond St., Chicago 32, Ill.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Frostair Div., General Tire & Rubber Co., 332 S. Michigan Ave., Chicago 4, Ill.
Frosty Foods Equip. Co., 305 Benson Blvd., Sioux City 15, Ia.
General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
General Machine Co., Inc., 4th & Furnace St., Emmaus, Pa.
General Refrigerator & Store Fixtures Co., 856 N. Broad St., Phila., Pa.
Gennett & Sons, Inc., 1 Main St., Richmond, Ind.
Gibson Refrigerator Co., 515 W. Williams St., Greenville, Mich.
Harder Refrigerator Corp., Div. of Tyler Fixture Corp., Cobleskill, N.Y.
Havérly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.

C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
Holverson Co., 209 E. Lake St., Minneapolis 3, Minn.
Hotpoint, Inc., 5600 W. Taylor St., Chicago 44, Ill.
Hussman Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
Ice Cooling Appliance Corp., 33 S. Clark St., Chicago 3, Ill.
Ideal Cooler Corp., 2953 Easton Ave., St. Louis 6, Mo.
International Harvester Co., 180 N. Michigan Ave., Chicago 1, Ill.
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
Leonard Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
McCall Refrigerator Corp., Hudson, N.Y.
McCray Refrigerator Co., Kendallville, Ind.
Marquette Appliances, Inc., 307 E. Hennepin Ave., Minneapolis 14, Minn.
Master-Bilt Refrigeration Mfg. Co., 920 Palm St., St. Louis 7, Mo.
Masterfreeze Corp., Sister Bay, Wis.
Matthews Refrigerator & Door Co., 5103 S. E. Powell Blvd., Portland 6, Ore.
Modern Appliance Co., 111 S. Ellsworth, San Mateo, Cal.
Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
National Cooperatives, Inc., 343 S. Dearborn St. Chicago 4, Ill.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
C. Nelson Mfg. Co., 4010 N. Union St., St. Louie 15, Mo.
Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.
Paley Mfg. Corp., 244 Herkimer St., Brooklyn 16, N.Y.
Pennco, Inc., 201 N. Broad St., Phila. 7, Pa.
Refrigeration Corp. of America, Div. of Noma Elec. Corp., 55 W. 13th St., N.Y.C. 11
Revco, Inc., Deerfield, Mich.
Rex Mfg. Co., Inc., Western Ave., Connersville, Ind.
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.
W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
Royal Store Fixture Co., 847 N. Broad St., Phila. 23, Pa.
St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.
Sanitary Refrigerator Co., Fond du Lac, Wis. *(p. 127)*
Schaefer, Inc., 801 Washington Ave., N., Minneapolis 1, Minn.
C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
Schultz Bros., Inc., 2130 Midland Rd., Saginaw, Mich.
Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.
Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Spir-O-Freez Co., Inc., 1077 Castleton Ave., Staten Island 10, N.Y.
Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*
Emil Steinhorst & Sons, Inc., 612 South St., Utica 3, N.Y.
Stoddard Mfg. Co., 617 4th St., S.W., Mason City, Ia.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
United Refrigerator Mfg. Co., Inc., 350 Robert St., St. Paul 1, Minn.
U. S. Thermo Control Co., 44 S. 12th St., Minneapolis 4, Minn.
Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.
Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.
Wells & Brunnell Mfg. Co., Inc., P.O. Box 1555, Tacoma 1, Wash.
Western Mineral Products Co., 1720 Madison St., N.E., Minneapolis 13, Minn.

Westinghouse Elec. Corp., 653 Page Blvd., Springfield 2, Mass.
Whiting Corp., 38 S. Dearborn St., Chicago 3, Ill.
Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.
Yoder Corp., 133 W. 80th St., Cleveland 2, O.
York Corp., York, Pa. *(p. 163)*

HOOKS (See also SUPPORTS; also HARDWARE SPECIALTIES)

Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.

HOSE, FLEXIBLE METAL (See also CHARGING LINES; also FLEXIBLE CONNECTIONS; also TUBING, FLEXIBLE)

Aeroquip Corp., 300 S. East Ave., Jackson, Mich. *(p. 83)*
American Brass Co., Waterbury 83, Ct.
Atlantic Metal Hose Co., Inc., 124 W. 64th St., N.Y.C. 23
Brockway Co., 361 Church St., Naugatuck, Ct.
Chicago Metal Hose Corp., Maywood, Ill.
Eclipse Aviation Metal Hose Dept., Div. of Bendix Aviation Corp., Phila. 44, Pa.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Packless Metal Products Corp., 31 Winthrop Ave., New Rochelle, N.Y.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Seamlex Co., Inc., 4123-24th St., Long Island City 1, N.Y. *(p. 218)*
Titeflex, Inc., 500 Frelinghuysen Ave., Newark 5, N.J.
John M. Watt's Sons, 112 Walnut St., Phila., Pa.

HOSE, RADIATOR, RUBBER, WATER, etc.

Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
DeVilbiss Co., 300 Phillips Ave., Toledo 1, O.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Hudson Products Co., Inc., 4400 St. Aubin, Detroit 7, Mich.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
John M. Watt's Sons, 112 Walnut St., Phila., Pa.

HOSE, FLEXIBLE, SYNTHETIC

Aeroquip Corp., 300 S. East Ave., Jackson, Mch. *(p. 83)*

HOSE CLAMPS (See FITTINGS, HOSE)

HOUSINGS (See particular type, i.e., BLOWER HOUSINGS)

HUMIDIFIERS & HUMIDIFYING SYSTEMS (See also SPRAY HUMIDIFIERS)

Air & Refrigeration Corp., 475-5th Ave., N.Y.C. 17
American Blower Corp., Div. of American Radiator & Standard Sanitary Corp., 8111 Tireman Ave., Detroit 32, Mich.
American Coils Co., 25 Lexington St., Newark 5, N.J.
American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.
Armstrong Machine Wks., 846 Maple St., Three Rivers, Mich.
Bahnson Co., 1001 S. Marshall St., Winston-Salem 7, N C.
Bowser, Inc., Terryville, Ct.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Drying Systems, Inc., 1810½ Foster Ave., Chicago 40, Ill.
Farr Co., 2615 Southwest Dr., Los Angeles, Cal.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Monitor Equip. Corp., Riverdale, N.Y.C. 63
National Engrg. & Mfg. Co., 213 W. 19th St., Kansas City 8, Mo.
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Fred D. Pfening Co., 1075 W. 5th Ave., Columbus 8, O.
Rhode Island Humidifier & Ventilating Co., 99 Chauncey St., Boston 11, Mass.
Ross Sprinkler Co., 34 Roberts St., Pasadena 3, Cal.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
H. J. Somers, Inc., 6063 Wabash Ave., Detroit 8, Mich.
Spraying Systems Co., 4021 W. Lake St., Chicago 24, Ill.
Standard Engrg. Wks., 289 Roosevelt Ave., Pawtucket, R.I.
B. F. Sturtevant Div. Westinghouse Elec. Corp., Hyde Park, Boston 39, Mass. *(p. 148)*
Supreme Elec. Products Co., 194 Vassar St., Rochester 7, N.Y.
Techtman Industries, 714 W. Wisconsin Ave., Milwaukee 1, Wis.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
Stanley Wertheim & Son, 58-01-41st Dr., Woodside, N.Y.C.
York Corp., York, Pa. *(p. 163)*

HUMIDISTATS (See CONTROLS, HUMIDITY)

HYDROCARBONS

Carbide & Carbon Chemicals Corp., Unit. of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17

HYDROGEN

Air Reduction Sales Co., 60 E. 42nd St., N.Y.C. 17
Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17 *(p. 197)*
Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.

HYDROMETERS

General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*

HYGROMETERS

American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.
Bristol Co., Waterbury 91, Ct. (Recording only)
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
GM Mfg. Co., 50 W. 3rd St., N.Y.C. 12
General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Henry J. Green, 1191 Bedford Ave., Brooklyn 16, N.Y. (Also recording)
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti. 12, O.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 185)*
Carl H. Wagner & Son, 1944 N. Albany Ave., Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

ICE CANS

American Rolling Mill Co., Middletown, O.
Frick Co., Waynesboro, Pa. *(p. 47)*
Knickerbocker Stamping Co., Parkersburg, W. Va.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p.130)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

ICE CREAM CABINETS

Ace Cabinet Corp., New Bedford, Mass.
American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
American Refrigerator & Machine, Inc., 2700 University, N.E., Minneapolis, Minn.
Authorized Cabinet Co., Grand Haven, Mich.
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Fraser & Johnston Co., 725 Potrero Ave., San Francisco 10, Cal.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
Stanley Knight Corp., 3430 N. Pulaski Rd., Chicago 41, Ill.
Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. 62)*
Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo. (Sectional Construction)
C. Nelson Mfg. Co., 4016 N. Union St., St. Louis 15, Mo.
Paley Mfg. Corp., 244 Herkimer St., Brooklyn 16, N.Y.
Pennco, Inc., 201 N. Broad St., Phila. 7, Pa.
Piper Co., Govans Post Office, Baltimore 12, Md.
Refrigeration Corp. of America, Div. of Noma Elec. Corp. 55 W. 13th St., N.Y.C. 11
W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.
Savage Arms Corp., 311 Turner St., Utica 1, N.Y.
Schaefer, Inc., 801 Washington Ave., N., Minneapolis 1, Minn.
Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Sweden Freezer Mfg. Co., 1140 W. 53rd St., Seattle 7, Wash.
Tekni-Craft, 845-5th, Beloit, Wis.
Emery Thompson Machine & Supply Co., 1349 Inwood Ave., N.Y.C. 52
Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.
Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.

ICE CREAM CABINET CONVERSION UNITS

Ace Cabinet Corp., New Bedford, Mass.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill. *(p. 167)*
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. 62)*
Standard Refrigeration Co., 232 S. Hoyne Ave., Chicago 20, Ill.
Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*

ICE CREAM FREEZERS

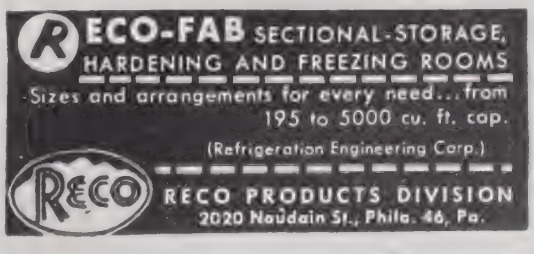
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse. N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Maysteel Products, Inc., 135 W. Wells, St. Milwaukee 3, Wis.
Port Morris Machine Co., 712 E. 135th St., N.Y.C. 54 (Frozen Custard Machine)
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Sweden Freezer Mfg. Co., 1140 W. 53rd St., Seattle 7, Wash.
Taylor Freezer Corp., Beloit, Wis. (Counter)
Tekni-Craft, 845-5th, Beloit, Wis.
Emery Thompson Machine & Supply Co., 1349 Inwood Ave., N.Y.C. 52
Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.
Votator Div. Girdler Corp., Louisville 1, Ky. *(p. 71)*

ICE CREAM HARDENING CABINETS

Ace Cabinet Corp., New Bedford, Mass.
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Holverson Co., 209 E. Lake St., Minneapolis 3, Minn.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. 62)*
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo. (Sectional Construction)
C. Nelson Mfg. Co., 4016 N. Union St., St. Louis 15, Mo.
St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.
Schaefer, Inc., 801 Washington Ave., N., Minneapolis 1, Minn.
Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Sweden Freezer Mfg. Co., 1140 W. 53rd St., Seattle 7, Wash.
Taylor Freezer Corp., Beloit, Wis.
Tekni-Craft, 845-5th, Beloit, Wis.
Emery Thompson Machine & Supply Co., 1349 Inwood Ave., N.Y.C. 52
Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.

ICE CREAM HARDENING ROOMS

Cork Insulation Co., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo. (Sectional Construction)
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 129)*
Refrigeration Corp. of America, Div. of Noma Elec. Corp., 55 W. 13th St., N.Y.C. 11
St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.
Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Sweden Freezer Mfg. Co., 1140 W. 53rd St., Seattle 7, Wash.



Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

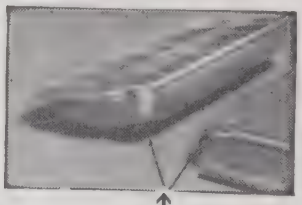
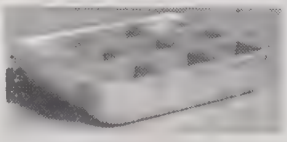
ICE CUBE CUTTERS

Gifford-Wood Co., Hill St., Hudson, N.Y.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Thermo Cuber Co., Inc., 3260 W. Grand Ave., Chicago 51, Ill.

ICE CUBE MAKERS
(A—Ammonia; B—Other refrigerants)

(B) Bemco Mfg. Corp., 1504 Minor at Pike, Seattle 1, Wash.
(B) Betz Corp., 445 State St., Hammond, Ind.
(B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(B) Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
(B) Holverson Co., 209 E. Lake St., Minneapolis 3, Minn.
(B) Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
(B) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*

BULK ICE MAKER
Four Models 250 to 2000 lbs. daily cap., cube or block ice. Electric, Gasoline or Diesel engine driven.
(Refrigeration Engineering Corp.)
RECO
RECO PRODUCTS DIVISION
2020 Naudain St., Phila. 46, Pa.

ALUMINUM ICE CUBE TRAYS
Immediate Delivery

RELEASE NOTCH

Write for Particulars
EDISON COOLING CORP.
310 E. 149th St. New York 51, N.Y.

(B) Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
(B) McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudian St., Phila. 46, Pa. *(p. 129)*
(A,B) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
(B) Standard Refrigeration Co., 232 S. Hoyne Ave., Chicago 20, Ill.
(A,B) Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*
(B) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(B) United Refrigerator Mfg. Co., Inc., 350 Robert St., St. Paul 1, Minn.
(B) Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
(B) York Corp., York, Pa. *(p. 163)*

ICE CUBE TRAYS

Aluminum Goods Mfg. Co., Manitowoc, Wis.
Edison Cooling Corp., 310 E. 149th St., Bronx 51, N.Y. *(p. 130)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Hoosier Cardinal Corp., 601 E. Eichel Ave., Evansville 7, Ind.
Inland Mfg. Div., Gen'l. Motors Corp., Dayton 1, O.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Plastray Corp., 823 Fisher Bldg., Detroit 2, Mich.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.

ICE MAKERS, CABINET TYPE

Bemco Mfg. Corp., 1504 Minor at Pike, Seattle 1, Wash.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Esco Cabinet Co., West Chester, Pa.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Holverson Co., 209 E. Lake St., Minneapolis 3, Minn.
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 129)*
Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.

ICE MANUFACTURING ACCESSORIES (See also ICE CANS)

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Gifford-Wood Co., Hill St., Hudson, N.Y.
Jamison Cold Storage Door Co., Hagerstown, Md. *(p. 45)*
Knickerbocker Stamping Co., Parkersburg, W.Va.
Link-Belt Co., 2410 W. 18th St., Chicago 8, Ill.
Jos. A. Martocello & Co., 229 N. 14th St., Phila. 7, Pa. *(p. 157)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*

ICE PLANTS & COMPONENTS
• Can Dumpers • Can Fillers
• Brine Agitators • Pressure Vessels
• Tank Framing & Covers
(Refrigeration Engineering Corp.)
RECO
RECO PRODUCTS DIVISION
2020 Naudain St., Phila. 46, Pa.

Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

ICE MANUFACTURING MACHINES, AUTOMATIC

Flakice Corp., 360 Furman St., Brooklyn 2, N.Y.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 163)*

ICE STORAGE CHESTS

Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Gustav Glaser Co., Inc., 2 Wait St., Paterson 4, N.J.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Ice Cooling Appliance Corp., 33 S. Clark St., Chicago 3, Ill.
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.

ICE TANKS (See also BRINE TANKS)

Audiffren Refrigerating Co., Proctor, Vt.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Chattanooga Boiler & Tank Co., P.O. Box 110, Chattanooga, Tenn.
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Gustav Glaser Co., Inc., 2 Wait St., Paterson 4, N.J.
R. Munroe & Sons Mfg. Corp., 23rd & Smallman Sts., Pittsburgh 22, Pa.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Quaker City Iron Wks., Aramingo Ave. & E. Tioga St., Phila., Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Reynolds Mfg. Co., Inc., Springfield, Mo.
Stover Steel Tank & Mfg. Co., 100 S. Hancock St., Freeport, Ill.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

ICE TANK TOPS

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Jamison Cold Storage Door Co., Hagerstown, Md. *(p. 45)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Reynolds Mfg. Co., Inc., Springfield, Mo.
Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Angeles 11, Cal. *(p. 80)*
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

INDICATORS (See particular type, i.e., FLOW RATE; LIQUID; LEAK; MOISTURE, etc.)

INHIBITORS, CORROSION (See also WATER TREATING)

American Chemical Paint Co., Ambler, Pa. *(p. 41)*
American Sand-Banum Co., Inc., 9 Rockefeller Plaza, N.Y.C. 20 *(p. 36)*
W. H. & L. D. Betz, Gillingham & Worth Sts., Phila. 24, Pa.
Black Bear Co., Inc., 44-23rd St., Long Island City 1, N.Y.
Henry Bower Chemical Mfg. Co., Gray's Ferry Rd. & 29th St., Phila. 46, Pa.
Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Chemical Solvent Co., Box 487, Birmingham, Ala.
Dearborn Chemical Co., 310 S. Michigan Ave., Chicago, Ill.
Elgin Softener Corp., Elgin, Ill.
Feedwaters, Inc., 140 Cedar St., N.Y.C. 6
Ferro-Nil Corp., 381-4th Ave., N.Y.C. 16
D. W. Haering & Co., 205 W. Wacker Dr., Chicago 6, Ill.
E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
National Aluminate Co., 6216 W. 66th Place, Chicago 38, Ill. *(p. 244)*
New Jersey Zinc Co., 160 Front St., N.Y.C. 7
North American Fibre Products Co., Standard Bldg., Cleveland 13, O.
Nox-Rust Chemical Corp., 2429 S. Halsted St., Chicago 8, Ill.
Oakite Products, Inc., 22 Thames St., N.Y.C. 6
Perolin Co., Inc., 10 E. 40th St., N.Y.C. 16
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Tempo Chemical Co., Inc., 47-02-5th St., Long Island City 1, N.Y.
Tuco Products Corp., 30 Church St., N.Y.C. 7
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
Wilbur-Williams Co., 43 Leon St., Boston, Mass.

INJECTORS, REFRIGERANT

Cold Control, Inc., 111 Broadway, N.Y.C. 6
H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 131)*

INSERTS FOR MOLDING, etc.

Aircraft Screw Products Co., Inc., 47-23-35th St., Long Island City 1, N.Y.
Ohio Nut & Bolt Co., 600 Front St., Berea, O.

INSTRUMENT PANELS

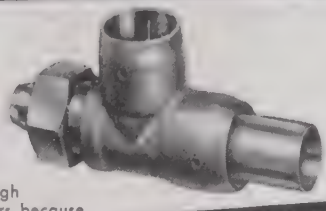
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Bristol Co., Waterbury 91, Ct.
Falstrom Co., 13 Falstrom Court, Passaic, N.J.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Marshalltown Mfg. Co., Marshalltown, Ia.

INSULATING PLASTER & CONCRETE

Airtex Corp., 333 N. Michigan Ave., Chicago 1, Ill.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Robert A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
Munn and Steele, Inc., 130 Lister Ave., Newark 5, N.J.
National Gypsum Co., 325 Delaware Ave., Buffalo 2, N.Y. *(p. 133)*

(Continued)

PHILLIPS INJECTOR



PHILLIPS Refrigerant Injectors assure better heat transmission through certain evaporators because liquid refrigerant is continuously recirculated at higher mass velocities. Trouble-free operation. For both single and multiple evaporator installations. Write for information.

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3255 W. Carroll Avenue
CHICAGO 24, ILLINOIS

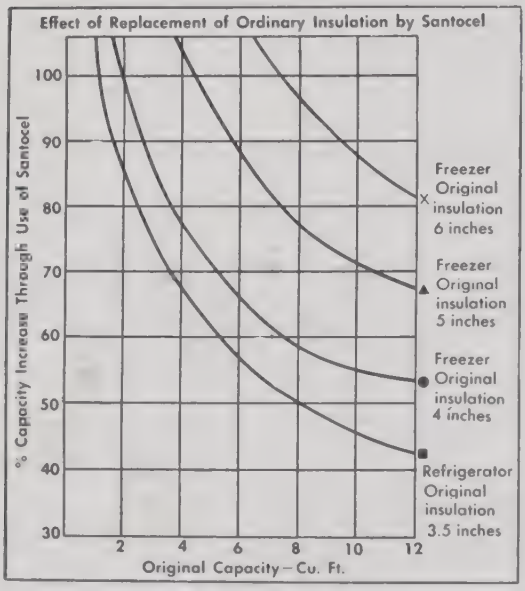
quick facts about SANTOCEL "world's most efficient insulation"

What it is

Santocel is Monsanto's unique silica aerogel, the only insulation with heat transmission lower than the theoretical, calculated value for "still air."

What it offers

Santocel's unprecedented insulating efficiency presents important opportunities for space savings and hitherto impossible applications.



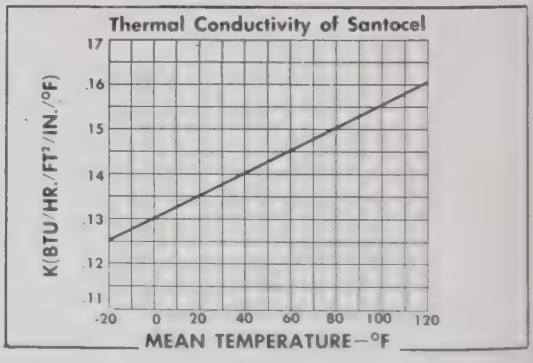
Where it is used

Applications where Santocel has already produced results heretofore considered unobtainable, include:

- Domestic Refrigerators
- Farm and Home Freezers
- Industrial Metal Chilling Cabinets
- Stratosphere Chambers
- Frozen Food Lockers
- Electric Ovens
- Portable Frozen Food Shipping Containers
- Thermos Jugs
- Railroad Tank Cars
- Insulated Doors
- Yacht Refrigerating Equipment

High Efficiency

Santocel's amazing flow properties enable fast, automatic filling by the most efficient and economical methods. Assembly is simplified, production speeded.



You can capitalize on the outstanding space saving features of Santocel which offer maximum storage capacity with minimum over-all dimensions. Get complete information . . . technical data on Santocel from: MONSANTO CHEMICAL COMPANY, Merrimac Division, Boston 49, Massachusetts.

Santocel: Reg. U. S. Pat. Off.

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MONSANTO CHEMICALS — PLASTICS

SERVING INDUSTRY . . . WHICH SERVES MANKIND

INSULATING PLASTER (Continued)

Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Refractory & Insulation Corp., 120 Wall St., N.Y.C. 5
Universal Zonolite Insulation Co., 135 S. La Salle St., Chicago 3, Ill. *(p. 135)*
Western Mineral Products Co., 1720 Madison St., N.E., Minneapolis 13, Minn.

INSULATION, AEROGEL

Merrimac Div., Monsanto Chemical Co., Everett 49, Mass. *(p. 132)*

INSULATION, ASBESTOS, MAGNESIA, etc.

Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
Armstrong Cork Co., Lancaster, Pa. *(p. 136)*
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
Keasbey & Mattison Co., Ambler, Pa.
Mundet Cork Corp., 7101 Tonnelle Ave., N. Bergen, N.J. *(p. 141)*
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Refractory & Insulation Corp., 120 Wall St., N.Y.C. 5
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Standard Asbestos Mfg. Co., 820 W. Lake St., Chicago 7, Ill.
Standard Asbestos Mfg. & Insulation Co., 410 N. Olive St., Kansas City 1, Mo.

INSULATION, BALSA

Geo. D. Emery Co., 220-11th Ave., N.Y.C. 1
International Balsa Corp., 96 Boyd Ave., Jersey City 4, N.J.

INSULATION, CELLULOSE FIBRE

Airtex Corp., 333 N. Michigan Ave., Chicago 1, Ill.
Cellufoam Products Div., Masonite Corp., 111 W. Washington St., Chicago 2, Ill.
Celotex Corp., 120 S. La Salle St., Chicago 3, Ill.
Hinde & Dauch Paper Co., P.O. Box 86, Sandusky, O.
Insul-Wool Insulation Corp., 121 N. Dodge St., Wichita, 12, Kan.
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
Kimberly-Clark Corp., Neenah, Wis.
B. F. Nelson Mfg. Co., 401 Main St., N.E., Minneapolis 13, Minn.
Pacific Lumber Co., 100 Bush St., San Francisco 4, Cal. *(p. 137)*
Plastergon Wall Board Co., Box 40, Sta. B, Buffalo 7, N.Y.
Standard Asbestos Mfg. & Insulation Co., 410 N. Olive St., Kansas City 1, Mo.
Wood Conversion Co., First Nat'l. Bank Bldg., St. Paul 1, Mass *(p. 139)*

INSULATION, CORK

Armstrong Cork Co., Lancaster, Pa. *(p. 136)*
Cork Import Corp., 39 Park Place, Englewood, N.J.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
La Suberina, S.A., San Feliu de Guixols (Catalonia) Spain *(p. 140)*
Albert O. Lloveras, 143 Waverly Place, N.Y.C. 14 *(p. 140)*
Mundet Cork Corp., 7101 Tonnelle Ave., N. Bergen, N.J. *(p. 141)*
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Standard Asbestos Mfg. & Insulation Co., 410 N. Olive St., Kansas City 1, Mo.
United Cork Cos., Kearny, N.J. *(p. 143)*

8 IMPORTANT REASONS WHY
ZEROCEL INSULATION
SHOULD BE YOUR CHOICE

1 EFFICIENCY "K" Factor .24 btu at 60°F.
2 FIREPROOF
3 WILL NOT ABSORB MOISTURE
4 WILL NOT ABSORB ODORS
5 WILL NOT SETTLE
6 IMMUNE TO FUNGUS AND DECAY
7 WILL NOT ATTRACT VERMIN
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Ask for free copy of the booklet, "Fire-proof Refrigeration Construction."

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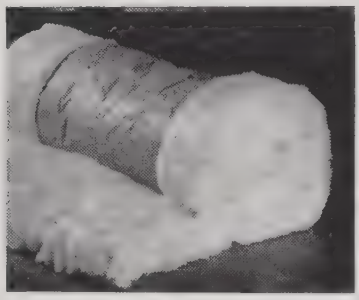
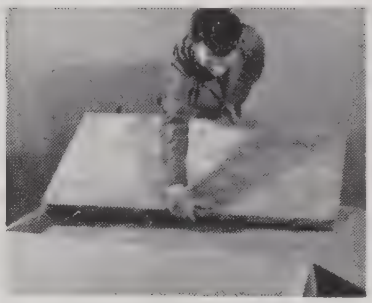
OWENS-CORNING
FIBERGLAS
T.M. REG. U.S. PAT. OFF
Insulations

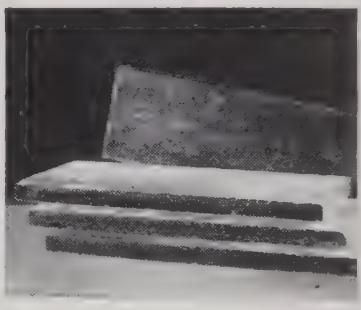
FOR EVERY LOW-TEMPERATURE NEED



INSULATING WOOL, TYPE TW-F
A soft, resilient, light-weight insulation—in roll, bat, bulk and shredded forms—suitable for many applications and temperatures from sub-zero to 1000°F.



PF (Pre-Formed) INSULATION
Widely used for domestic and commercial refrigerators and other low-temperature equipment. Available in seven standard densities from 2½ to 10.5 lbs. per cu. ft.



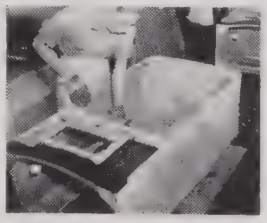
AE (Asphalt-Enclosed) BOARD
Made of a core of Fiberglas PF (Pre-Formed) Insulation completely enclosed with an asphalt coating. Easy to cut and install. Boards (12″ x 36″) available in 6 lb. or 9 lb. per cu. ft. density.

● A unique combination of qualities has made FIBERGLAS the preferred insulating material for a wide variety of low-temperature applications.

In all of its easy-to-use forms, Fiberglas provides high thermal insulating efficiency. Being inorganic, Fiberglas is odorless and does not pick up odors in service . . . provides no sustenance for insects, vermin or rodents. It is highly moisture-resistant. Light in weight, Fiberglas has good mechanical strength—doesn't settle, "dust" or disintegrate under vibration.

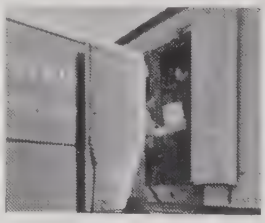
●

Write for complete technical information and application data concerning Fiberglas Insulations for low-temperature service.

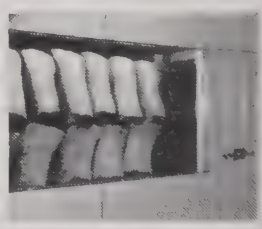


Refrigerator Units

R.R. Refrigerator Cars



Refrigerator Trucks



Cooler Rooms

Food Terminals

Ship Refrigerator Rooms

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Universal Zonolite Insulation Company

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ZONOLITE
BRAND
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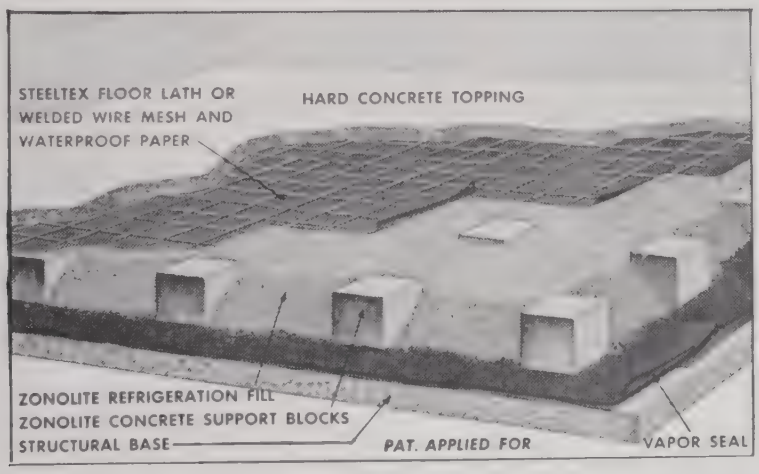
Plants in Principal Cities
—United States and
Canada

NEW WAY TO BUILD COLD STORAGE FLOORS!



Fireproof..Rotproof..Verminproof..Lightweight!

Entirely different from conventional construction, this floor combines low cost with maximum efficiency, permanence and ease of installation.

The high percentage of Zonolite Refrigeration Fill used in this type construction, assures high efficiency at a low dollar cost. A concrete slab resting on support blocks of Zonolite Insulating Concrete, permits live loads up to 300 lbs. per square foot wtih ample safety margin.

You will find your Zonolite floor outlasts the structure itself. Zonolite is impervious to rot under even the most abnormal conditions. Fireproof, odorless and non-settling, it cannot harbor vermin or rodents.

For full details on how to construct this floor, send for Zonolite's New FREE book, "Cold Storage Insulation."

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Chicago, Illinois

Armstrong Cork Company

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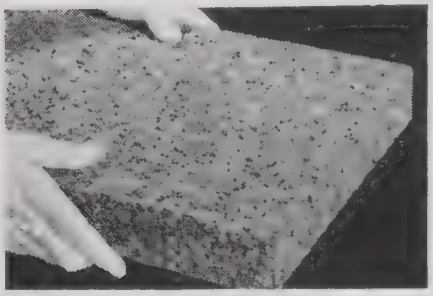
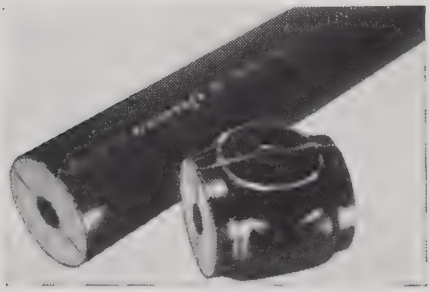
Offices and distributors in principal cities

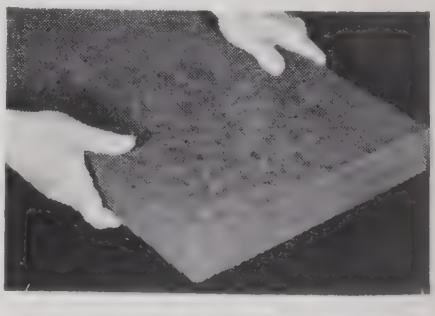
ARMSTRONG'S LOW-TEMPERATURE INSULATIONS



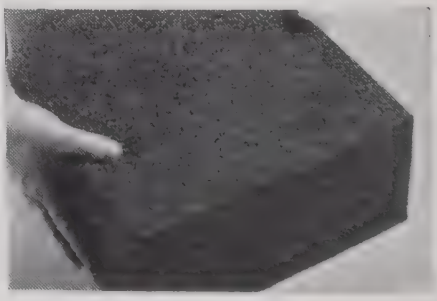
ARMSTRONG'S CORKBOARD—for cold rooms, ducts, roofs, and refrigerated equipment. Thermal coefficient is 0.27 Btu at 60°F. Weighs 0.65 lbs per bd. ft. Fire retardent, moisture resistant, structurally strong, easily handled, and offers lasting dependability. Sizes: 12″, 18″, 24″, and 36″ x 36″. Thicknesses: 1″, 1½″, 2″, 3″, 4″, and 6″.



ARMSTRONG'S CORK COVERING—for low-temperature lines. Made in 36″ split lengths in three thicknesses to fit all standard pipe sizes. Also made in greater thicknesses for special requirements. Fitting covers are made in the same thicknesses as pipe covering and are designed for all standard fitting.



ARMSTRONG'S MINERAL WOOL BOARD— a board-type, low-temperature insulating material made of mineral wool with a waterproof binding agent. Thermal coefficient is 0.33 Btu at 75°F. Weighs 1.25 lbs. per bd. ft. Equals or exceeds Federal Specification HH-M-371-a for board or block form insulation. Size: 12″ x 36″. Thicknesses: 1″, 1½″, 2″, 3″, and 4″.



FOAMGLAS*—a block-type, low-temperature insulating material made of glass in cellular form. Impervious to the passage of air and moisture, fireproof, easy to handle. Thermal coefficient is 0.40 Btu at 50°F. Size: 12″ x 18″. Thicknesses: 2″, 3″, 4″, and 5″.

* T.M. Reg. U. S. Pat. Off. by Pittsburgh-Corning Corp.

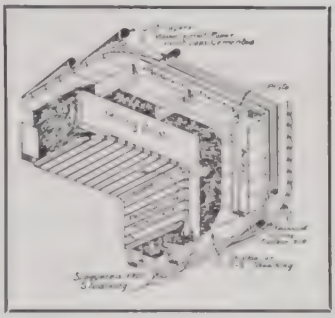
COMPLETE CONTRACT SERVICE

Armstrong Cork Company offers materials, workmanship, and engineering advice on any insulation job involving temperatures from 300° below zero to 2800°F. For complete information on Armstrong's Contract Service or low-temperature insulation materials, write to Armstrong Cork Company, Building Materials Division, 938 Concord Street, Lancaster, Penna.

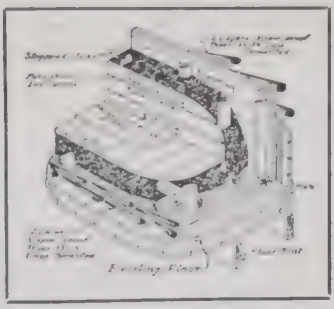
Specify PALCO INSULATION WOOL

FOR UNIFORM LOW TEMPERATURE CONTROL

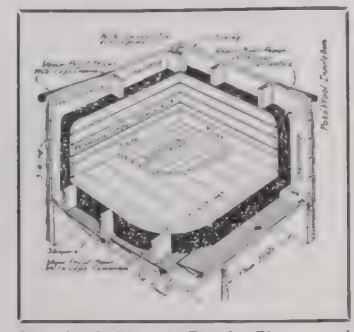
WHEN building or expanding frozen food lockers and cold storage plants, experienced refrigeration engineers know PALCO WOOL Insulation meets the most rigid and exacting requirements of non-settling permanence, long-life endurance, low thermal conductivity and fire resistance. Produced from the bark of California Redwoods, PALCO WOOL Insulation is inherently moisture resistant, odor proof, and vermin repellent. Readily adaptable to all types of construction, PALCO WOOL Insulation not only keeps operating costs at a minimum through its high efficiency, but also effects savings in capital charges by reducing plant investment and depreciation. Investigate today how to get continuing economies with PALCO WOOL Insulation. Write for full details.



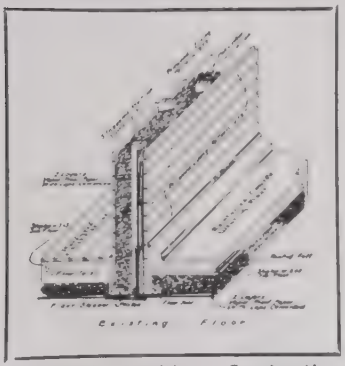
Typical Wall and Ceiling Wood Finish Construction for Low Temperatures.



Normal Frame Construction, Wood Walls, Flooring, and Exterior Siding for Cold Storage.



Standard Wood Finish Floor and Wall Construction for Freezers, etc.



Accepted Partition Construction between Freezer and Cooler with Wood Finish Wall and Floor, and Concrete Chill Floor.

CHECK THESE SUPERIOR QUALITIES OF PALCO WOOL INSULATION

- √ Low Thermal Conductivity
- √ Durable
- √ Moisture Resistant
- √ Vermin Repellent
- √ Non-Settling
- √ Flame-Proofed
- √ Odor-Proof
- √ Economical

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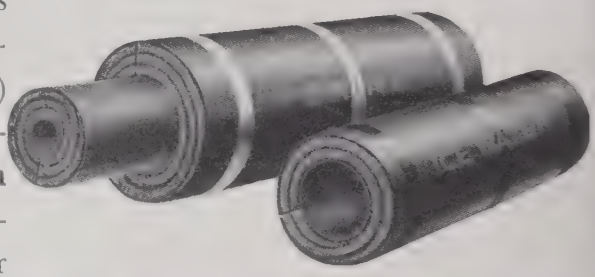
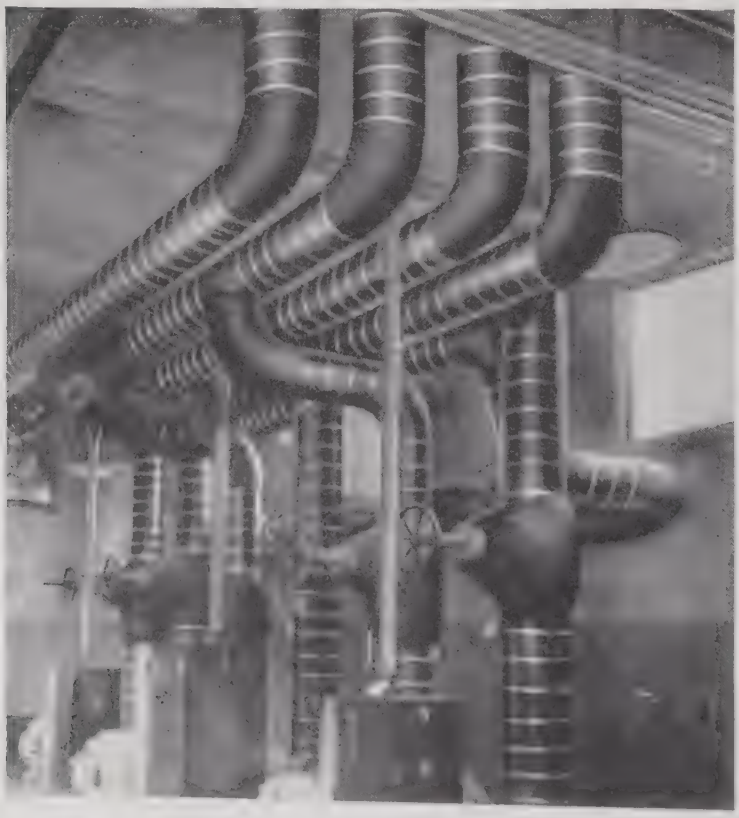
ARMORAK INSULATION FOR LOW TEMPERATURE PIPING



ARMORAK Sectional Covering is made in three standard thicknesses (Ice Water, Brine and Heavy) and in any special thickness, fabricated to fit any size of wrought iron pipe, copper or brass tubing. It is furnished in single and double layer construction for use at temperatures above 35° F. and in double layer construction for temperatures below 35° F. It is built up of successive layers of specially selected felts sealed to asphalt saturated paper and wound on mandrels of the desired size until the proper thickness is attained. The sections are then completely wrapped with a heavy armor-like outer jacket providing a strong, durable, waterproof and air-tight surface. Fittings, valves, flanges, and other irregular shaped surfaces are insulated on the job with insulating felt built up in successive layers to the proper pre-determined thickness. Each layer of felt is waterproofed with Rakcoseal special sealing compound and secured in place with windings of Jute Wrapping Twine. The outside layer of felt is primed with Rakcoseal and finished with Water-proof Cement.



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New York **Chicago**

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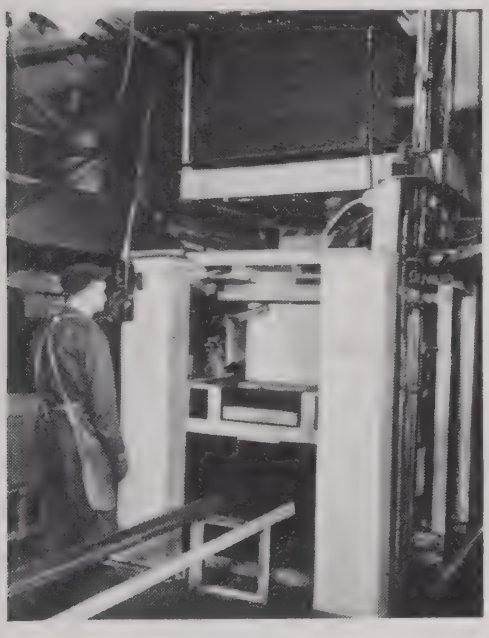

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K-25 Fiber* processed to the proper low density, provides an efficient and economical insulation material for walls, floors, ceilings and partitions of locker plants, cold storage plants, walk-in coolers; insulated refrigerator and cooler doors and other applications where a fill-type insulation is advantageous. In these applications, the compressed bales of K-25 Fiber are broken down, fluffed or reprocessed to the proper density by a portable bale breaker. The fiber is hand-packed into the space to be insulated. K-25 Fiber will give the best results when packed to a density of about 4 pounds per cubic foot.

Installation

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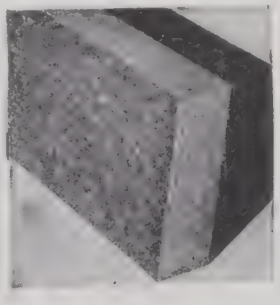

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Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Stic-Klip Mfg. Co., 50 Regent St., Cambridge 40, Mass. *(p. 144)*
United Cork Cos., Kearny, N.J. *(p. 143)*

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Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
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York Corp., York, Pa. *(p. 163)*

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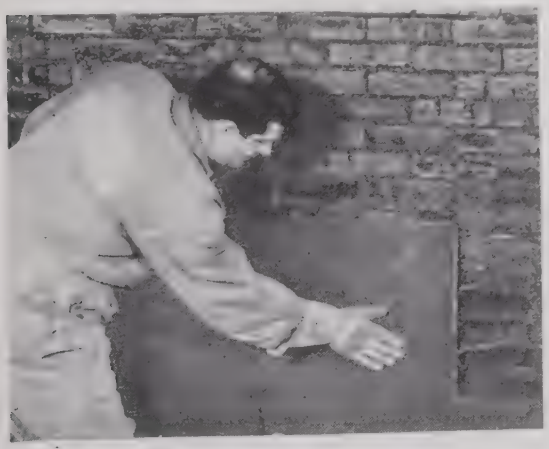

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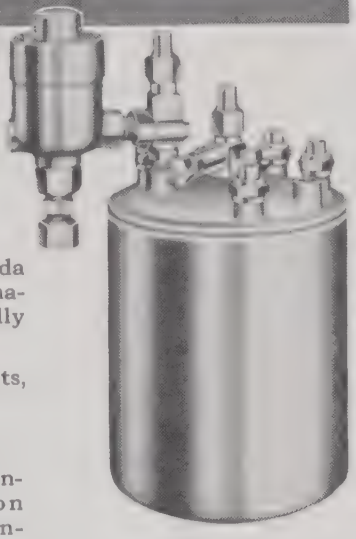
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Heat-X-Changer Co., Inc., 415 Lexington Ave., N.Y.C. 17 *(p. 240)*
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

LIQUID INDICATORS

Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill. *(p. 55)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Henry Valve Co., Melrose Park, Ill. *(p. 106)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 238)*
Lunkenheimer Co., Beekman St., & Waverly Ave., Cin'ti 14, O.
Mueller Brass Co., Port Huron, Mich.
Nathan Mfg. Co., 416 E. 106th St., N.Y.C. 29
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

TEMPRITE

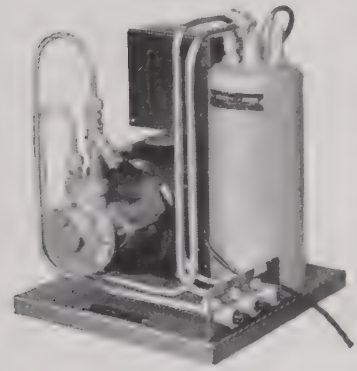
Instantaneous
WATER & BEVERAGE
COOLERS
CARBONATORS
REFRIGERATION ACCESSORIES
and VALVES



WATER COOLING — Drinking water coolers (instantaneous, storage and self-contained cabinet types) commercial and industrial water coolers for bakeries, bottling plants, factories, mills, stores, air conditioning applications, etc.

BEVERAGE COOLING — Draught beer coolers, Soda fountain coolers, Soft drink coolers, Vending machine coolers (both coin operated and manually operated).

MISCELLANEOUS COOLING—Chemical cooling units, Oil cooling units, Syrup cooling units.



CARBONATORS — Carbonators and combination water cooler and carbonators available for soda fountains, vending machines, taverns, etc. Gives maximum carbonation at lowest possible CO_2 gas pressure.

MODEL CB-305
COOLER-CARBONATOR

(Above) MODEL 26-S-2
BEVERAGE COOLER

(Below) MODEL PB-10
DRINKING WATER COOLER

REFRIGERATION ACCESSORIES — "Oilrite" oil separators, Adaptor blocks, Accumulator heat exchangers, Equalizer tanks, Beer and water faucets, Beer connecting tubes.

VALVES—Suction pressure control valves, two temperature valves, Float valves.

WRITE NOW FOR DESCRIPTIVE LITERATURE

TEMPRITE PRODUCTS CORPORATION

45 PIQUETTE • DETROIT 2, MICH.

Westinghouse Electric Corporation

Sturtevant Division

Air Conditioning, Heating, Ventilating, Dust Control and Fume Removal Equipment, Electronic Air Cleaners, Dryers, Compressors, Mechanical Draft Equipment

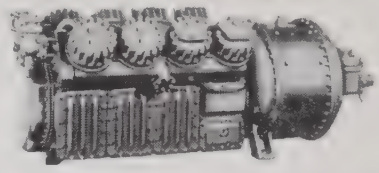
HYDE PARK **BOSTON 36, MASS.**

Westinghouse-Sturtevant Division manufactures a complete line of air conditioning equipment for industrial and commercial applications, for both comfort and process air conditioning. Offices and Distributors are located in principal cities. Refer to your local classified telephone directory. Additional product data on all equipment is available on request.

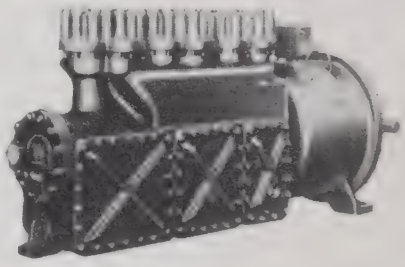
REFRIGERANT COMPRESSORS—Hermetically Sealed, Freon 12, Type CLS refrigerant compressors are designed for air conditioning and industrial refrigeration applications requiring suction evaporating temperatures ranging from 10°F. to 50°F.

The hermetically-sealed compressors are reciprocating multi-cylinder, in-line type in sizes 7½ to and including 60 horsepower; 90 degree V-type on the 2, 3, 5, 75 and 100 horsepower sizes. They are protected by manually reset high and low pressure switch. Driving motors are polyphase, refrigerant-cooled induction type. Motors are protected by thermal overload protection in the De-ion magnetic line-starters.

Mounting supports are available to permit mounting the compressor and suitable Westinghouse water-cooled condenser as a unit. Type CLS compressors are made in 12 sizes ranging in nominal capacities of from 2 to 100 tons refrigeration effect.

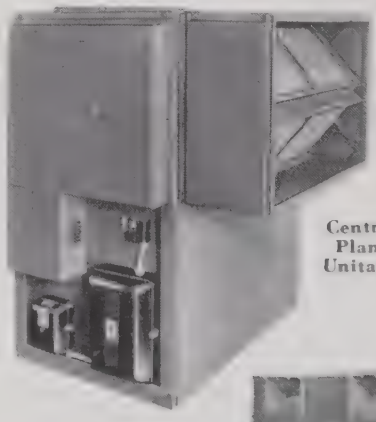


Sixteen-Cylinder V-Type Compressor



Six-Cylinder In-Line Type Compressor

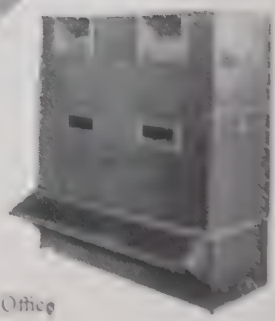
UNITAIRE* AIR CONDITIONING UNITS—Central-plant Type Unitaires are completely self-contained, with all the component parts on an air conditioning system, designed especially for installation with supply and return air ducts. Unitaires are completely assembled, piped, refrigerant charged, wired and adjusted at the factory. Each is given a complete operating test to assure satisfactory functioning when installed. All that is required is connection of water and electrical service and attachment to the duct system.



Central Plant Unitaire

SELF-CONTAINED TYPE UNITAIRES are widely used in small stores, restaurants, office suites and similar establishments. Usually they are installed in the room to be air conditioned, but in many cases they are remotely located with ducts supplying one or more spaces with conditioned air.

EVAPORATIVE CONDENSERS—Where water is scarce or expensive, or its use or disposal restricted, the "Aquamiser" provides savings in water consumption. Westinghouse Aquamisers are available to give a range of nominal capacities of 5 tons to 100 tons net refrigeration effect.

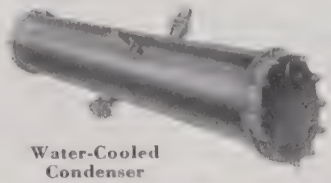


"Aquamiser" Evaporative Condenser

* Reg. U.S. Patent Office

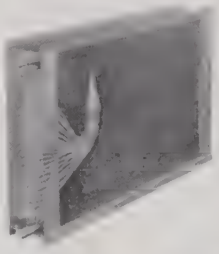
Westinghouse Electric Corporation

WATER COOLED CONDENSERS—Where sufficient water is available from city water supply or cooling tower, the water-cooled condenser is economical and efficient. To economically match refrigeration load requirements, Westinghouse water-cooled condensers are available in 14 sizes, ranging in nominal capacities from 2 to 100 tons net refrigeration effect.



Water-Cooled Condenser

WATER CHILLING UNITS—For the chilling of water for distribution to a number of cooling coils, for surface dehumidifiers and for use in industrial processes, these efficient water-chilling units range in size from 5 to 110 tons net refrigeration effect.



Heat Transfer Surface

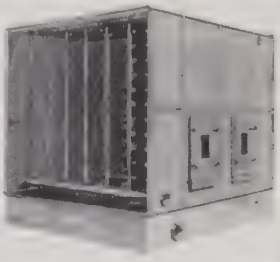
Water Chilling Unit

HEAT TRANSFER SURFACES—Direct expansion coils with patented refrigerant distributor, chilled-water coils, steam and hot water coils are available in a great many sizes for all types of systems.

AIR CONDITIONING UNITS—Two types are available—a horizontal type for ceiling suspension and a vertical type for floor installation. In the horizontal and vertical types, year-round air conditioning is accomplished by the installation of cooling and dehumidifying surfaces for summer, and heating surfaces for winter air conditioning.

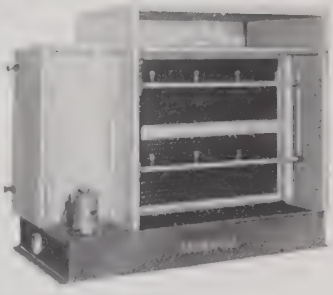
Air Conditioning Unit, Type AH

AIR WASHERS—Three types available—Type H, with one bank of nozzles, used primarily for humidifying; Type C, with two banks of nozzles, used for evaporative cooling, humidifying and dehumidifying; and Type S with one bank of nozzles, used for humidifying, especially where space is limited.



Air Washer

SURFACE DEHUMIDIFIERS—Available for either chilled water or directly expanded refrigerant, these wet surface units control accurately wet and dry bulb conditions in industrial and comfort air conditioning systems. Capacities range from 2000 to 50,000 cfm, with cooling capacities of 5 to 250 tons.



Unit Dehumidifier

AXIFLO PRESSURE FANS are available in either straight through or elbow types, with either three-bladed aluminum or 8-bladed steel wheels.

MULTIVANE CENTRIFUGAL FANS are high capacity, low speed fans suitable for use in public buildings because of low tip speed, low outlet velocity, and small size for given duty.

SILENTVANE CENTRIFUGAL FANS are high speed, high efficiency fans engineered for installations where noise cannot be tolerated. These fans have rising pressure and non-overloading characteristics.

Silentvane Centrifugal Fan, Design 6

Elbow Axiflo Pressure Fan

LIQUID LEVEL CONTROLS (See CONTROLS, LIQUID LEVEL; also HIGHSIDE FLOATS; also LOWSIDE FLOATS; also FLOATS; also FLOAT SWITCHES)

LIQUID LEVEL INDICATORS (See GAUGES, LIQUID LEVEL)

LITHIUM BROMIDE

Dow Chemical Co., Midland, Mich.

LITHIUM CHLORIDE

Mallinckrodt Chemical Wks., 2nd & Mallinckrodt St., St. Louis, Mo.

LOCK NUTS (See NUTS, LOCK)

LOCK WASHERS (See WASHERS, LOCK)

LOCKER PLANTS

Armstrong Cork Co., Lancaster, Pa. *(p. 136)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Frosty Foods Equip. Co., 305 Benson Blvd., Sioux City 15, Ia.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Locker Engrg. Co., 521 N. La Cienga Blvd., Los Angeles 36,Cal.
C Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
York Corp., York, Pa. *(p. 163)*

LOCKERS

All-Steel Equip., Inc., Kensington Ave., Aurora, Ill
Frick Co., Waynesboro, Pa. *(p. 47)*
Frosty Foods Equip. Co., 305 Benson Blvd., Sioux City 15, Ia.
Knickerbocker Stamping Co., Parkersburg, W. Va.
Locker Engrg. Co., 521 N. La Cienga Blvd., Los Angeles 36, Cal.
McGraw Mfg. Co., 2124 Y St., Lincoln 1, Neb.
Master Mfg. Corp., 119 Main St., Sioux City 4, Ia.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Safe-Way Food Locker Co., 175 W. Jackson Blvd., Chicago 4, Ill.

LOCKS

Corbin Cabinet Lock Div., New Britain, Ct.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Standard-Keil Hardware Mfg. Co., Inc., 2413 Atlantic Ave., Brooklyn 33, N.Y. *(p. 123)*

LOUVRED BAFFLES (See BAFFLES)

LOUVRES (See also SHUTTERS)

Arex Co., 333 N. Michigan Ave., Chicago 1, Ill.
Burt Mfg. Co., 926 S. High St., Akron 11, O.
E. K. Campbell Heating Co., 1809 Manchester St., Kansas City 3, Mo.
Diamond Mfg. Co., Wyoming, Pa.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7, Ill.
Lockjoint Wood Products Co., 1721 Mildred Ave., Wichita 7, Kan.
Herman Nelson Corp., 1824-3rd Ave., Moline, Ill.

LOW TEMPERATURE CABINETS (See HOME & FARM FREEZERS; also CABINETS, CONSTANT TEMPERATURE)

LOWSIDE EQUIPMENT (See particular type)

LOWSIDE FLOATS (See also FLOAT SWITCHES) (A—Ammonia; B—Other refrigerants)

(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(B) Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
(A,B) King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 158)*
(A,B) H. A. Phillips & Co., 3255 W. Carroll Ave., Chicago 24, Ill. *(p. 114)*
(A, B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A,B) York Corp., York, Pa. *(p. 163)*

LUBRICANTS (See GREASES, OILS)

LUBRICATORS & LUBRICATING SYSTEMS (See also GREASE CUPS)

Bowser, Inc., 1302 E. Creighton Ave., Ft. Wayne 2, Ind.
Frick Co., Waynesboro, Pa. *(p. 47)*
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Manzel Bros. Co., Div. of Frontier Industries, Inc., 309 Babcock St., Buffalo 10, N.Y.
Nathan Mfg. Co., 416 E. 106th St., N.Y.C. 29
Wm. W. Nugent & Co., Inc., 410 Hermitage Ave., Chicago 22, Ill.
Ohio Grease Co., N. Spring St., Loudonville, O.
Oil Rite Corp., 3466 S. 13th St., Milwaukee 7, Wis.
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y

LUGS, TANKS

Arrow Tank Co., Inc., 16 Barnett St., Buffalo 15, N.Y.
Lakeside Malleable Castings Co., Racine, Wis.
Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Angeles 11, Cal. *(p. 80)*

MACHINE WORK, CONTRACT (See also SCREW MACHINE PARTS, etc.)

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia.
Eastern Machine Screw Corp., Truman & Barclay Sts., New Haven 6, Ct.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Magnus Brass Mfg. Co., 533 Reading Rd., Cin'ti. 2, O.
Newman Bros., Inc., 666 W. 4th St., Cin'ti. 3, O.
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
Ross Sprinkler Co., 34 Roberts St., Pasadena 3, Cal.

MAGNESIUM CHLORIDE

Dow Chemical Co., Midland, Mich.

MAGNETIC VALVES (See SOLENOID VALVES)

MANIFOLDS

Frick Co., Waynesboro, Pa. *(p. 47)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Mueller Brass Co., Port Huron, Mich.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

MANOMETERS

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Hays Corp., Michigan City, Ind.
Meriam Instrument Co., 10920 Madison Ave., Cleveland 2, O.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*

MARGARINE CHILLERS (See LARD CHILLERS)

MATERIAL HANDLING EQUIPMENT (See CONVEYORS)

MEDALLIONS (See NAME PLATES)

MEMBRANES (See VAPOR SEALS)

MESH, METAL (See also WIRE CLOTH)

Audubon Wire Cloth Corp., Richmond St. & Castor Ave., Phila. 34, Pa.
Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
Cyclone Fence Div., American Steel & Wire Co., U. S Steel Corp. Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Kentucky Metal Products Co., Preston St. & Audubon Park, Louisville 4, Ky.
Newark Wire Cloth, 351 Verona Ave., Newark 4, N.J.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
Wheeling Corrugating Co., Wheeling, W. Va.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

METAL SPECIALTIES (See also particular item)

Alpha Metals, Inc., 363 Hudson Ave., Brooklyn 1, N.Y
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.

METAL SPINNING

L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Superior Spinning & Stamping Co., 4057 Fitch Rd., Toledo 12, O.

METAL STAMPING (See STAMPINGS)

METAL, BEARING (See BABBITT, BRONZE, etc.)

METAL, LAMINATED

General Plate Div., Metals & Controls Corp., Attleboro, Mass.

METAL, PERFORATED

Bristol Brass Corp., Bristol, Ct.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Chicago Perforating Co., 2445 W. 24th Place, Chicago 8, Ill.
Diamond Mfg. Co., Wyoming, Pa.
McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
Manhattan Perforated Metal Co., Inc., 43-17-37th St., Long Island City 1, N.Y.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.

METAL, THERMOSTATIC (See THERMOSTATIC BI-METALS)

METERS, BRINE

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

METERS, COIN OPERATED

General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
International Register Co., 2620 W. Washington Blvd., Chicago 12, Ill.

METERS, ELECTRIC

AC Spark Plug Div., Gen'l. Motors Corp., Flint 2, Mich.
Bristol Co., Waterbury 91, Ct. (Recording)
Fischer & Porter Co., Hatboro, Pa. (Flow Rate)
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Roller-Smith Div., Realty & Industrial Corp., Bethlehem, Pa.
Sangamo Elec. Co., Springfield, Ill.
Westinghouse Elec. Corp., Plane & Orange Sts., Newark 1, N.J.
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

METERS, FLOW

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Barton Instrument Co., 3500 Union Pacific Ave., Los Angeles 23, Cal.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Fischer & Porter Co., Hatboro, Pa.
Hays Corp., Michigan City, Ind.
Meriam Instrument Co., 10920 Madison Ave., Cleveland 2, O.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Pittsburgh Equitable Meter Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*

METERS, RECORDING

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Fischer & Porter Co., Hatboro, Pa. (Flow Rate)
Hays Corp., Michigan City, Ind.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Roller-Smith Div., Realty & Industrial Corp., Bethlehem, Pa.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Westinghouse Elec. Corp., Plane & Orange Sts., Newark 1, N.J.

METERS, REFRIGERANT

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Fischer & Porter Co., Hatboro, Pa.

METERS, RUNNING TIME

Bristol Co., Waterbury 91, Ct.
R. W. Cramer Co., Centerbrook, Ct.
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

METERS, VENTURI

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.

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Eston
REFRIGERANTS
ARE BETTER!

Distributors of
FREON 11-12-22-113

Eston
CHEMICALS, Inc.
3100 East 26th Street
Los Angeles 23, Calif.

Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. (Water) *(p. 76)*

METHYL CHLORIDE

Ansul Chemical Co., Marinette, Wis. *(p. 175)*
Dow Chemical Co., Midland, Mich.
Electrochemicals Dept., E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del. *(p. 153)*
Eston Chemicals, Inc., 3100 E. 26th St., Los Angeles 23, Cal. *(p. 152)*
Great Western Div., Dow Chemical Co., 310 Sansome St., San Francisco 4, Cal.
Virginia Smelting Co., W. Norfolk, Va. *(p. 177)*

METHYLENE CHLORIDE

Dow Chemical Co., Midland, Mich.
Electrochemicals Dept., E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del. *(p. 153)*
Eston Chemicals, Inc., 3100 E. 26th St., Los Angeles 23, Cal. *(p. 152)*
Virginia Smelting Co., W. Norfolk, Va. *(p. 177)*

MILK COOLERS

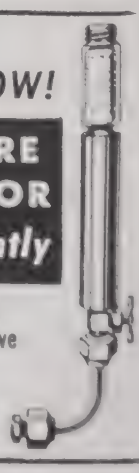
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Central Supply Co., 210 S. Capitol Ave., Indpls., Ind.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Esco Cabinet Co., West Chester, Pa.
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
International Harvester Co., 180 N. Michigan Ave., Chicago 1, Ill.
Jensen Machinery Co., Inc., Nelson St. at Locust Ave., Bloomfield, N.J. (Dairies) *(p. 241)*
La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
Master-Bilt Refrigeration Mfg. Co., 920 Palm St., St Louis 7, Mo.
National Cooperatives, Inc., 343 S. Dearborn St., Chicago 4, Ill.
W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
Schultz Bros., Inc., 2130 Midland Rd., Saginaw, Mich.
Emil Steinhorst & Sons, Inc., 612 South St., Utica 3, N.Y.
Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.

MIRRORS

Semon Bache & Co., 636 Greenwich St., N.Y.C. 14

Is the System Wet?
Don't guess—KNOW!

DFN MOISTURE INDICATOR *tells instantly*



For Freon and Methyl Chloride.
Accurate . . . fast . . . inexpensive
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McIntire Connector Co.
252 Jefferson St., Newark, N. J.

For YOUR Refrigeration Needs

Artic

DU PONT METHYL CHLORIDE

99.5% Pure · Dry · Uniform

DU PONT METHYL CHLORIDE SPECIFICATIONS

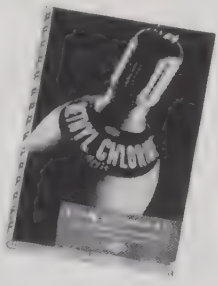
Purity . . . . . 99.5% Methyl Chloride
Moisture . . . . 0.008% by wgt. max.
Acid as (HCl). . 0.001% by wgt. max.
Residue on Evaporation 0.01% by wgt. max.
Boiling Range (760 mm) 24.6° to -23.6° C.
Color . . . . . water white, clear

District Sales Offices: Electrochemicals Dept., Baltimore, Boston, Charlotte, Chicago, Cincinnati, Cleveland, Detroit, El Monte (Calif.), New York, Philadelphia, Pittsburgh, San Francisco; And Ammonia Dept.: Offices in New York, Philadelphia, Chicago, St. Louis.

ADVANTAGES: Small volume displacement per unit of refrigeration . . . Non-corrosive to ordinary equipment . . . High operating efficiency . . . Easily handled and serviced . . . Operates at positive low side pressures . . . Provides quick cooling . . . Permits steady control.

AVAILABLE: Standard cylinders: 70, 100, 145 and 300-lbs. net, 1300-lb. tanks in multi-unit tank cars and single unit tank cars, 40,000 or 77,000 lbs. Smaller containers from distributors.

SEND FOR THIS NEW BOOK. 72 pages of information for engineers, designers, builders of refrigeration equipment. Write to district office. E. I. du Pont de Nemours & Co. (Inc.) Electrochemicals Dept., Wilmington 98, Del.



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DU PONT ELECTROCHEMICALS

MOISTURE INDICATORS

McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 152)*

MOLDED PRODUCTS (See particular type)

MONEL (See also mill forms, i.e., SHEET, etc.)

International Nickel Co., 67 Wall St., N.Y.C. 5

MONORAILS (See CONVEYORS, MONORAIL)

MORTUARY REFRIGERATORS (See REFRIGERATORS, MORTUARY)

MOTOR-GENERATORS

Century Elec. Co., 1806 Pine St., St. Louis 3, Mo.
Crocker-Wheeler Div., Joshua Hendy Iron Wks., Ampere, N.J.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Holtzer-Cabot Div., First Industrial Corp., 129 Armory St., Boston 17, Mass.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Master Elec. Co., 126 Davis Ave., Dayton 1, O.
Star Elec. Motors Co., 200 Bloomfield Ave., Bloomfield, N.J.
U. S. Galvanizing & Plating Equip. Corp., 27 Heyward St., Brooklyn 11, N.Y.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

MOTOR BASES, BRACKETS & SUPPORTS (See BASES, MOTOR)

MOTOR PROTECTORS

Spencer Thermostat Co., Unit of Metals & Controls Corp., 34 Forest St., Attleboro, Mass.

MOTOR STARTERS (See STARTERS)

MOTORS, A.C., POLYPHASE

Louis Allis Co., 427 E. Stewart St., Milwaukee 7, Wis.
Allis-Chalmers Mfg., Co., Milwaukee 1, Wis.
Baldor Elec. Co., 4353 Duncan Ave., St. Louis 15, Mo.
Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Bodine Elec. Co., 2254 W. Ohio St., Chicago 12, Ill.
Burke Elec. Co., Erie, Pa.
Century Elec. Co., 1806 Pine St., St. Louis 3, Mo.
Cleveland Elec. Motor Co., 5213 Chester Ave., Cleveland 3, O.
Continental Elec. Co., Inc., 325 Ferry St., Newark 5, N.J.
R. W. Cramer Co., Centerbrook, Ct.
Crocker-Wheeler Div., Joshua Hendy Iron Wks., Ampere, N.J.
Delco Products Div., Gen'l. Motors Corp., 329 E. 1st St., Dayton, O.
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
Eastern Air Devices, Inc., 130 Flatbush Ave., Brooklyn, N.Y.
Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.
Elliott Co., Jeannette, Pa.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Holtzer-Cabot Div., First Industrial Corp., 125 Armory St., Boston 17, Mass.
Howell Elec. Motors Co., 409 N. Roosevelt, Howell, Mich.
Ideal Elec. & Mfg. Co., Mansfield, O.
Imperial Elec. Co., 84 Ira St., Akron, O.
Kimble Elec., Div. of Miehle Printing Press & Mfg. Co., 2850 Mt. Pleasant St., Burlington, Ia.
Kingston-Conley Elec. Co., 86 Brook Ave., N. Plainfield, N.J.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Marathon Elec. Mfg. Corp., Wausau, Wis.
Marble-Card Elec. Co., Gladstone, Mich.
Master Elec. Co., 126 Davis Ave., Dayton 1, O.
Peerless Elec. Co., 2000 W. Market St., Warren, O.
Reliance Elec. & Engrg. Co., 1088 Ivanhoe Rd., Cleveland 10, O.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Robbins & Myers, Inc., Springfield, O.
Small Motors, Inc., 2068 Elston Ave., Chicago 14, Ill.
Star Elec. Motors Co., 200 Bloomfield Ave., Bloomfield, N.J.
Sterling Elec. Motors, Inc., 5401 Anaheim-Telegraph Rd., Los Angeles 22, Cal.
U. S. Elec'l. Motors, Inc., Los Angeles 54, Cal. & Milford, Ct.
Wagner Elec. Corp., 6400 Plymouth Ave., St. Louis 14, Mo.
Watson Flagg Machine Co., 845 E. 25th St., Paterson 3, N.J.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.

MOTORS, A.C., SINGLE PHASE

Air-Way Elec. Appliance Corp., 2101 Auburn Ave., Toledo 1, O.
Baldor Elec. Co., 4353 Duncan Ave., St. Louis 15, Mo.
Bodine Elec. Co., 2254 W. Ohio St., Chicago 12, Ill.
Century Elec. Co., 1806 Pine St., St. Louis 3, Mo.
Delco Products Div., Gen'l. Motors Corp., 329 E. 1st St., Dayton, O.
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
Eastern Air Devices, Inc., 130 Flatbush Ave., Brooklyn, N.Y.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Holtzer-Cabot Div., First Industrial Corp., 125 Armory St., Boston 17, Mass.
Jack & Heintz Precision Industries, Inc., Cleveland 1, O.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Kimble Elec., Div. of Miehle Printing Press & Mfg. Co., 2850 Mt. Pleasant St., Burlington, Ia.
Kingston-Conley Elec. Co., 86 Brook Ave., N. Plainfield, N.J.
Marathon Elec. Mfg. Corp., Wausau, Wis.
Master Elec. Co., 126 Davis Ave., Dayton 1, O.
Ohio Elec. Mfg. Co., Bellford Ave., Cleveland 4, O.
Peerless Elec. Co., 2000 W. Market St., Warren, O.
Raytheon Mfg. Co., Div., Russell Elec. Co., 340 W. Huron St., Chicago 10, Ill.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Robbins & Myers, Inc., Springfield, O.
Small Motors, Inc., 2068 Elston Ave., Chicago 14, Ill.
F. A. Smith Mfg. Co., Inc., Union & Augusta Sts., Rochester 2, N.Y.
Victor Elec. Products, Inc., 2950 Roberston Rd., Cin'ti. 9, O.
Wagner Elec. Corp., 6400 Plymouth Ave., St. Louis 14, Mo.
Westinghouse Elec Corp., Lima, O.

MOTORS, CONTROL, INSTRUMENT, MINIATURE, etc.

Barber-Colman Co., Rockford, Ill.
Bodine Elec. Co., 2254 W. Ohio St., Chicago 12, Ill.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
Eastern Air Devices, Inc., 130 Flatbush Ave., Brooklyn, N.Y.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Master Elec. Co., 126 Davis Ave., Dayton 1, O.
Raytheon Mfg. Co., Div., Russell Elec. Co. 340 W. Huron St., Chicago 10, Ill.
Redmond Co., Inc., Owosso, Mich.
Signal Elec. Mfg. Co., Menominee, Mich.
Small Motors, Inc., 2068 Elston Ave., Chicago 14, Ill.
F. A. Smith Mfg. Co., Inc., Union & Augusta Sts., Rochester 2, N.Y.

MOTORS, D.C.

Air-Way Elec. Appliance Corp., 2101 Auburn Ave., Toledo 1, O.
Louis Allis Co., 427 E. Stewart St., Milwaukee 7, Wis.
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Baldor Elec. Co., 4353 Duncan Ave., St. Louis 15, Mo.
Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Bodine Elec. Co., 2254 W. Ohio St., Chicago 12, Ill.
Burke Elec. Co., Erie, Pa.
Century Elec. Co., 1806 Pine St., St. Louis 3, Mo.
Continental Elec. Co., Inc., 325 Ferry St., Newark 5, N.J.

Crocker-Wheeler Div., Joshua Henry Iron Wks., Ampere, N.J.
Delco Products Div., Gen'l. Motors Corp., 329 E. 1st St., Dayton, O.
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
Eastern Air Devices, Inc., 130 Flatbush Ave., Brooklyn, N.Y.
Elliott Co., Jeanette, Pa.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Holtzer-Cabot Div., First Industrial Corp., 125 Armory St., Boston 17, Mass.
Ideal Elec. & Mfg. Co., Mansfield, O.
Imperial Elec. Co., 84 Ira St., Akron, O.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Kimble Elec., Div. of Miehle Printing Press & Mfg. Co., 2850 Mt. Pleasant St., Burlington, Ia.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Marathon Elec. Mfg. Corp., Wausau, Wis.
Marble-Card Elec. Co., Gladstone, Mich.
Master Elec. Co., 126 Davis Ave., Dayton 1, O.
National Engrg. & Mfg. Co., 213 W. 19th St., Kansas City 8, Mo.
Peerless Elec Co., 2000 W. Market St., Warren, O.
Reliance Elec. & Engrg. Co., 1088 Ivanhoe Rd., Cleveland 10, O.
Robbins & Myers, Inc., Springfield, O.
Small Motors, Inc., 2068 Elston Ave., Chicago 14, Ill.
Star Elec. Motors Co., 200 Bloomfield Ave., Bloomfield, N.J.
Sterling Elec. Motors, Inc., 5401 Anaheim-Telegraph Rd., Los Angeles 22, Cal.
U. S. Elec'l. Motors, Inc., Los Angeles 54, Cal. & Milford, Ct.
Wagner Elec. Corp., 6400 Plymouth Ave., St. Louis 14, Mo.
Westinghouse Elec. Corp., Lima, O.

MOTORS, DAMPER (See DAMPER MOTORS)

MOTORS, GEARHEAD

Louis Allis Co., 427 E. Stewart St., Milwaukee 7, Wis.
Bodine Elec. Co., 2554 W. Ohio St., Chicago 12, Ill.
Century Elec. Co., 1806 Pine St., St. Louis 3, Mo.
Cleveland Elec. Motor Co., 5213 Chester Ave., Cleveland 3, O.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Howell Elec. Motors Co., 409 N Roosevelt, Howell, Mich.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Link-Belt Co., 2045 W Hunting Park, Phila. 40, Pa.
Master Elec. Co., 126 Davis Ave., Dayton 1, O.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Phila. 20, Pa.
Star Elec. Motors Co., 200 Bloomfield Ave., Bloomfield, N.J.
Sterling Elec. Motors, Inc., 5401 Anaheim-Telegraph Rd., Los Angeles 22, Cal.
Watson-Flagg Machine Co., 845 E. 25th St., Paterson 3, N.J.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

MOTORS FOR HERMETIC COMPRESSORS

Louis Allis Co., 427 E. Stewart St., Milwaukee 7, Wis.
Century Elec. Co., 1806 Pine St., St. Louis 3, Mo.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Westinghouse Elec. Corp., Lima, O.

MOTORS, HIGH FREQUENCY

Louis Allis Co., 427 E. Stewart St., Milwaukee 7, Wis.
Eastern Air Devices, Inc., 130 Flatbush Ave., Brooklyn N.Y.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Onsrud Machine Wks., Inc., 3900 Palmer St., Chicago 47, Ill.

MOTORS, SPECIAL

Louis Allis Co., 427 E. Stewart St., Milwaukee 7, Wis.
Baldor Elec. Co., 4353 Duncan Ave., St. Louis 15, Mo.
Bodine Elec. Co., 2254 W. Ohio St., Chicago 12, Ill.
Burke Elec. Co., Erie, Pa.
Cleveland Elec. Motor Co., 5213 Chester Ave., Cleveland 3, O.
Continental Elec. Co., Inc., 325 Ferry St., Newark 5, N.J.
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Holtzer-Cabot Div., First Industrial Corp., 125 Armory St., Boston 17, Mass.
Howell Elec. Motors Co., 409 N. Roosevelt, Howell, Mich.
Ideal Elec. & Mfg. Co., Mansfield, O.
Imperial Elec. Co., 84 Ira St., Akron, O.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Kimble Elec., Div. of Miehle Printing Press & Mfg. Co., 2850 Mt. Pleasant St., Burlington, Ia.
Marathon Elec. Mfg. Corp., Wausau, Wis.
Marble-Card Elec. Co., Gladstone, Mich.
Master Elec. Co., 126 Davis Ave., Dayton 1, O.
Moore Co., 544 Westport Rd., Kansas City 2, Mo. (Slow speed)
Peerless Elec. Co., 2000 W. Market St., Warren, O.
Raytheon Mfg. Co., Div., Russell Elec. Co., 340 W. Huron St., Chicago 10, Ill.
Robbins & Myers, Inc., Springfield, O.
Signal Elec. Mfg. Co., Menominee, Mich.
Small Motors, Inc., 2068 Elston Ave., Chicago 14, Ill.
Star Elec. Motors Co., 200 Bloomfield Ave., Bloomfield, N.J.
Wagner Elec. Corp., 6400 Plymouth Ave., St. Louis 14, Mo.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.

MOTORS, UNIVERSAL

Bodine Elec. Co., 2254 W. Ohio St., Chicago 12, Ill.
Diehl Mfg. Co., 1152 Finderne Ave., Somerville, N.J.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Signal Elec. Mfg. Co., Menominee, Mich.
Small Motors, Inc., 2068 Elston Ave., Chicago 14, Ill.
Westinghouse Elec. Corp., Lima, O.

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325 W. Huron St. Chicago 10, Ill.
766 S. Los Angeles St., Los Angeles 14, Calif.

MOULDINGS

Brasco Mfg. Co., Harvey, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich. (Plastic)
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber & Plastic)
R. D. Werner Co., Inc., 295-5th Ave., N.Y.C. 16

MOUNTINGS, VIBRATION ABSORBING (See VIBRATION ABSORBING BASES)

MUFFLERS, PIPE LINE

Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*

NAILS

Aluminum Co. of America, Pittsburgh 19, Pa.
American Rolling Mill Co., Middletown, O. (Stainless Steel)
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Bethlehem Steel Co., Bethlehem, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Il
International Nickel Co., 67 Wall St., N.Y.C. 5 (Nickel Alloy)
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18
Alan Wood Steel Co., Steel Mill Rd., Conshohocken, Pa. (Cut)
Youngstown Sheet & Tube Co., Youngstown, O.

NAME PLATES

Acromark Co., 5 Morrell St., Elizabeth 4, N.J.
American Emblem Co., Inc., 9 Genesee St., New Hartford, Utica 1, N.Y.
Croname, Inc., 3701 N. Ravenswood, Chicago 13, Ill.
Fox Co., Fox Lane, Cin'ti. 23, O.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Hoosier Cardinal Corp., 601 W. Eichel Ave., Evansville 7, Ind. (Plastic only)
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Geo. J. Mayer Co., 15 N. Pennsylvania St., Indpls. 9, Ind.
A.B. Murray, Inc., 604 Green Lane, Elizabeth, N.J.
Newman Bros., Inc., 666 W. 4th St., Cin'ti. 3, O.
Sneath Glass Co., Hartford City, Ind.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.

NEEDLE VALVE ASSEMBLIES

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
Jerguson Gage & Valve, 87 Fellsway, Somervile, Mass.
Mayson Mfg. Co., Inc., 4332 Horatio St., Detroit 10, Mich.

NEEDLES, VALVE

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Illl.
Mayson Mfg. Co., Inc., 4332 Horatio St., Detroit 10, Mich.

NEGATIVE TEMPERATURE COEFFICIENT MATERIAL

Keystone Carbon Co., Inc., 1935 State St., St. Marys, Pa.

NETTING, WIRE (See MESH, METAL)

NICKEL (See also mill forms, i.e., ROD; also SHEET, etc.)

Haynes Stellite Co., Unit. of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C. 5

NIPPLES, PIPE (See also FITTINGS)

Dresser Mfg. Div., Dresser Industries, Inc., 490 Fisher Ave., Bradford, Pa.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 113)*

NITROGEN

Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17 *(p. 197)*

NOSES, SEAL

American Crucible Products Co., Lorain, O.
Chicago Seal Co., 232 S. Hoyne Ave., Chicago 20, Ill.
Federal-Mogul Corp., Shoemaker & Lillibridge Sts., Detroit 13, Mich.

NOZZLES

April Showers Co., 4126-8th St., N.W., Washington 11, D.C. *(p. 186)*
Binks Mfg. Co., 3114 Carroll Ave., Chicago 12, Ill.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Clarage Fan Co., Porter St., Kalamazoo 16, Mich.
Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia.
Fluor Corp., Ltd., 2500 S. Atlantic Blvd., Los Angeles 22, Cal.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. *(p. 79)*
Jos. A. Martocello & Co., 229 N. 14th St., Phila. 7, Pa. *(p. 157)*
Monarch Mfg. Wks., Inc., Salmon & Westmorland Sts., Phila. 34, Pa.
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Phillips Cooling Tower Co., Inc., 114 Liberty St., N.Y.C. 6
J. F. Pritchard & Co., 2200 Fidelity Bldg., Kansas City 6, Kan. *(p. 81)*
Rega Mfg. Co., 79 Mt. Hope Ave., Rochester, N.Y.
Ross Sprinkler Co., 34 Roberts St., Pasadena 3, Cal.
Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Angeles 11, Cal. *(p. 80)*
Spray Engrg. Co., 114 Central St., Somerville 45, Mass.
Spraying Systems Co., 4021 W. Lake St., Chicago 24, Ill.
Supreme Elec. Products Co., 194 Vassar St., Rochester 7, N.Y.
Thermal Industries, P.O. Box 725, Indio, Cal.
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
Water Cooling Equip. Corp., Afton Sta., St. Louis 23, Mo.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
Yarnall-Waring Co., Chestnut Hill, Phila. 18, Pa.

NUTS

Aircraft Screw Products Co., Inc., 47-23-35th St., Long Island City 1, N.Y.
W. Ames & Co., Jersey City, N.J.
Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 55)*
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.

(Continued)

Jos. A. Martocello & Company

Consult Index for Complete Line

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Clear Ice Making Systems & Refrigerating Supplies

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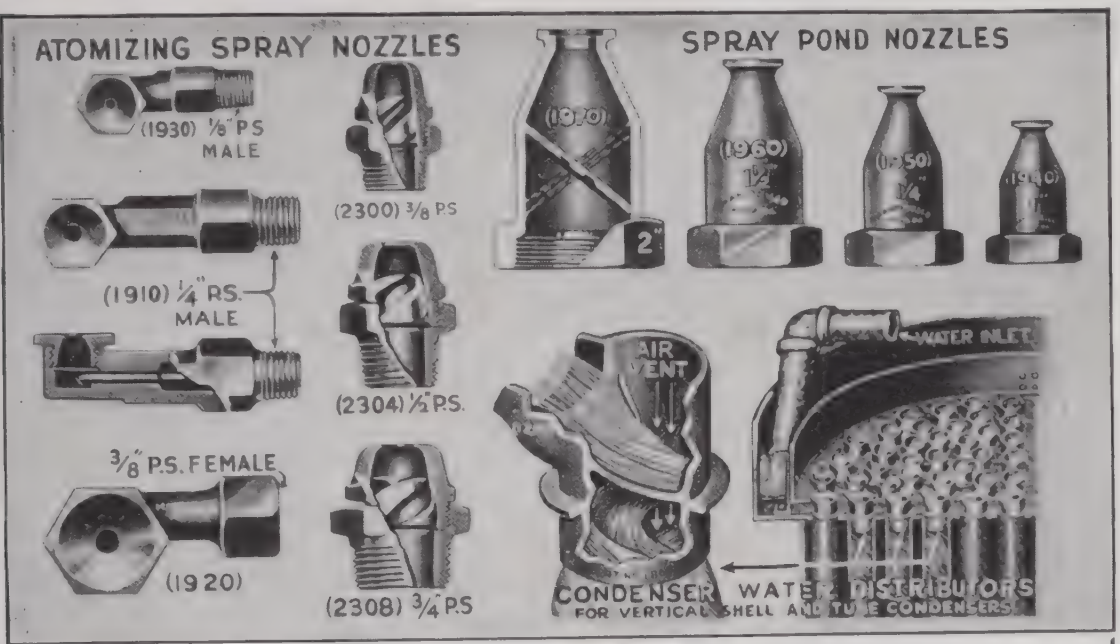

ATOMIZING SPRAY NOZZLES

MARTOCELLO SPRAY NOZZLES produce a uniform, wide spray with less friction and at a minimum pressure. They are manufactured with precision and of a design which has been thoroughly tested for maximum results and durability. They are guaranteed to give satisfaction. Successful, efficient results depend largely upon selecting the proper number, type and size of Nozzles suitable for your installation. We prefer, when possible, to figure from specifications to recommend the proper Nozzles (standard or special), complete piping or materials that may be required.

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MARTOCELLO SPRAY POND NOZZLES are of ONE PIECE CONSTRUCTION, cast of sturdy High Grade Red Brass, with inlet and outlet accurately machined, offer LESS FRICTION and are LESS CLOGGING than any other design, and therefore will provide long lasting, best OVERALL EFFICIENCY.

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NUTS (Continued)

Clark Bros. Bolt Co., Milldale, Ct.
Columbus Bolt Wks. Co., 291 Marconi Blvd., Columbus 16, O.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Wm. R. Haskell Mfg. Co., 24 Commerce St., Pawtucket, R.I.
Haynes Stellite Co., Unit. of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C. 5
Lakeside Malleable Castings Co., Racine, Wis.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Radio Frequency Labs., Inc., Boonton, N.J.
Republic Steel Corp. Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Tinnerman Products, Inc., 2038 Fulton Rd., Cleveland 13, O

NUTS, LOCK

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Clark Bros. Bolt Co., Milldale, Ct.
Elastic Stop Nut Corp. of America, 2330 Vauxhall Rd., Union, N.J.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Laminated Shim Co., Inc., Union St., Glenbrook, Ct.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Thompson-Bremer & Co., 1640 W. Hubbard St., Chicago 22, Ill.
Tinnerman Products, Inc., 2038 Fulton Rd., Cleveland 13, O.
Walworth Co., 60 E. 42nd St., N.Y.C. 17

NUTS, MACHINE

Bethlehem Steel Co., Bethlehem, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Corbin Screw Corp., New Britain, Ct.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Wm. H. Haskell Mfg. Co., 24 Commerce St., Pawtucket, R.I.
Laminated Shim Co., Inc., Union St., Glenbrook, Ct.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

NUTS, STOVE

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Corbin Screw Corp., New Britain, Ct.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

NUTS, WELD

Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

ODOR ELIMINATION UNITS & SYSTEMS (See also AIR PURIFICATION SYSTEMS)

W. B. Connor Engrg. Corp., 144 E. 32nd St., N.Y.C. 16 *(p. 20)*
Electroaire Corp., 41-38-37th St., Long Island City 1, N.Y.
Jensen Machinery Co., Inc., Nelson St. at Locust Ave., Bloomfield, N.J. (Dairies) *(p. 241)*
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
Liquid Conditioning Corp., 114 E. Price St., Linden, N.J.
Ohio Carbon Co., 12508 Berea Rd., Cleveland 11, O.
Payton Co., 546 W. Washington Blvd., Chicago 6, Ill.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Sperti, Inc., Norwood Sta., Cin'ti. 12, O.
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Tuco Products Corp., 30 Church St., N.Y.C. 7
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

OIL COOLERS & OIL COOLING SYSTEMS (See also COOLANT COOLERS)

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Frostrode Products, Inc., 19929 Exeter Rd., Detroit 3, Mich.
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Reliance Refrigerating Machine Co., 3401 N. Kedzie Ave., Chicago 18, Ill.
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

OIL ENGINES See **ENGINES, DIESEL)**

OIL SEPARATORS

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 124)*
Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
American District Steam Co., Bryant St., N. Tonawanda, N.Y.

(Continued)

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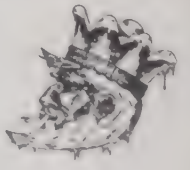

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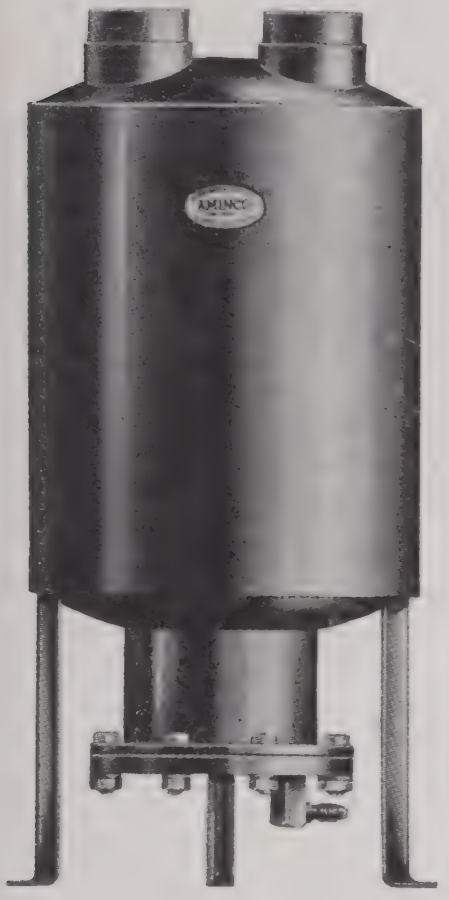

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Starting Load Regulating Valves

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14544 Third Avenue, Detroit 3, Michigan

(Manufacturing the former American Injector Co. line)

OIL SEPARATORS (Continued)

Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 159)*
Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Cochrane Corp., 17th St., below Allegheny Ave., Phila. 32, Pa.
Johnson Corp., 805 Wood St., Three Rivers, Mich.
King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 158)*
Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill. (Small)
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.
York Corp., York, Pa. *(p. 163)*

OILERS & OILING SYSTEMS (See LUBRICATORS)

OILS, LUBRICATING

Arkansas Fuel Oil Co., Div. of Cities Service Oil Co., Slattery Bldg., Shreveport, La.
Black Bear Co., Inc., 44-23rd St., Long Island City 1, N.Y.
Borne Scrymser Co., P.O. Box 256, Elizabeth, N.J.
Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Cities Service Oil Co. (Del.), 919 N. Michigan Ave., Chicago, Ill.
Cities Service Oil Co. (Pa.), 60 Wall Tower, N.Y.C. 5
Frick Co., Waynesboro, Pa. *(p. 47)*
Gulf Oil Corp., Gulf Bldg., Pittsburgh, Pa.
E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
E. H. Kellogg & Co., 93 Water St., N.Y.C. 5
Lubriplate Div., Fiske Bros. Refining Co., 129 Lockwood St., Newark, N.J.
Macmillan Petroleum Corp., 530 W. 6th St., Los Angeles 14, Cal.
New York & New Jersey Lubricant Co., 292 Madison Ave., N.Y.C. 17
Ohio Grease Co., N. Spring St., Loudonville, O.
Shell Oil Co., Inc., 50 W. 50th St., N.Y.C. 20
Sinclair Refining Co., 630-5th Ave., N.Y.C. 20
L. Sonneborn Sons, Inc., 88 Lexington Ave., N.Y.C. 16
Standard Oil Co. of California, 225 Bush St., San Francisco 20, Cal.
D. A. Stuart Oil Co., Ltd., 2727 S. Troy St., Chicago 23, Ill.
Sun Oil Co., 1608 Walnut St., Phila. 3, Pa. *(p. 161)*
Texas Co., 135 E. 42nd St., N.Y.C. 17
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

OZONE EQUIPMENT

Electroaire Corp., 41-38-37th St., Long Island City 1, N.Y.
General Ozone Corp., 1455 W. Congress St., Chicago 7, Ill.
Norwood Filtration Co , Northampton, Mass.
Scottdale Ozone Co., Scottdale, Pa.

PACKING, SHAFT, VALVE, etc.

American Metallic Packing Co., 3621 Mexico St., Pittsburgh 12, Pa.
Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Austin Mason Co., 15 Park Row, N.Y.C.
Belmont Packing & Rubber Co., Butler & Sepviva Sts., Phila. 7, Pa.
Buffalo Weaving & Belting Co., 262 Chandler St., Buffalo 7, N.Y.
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Chicago Belting Co., 113 N. Green St., Chicago 7, Ill.
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
C. Lee Cook Mfg. Co., 916 S. 8th St., Louisville 3, Ky.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Crane Packing Co., 1800 Cuyler Ave., Chicago 13, Ill.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del. (Plastic)
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Durametallic Corp, 2104 Factory St., Kalamazoo 24F, Mich.
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Endura Mfg. Corp., 45 N. 4th St., Quakertown, Pa.
Ernst Water Column & Gauge Co., 250 S. Livingston Ave., Livingston, N.J.
Eureka Packing Co., 248-46th St., Brooklyn 20, N.Y.
Excelsior Leather Washer Mfg. Co., Inc., Rockford, Ill.
Felt Products Mfg Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Fisher Leather Belting Co., Inc., 325 N. 3rd St., Phila. 6, Pa.
France Packing Co., Tacony, Phila. 35, Pa.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Hollow Center Packing Co., 6523 Euclid Ave., Cleveland 3, O.
E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
Janos Asbestos Co., 39 Cortlandt St., N.Y.C. 7
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Mabbs Hydraulic Packing Co., 431 S. Dearborn St., Chicago 5, Ill.
Marine Div., L. Katzenstein & Co., 358 West St., N.Y.C. 14
Milwaukee Leather Belting Co., 1114 N. Water St., Milwaukee 2, Wis.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo. (Tools
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
National Motor Bearing Co., Inc., Redwood City, Cal.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Horace G. Preston Co., 2581 Beecher Ave., Detroit 16, Mich.
Q-P Mfg. Co., Inc., 205 Rosemary St., Needham Heights, Mass.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Rains Wood Metal Packings, Spokane 5, Wash.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
J. E. Rhoads & Sons, 35 N. 6th St., Phila. 6, Pa.
Rhopac, Inc., 168 N. Clinton St., Chicago 6, Ill.
Rodpak Mfg. Co., 1315 Natoma St., San Francisco 3, Cal.
Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7
U. S. Graphite Co., Saginaw, Mich.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.
John M. Watt's Sons, 112 Walnut St., Phila. Pa.
York Corp., York, Pa. *(p. 163)*

PACKING, SHEET (See also GASKET MATERIAL)

Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Belmont Packing & Rubber Co., Butler & Sepviva Sts., Phila. 7, Pa.
Buffalo Weaving & Belting Co., 262 Chandler St., Buffalo 7, N.Y
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Endura Mfg. Corp., 45 N. 4th St., Quakertown, Pa. (Vegetable Fibre)
Eureka Packing Co., 248-46th St., Brooklyn 20, N.Y.

(Continued)

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According to estimates by leading refrigeration publications, *more Suniso Oils are used by manufacturers of refrigeration and air-conditioning equipment than all other brands combined.* Correct grades of Suniso are available for every type of refrigeration and air-conditioning equipment. Suniso Oils are compatible with all types of refrigerants.

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For detailed specifications of the various Suniso grades, write for Form A-1622. Now also available on request, is the revised edition of the 52-page Sun Technical Bulletin entitled, "Lubrication of Refrigeration and Air-Conditioning Equipment." For copies of this bulletin or additional data on Suniso Refrigeration Oils, write Sun Oil Company, Department RD1, Philadelphia 3, Pa.

SUNOCO

SUN PETROLEUM PRODUCTS

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PACKING (Continued)

Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
Goetz Branch, Johns-Manville Products Corp., Allen Ave., New Brunswick, N.J.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Hollow Center Packing Co., 6523 Euclid Ave., Cleveland 3, O.
Janos Asbestos Co., 39 Cortlandt St., N.Y.C. 7
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Keasbey & Mattison Co., Ambler, Pa.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Linear, Inc., State Rd. & Levick St,. Phila. 35, Pa.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J
Rhopac, Inc., 168 N. Clinton St., Chicago 6, Ill.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.
John M. Watt's Sons, 112 Walnut St., Phila., Pa.

PAINT (See FINISHES or particular type following)

PAINT, ASPHALT

American Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
Arco Co., 7301 Bessemer Ave., Cleveland 4, O.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Grand Rapids Varnish Corp., 1350 Steele Ave., S.W., Grand Rapids 2, Mich.
Alfred Hague & Co., 227-34th St., Brooklyn 32, N.Y.
Hetzel Roofing Products Co., 67 Main St., Newark, N.J.
A. C. Horn Co., Inc., 43-36-10th St., Long Island City, N.Y.
Inertol Co., Inc., 470 Frelinghuysen Ave., Newark 5, N.J.
Interchemical Corp., 57 State St., Newark, N.J.
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
Standard Varnish Wks., 2600 Richmond Terrace, Staten Island 3, N.Y.
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
U. S. Gutta Percha Paint Co., Dudley & Eddy Sts., Providence 1, R.I.
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
Wilbur-Williams Co., 43 Leon St., Boston, Mass.
Zapon Div., Atlas Powder Co., Stamford, Ct.

PAINT, CORROSION RESISTING

Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
American Chemical Paint Co., Ambler, Pa. *(p. 41)*
American Pipe & Construction Co., Box 3428 Terminal Annex P.O., Los Angeles, Cal.
Arco Co., 7301 Bessemer Ave., Cleveland 4, O.
Brooklyn Varnish Mfg. Co., Inc., 50 Jay St., Brooklyn 1, N.Y.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Ernecke & Salmstein Co., 1611 N. Sheffield Ave., Chicago 14, Ill.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Grand Rapids Varnish Corp., 1350 Steele Ave., S.W., Grand Rapids 2, Mich.
Alfred Hague & Co., 227-34th St., Brooklyn 32, N.Y.
Haveg Corp., Marshallton, Del.
A. C. Horn Co., Inc., 43-36-10th St., Long Island City, N.Y.
Inertol Co., Inc., 470 Frelinghuysen Ave., Newark 5, N.J.
Interchemical Corp., 57 State St., Newark, N.J.
Jones-Dabney Co., Div. of Devoe & Raynolds, 1481 S. 11th St., Louisville 8, Ky.
Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
National Lead Co., 111 Broadway, N.Y.C. 6
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
Standard Varnish Wks., 2600 Richmond Terrace, Staten Island 3, N.Y.
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
United Chromium, Inc., 51 E. 42nd St., N.Y.C. 17
U. S. Gutta Percha Paint Co., Dudley & Eddy Sts., Providence 1, R.I.
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.
Wailes Dove-Hermiston Corp., 448 South Ave., Westfield, N.J.
Wilbur-Williams Co., 43 Leon St., Boston, Mass.
Zapon Div., Atlas Powder Co., Stamford, Ct.

PAINT, HEAT RESISTING

Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
American Chemical Paint Co., Ambler, Pa. *(p. 41)*
Arco Co., 7301 Bessemer Ave., Cleveland 4, O.
Brooklyn Varnish Mfg. Co., Inc., 50 Jay St., Brooklyn 1, N.Y.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Ill.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Grand Rapids Varnish Corp., 1350 Steele Ave., S.W., Grand Rapids 2, Mich.
A. C. Horn Co., Inc., 43-36-10th St., Long Island City, N.Y.
Inertol Co., Inc., 470 Frelinghuysen Ave., Newark 5, N.J.
Interchemical Corp., 57 State St., Newark, N.J.
Jones-Dabney Co., Div. of Devoe-Raynolds, 1481 S. 11th St., Louisville 8, Ky.
Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
Standard Varnish Wks., 2600 Richmond Terrace, Staten Island 3, N.Y.
Steelcoate Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
U. S. Gutta Percha Paint Co., Dudley & Eddy Sts., Providence 1, R.I.
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.
Wailes Dove-Hermiston Corp., 448 South Ave., Westfield, N.J.
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
Wilbur-Williams Co., 43 Leon St., Boston, Mass.
Zapon Div., Atlas Powder Co., Stamford, Ct.

PAINT, METALLIC

Aetna Plywood & Veneer Co., 1731 Elston Ave., Chicago 22, Ill.
Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
Arco Co., 7301 Bessemer Ave., Cleveland 4, O.
Brooklyn Varnish Mfg. Co., Inc., 50 Jay St., Brooklyn 1, N.Y.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Grand Rapids Varnish Corp., 1350 Steele Ave., S.W., Grand Rapids 2, Mich.
Alfred Hague & Co., 227-34th St., Brooklyn 32, N.Y.
Hetzel Roofing Products Co., 67 Main St., Newark, N.J.
A. C. Horn Co., Inc., 43-36-10th St., Long Island City, N.Y.

(Continued)

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York, Pennsylvania

Complete Range of Mechanical Cooling Equipment, Refrigeration and Air Conditioning to Meet Every Need

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Compression refrigerating machines and complete systems for use with ammonia and "Freon-12" ammonia absorption refrigerating systems especially adapted to operation with exhaust steam.

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Equipment and engineering for all marine refrigerating and air conditioning requirements. This includes general service refrigeration, cargo space refrigeration and air conditioning.

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Interchemical Corp., 57 State St., Newark, N.J.
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Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Quigley Co., Inc., 527-5th Ave., N.Y.C. 17
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
Smooth-On Mfg. Co., 572 Communipaw Ave., Jersey City 4, N.J.
Standard Varnish Wks., 2600 Richmond Terrace, Staten Island 3, N.Y.
Steelcote Mfg. Co., 3418 Gratiot Ave., St. Louis 3, Mo.
U. S. Gutta Percha Paint Co., Dudley & Eddy Sts., Providence 1, R.I.
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
Wilbur-Williams Co., 43 Leon St., Boston, Mass.
Zapon Div., Atlas Powder Co., Stamford, Ct.

PANELS, CABINET

Ace Cabinet Corp., New Bedford, Mass.
Falstrom Co., 13 Falstrom Court, Passaic, N. J.
Motors Metal Mfg. Co., 5936 Milford Ave., Detroit 10, Mich.
Permanente Products Co., 1924 Broadway, Oakland 12, Cal.

PANELS, INSTRUMENT (See INSTRUMENT PANELS)

PAPER, ASBESTOS

Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.

PAPER, BUILDING

Aetna Plywood & Veneer Co., 1731 Elston Ave., Chicago 21, Ill.
Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
Angier Corp., Framingham, Mass.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Philip Carey Mfg. Co., Lockland, Cin'ti. 15, O.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
Lehon Co., 4425 S. Oakley Ave., Chicago 9, Ill.
C. Nelson Mfg. Co., 4016 N. Union St., St. Louis 15, Mo.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Wilmington Fiber Specialty Co., P.O. Drawer 1028, Wilmington 99, Del.
Wolverine Fabricating & Mfg. Co., Inc., Princess St. & M.C.R.R., Inkster, Mich.

PAPER & FABRIC, COATED OR IMPREGNATED (See also FIBRE)

Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

PAPER PRODUCTS, MOLDED

Panelyte Div., St. Regis Paper Co., 230 Park Ave., N.Y.C. 17

PARTS, HARDENED & GROUND

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.

PASSING DOORS (See COLD STORAGE DOORS)

PERFORATED METALS (See METAL, PERFORATED)

PHOSPHOR BRONZE

American Brass Co., Waterbury 88, Ct.
Bristol Brass Corp., Bristol, Ct.
Chase Brass & Copper Co., 236 Grand St., Waterbury, Ct.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.

PHOTOGRAPHIC COOLING EQUIPMENT (See also WATER COOLERS)

Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*

PILLOW BLOCKS (See BEARINGS)

PILOT LIGHTS

Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.

PINS, DOWEL

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Bethlehem Steel Co., Bethlehem, Pa.
Townsend Co., New Brighton, Pa.

PINS, SPECIAL

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Standard Steel Specialty Co., Beaver Falls, Pa.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

PIPE (See particular type following also TUBES, also TUBING)

PIPE, COPPER

American Brass Co., Waterbury 88, Ct.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Lewin Metals Div., Lewin-Mathes Co., 1111 Chouetau, Louis, Mo.
Mueller Brass Co., Port Huron, Mich.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Standard Valve Mfg. Co., 817 Albany St., Boston 19, Mass.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

PIPE, CORROSION RESISTING

American Brass Co., Waterbury 88, Ct.
Babcock & Wilcox Tube Co., Beaver Falls, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Duriron Co., Inc., Dayton 1, O.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Haveg Corp., Marshallton, Del.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
National Tube Co., U. S. Steel Corp., Subsidiary, P.O. Box 266, Pittsburgh, Pa.

Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc, 1948 N. 9th St., St. Louis 6, Mo.
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Warren Foundry & Pipe Corp., 11 Broadway, N.Y.C. 4

PIPE, FURNACE & STOVE

L. J. Mueller Furnace Co., 2005 W. Oklahoma Ave., Milwaukee 7, Wis.
National Gypsum Co., 325 Delaware Ave., Buffalo 2, N.Y. *(p. 133)*
Nebel Mfg. Co., 2366 Woodhill Rd., Cleveland 20, O.
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
Wheeling Corrugating Co., Wheeling, W. Va.

PIPE, HARD RUBBER & PLASTIC

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Haveg Corp., Marshallton, Del.
Luzerne Rubber Co., Trenton 9, N.J.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

PIPE, REFRIGERATION

Bethlehem Steel Co., Bethelhem, Pa.
A. M. Byers Co., Clark Bldg., Pittsburgh 22, Pa.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Republic Steel Corp., Republic Bdlg., Cleveland 1, O.
John Simmons Co., Inc., 50 Church St., N.Y.C. 7

PIPE, SEAMLESS

American Brass Co., Waterbury 88, Ct.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Babcock & Wilcox Tube Co., Beaver Falls, Pa.
Bethlehem Steel Co., Bethlehem, Pa.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
National Tube Co., U. S. Steel Corp. Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge Sts., Etna P.O., Pittsburgh 23, Pa.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Joseph T. Ryerson & Son Inc., 16th & Rockwell Sts., Chicago, Ill.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

PIPE, STEEL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Rolling Mill Co., Middletown, O.
Bethlehem Steel Co., Bethlehem, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks. Inc. 128 Linden St., Allentown, Pa.
National Tube Co., U. S. Steel Corp., Subsidiary, P.O Box 266, Pittsburgh, Pa.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa. (Steel & Stainless)
Pittsburgh Tube Co., 323-4th Ave., Pittsburgh 22, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.

PIPE, WROUGHT IRON

A. M. Byers Co., Clark Bldg., Pittsburgh 22, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
John Simmons Co., Inc., 50 Church St, N.Y.C. 7

PIPE BENDS & BENDING (See also PIPING, PREFABRICATED; also BARE IRON PIPE COILS; also RETURN BENDS)

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Fitzsimons Mfg. Co., 3375 E. Outer Dr., Detroit 12, Mich.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. *(p. 47)*
Joseph E. Lewis & Co., Inc., 1315 Carroll St., Baltimore 30, Md.
Mueller Brass Co., Port Huron, Mich. (Copper)
A. B. Murray Co., Inc., 601 Green Lane, Elizabeth, N. J.
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Reynolds Mfg. Co., Inc., Springfield, Mo.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4, O.
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Standard Valve Mfg. Co., 817 Albany St., Boston 19, Mass.
Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Wallace Tube Co., Subsidiary of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
York Corp., York, Pa. *(p. 163)*

PIPE CLAMPS & HANGERS (See CLAMPS & HANGERS)

PIPE COVERING (See INSULATION)

PIPING, PREFABRICATED (See also PIPE BENDING)

California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R .J., Newark 5, N.J. *(p. 239)*
Fitzsimons fg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Ric-Wil Co., Union Commerce Bldg., Cleveland 14, O.
Shaw-Kendall Engrg. Co., 120 S. Superior St., Toledo 4, O.
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.

PISTONS

Frick Co., Waynesboro, Pa. *(p. 47)*
McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.
Vilter Mfg. Co., 224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
York Corp., York, Pa. *(p. 163)*

PISTON PINS

Frick Co., Waynesboro, Pa. *(p. 47)*
McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.
Saginaw Malleable Iron Div., Gen'l. Motors Corp., Saginaw, Mich.
York Corp., York, Pa. *(p. 163)*

PISTON RINGS

American Metallic Packing Co., 3621 Mexico St., Pittsburgh 12, Pa.
Burd Piston Ring Co., 10th & 23rd Ave., Rockford, Ill
C. Lee Cook Mfg. Co., 916 S. 8th St., Louisville 3, Ky.
McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.
Superior Piston Ring Co., 321 S. Cicero Ave., Chicago 44, Ill.
Wilkening Mfg. Co., 2000 S. 71st St., Phila. 42, Pa.

PLASTER, INSULATING (See INSULATING PLASTER & CONCRETE)

PLASTIC MOLDING COMPOUNDS

Celanese Plastics Corp., 180 Madison Ave., N.Y.C. 16
Chemaco Corp., Berkeley Heights, N.J.
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Haveg Corp., Marshallton, Del.
Plaskon Div., Libbey, Owens, Ford Glass Co., 2112 Sylvan Ave., Toledo 6, O.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17

PLASTIC SHEETS, RODS & TUBES

Aetna Plywood & Veneer Co., 1731 Elston Ave., Chicago 22, Ill.
Celanese Plastics Corp., 180 Madison Ave., N.Y.C. 16
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Haveg Corp., Marshallton, Del.
Hudson Products Co., Inc., 4400 St. Aubin, Detroit 7 Mich.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
Wilmington Fiber Specialty Co., P.O. Drawer 1028, Wilmington 99, Del.

PLASTICIZERS

Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Celanese Chemical Corp., 180 Madison Ave., N.Y.C. 16
Dow Chemical Co., Midland, Mich.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.

PLATE, ALLOY

Bethlehem Steel Co., Bethlehem, Pa.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky. (Aluminum)
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.

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Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Lukens Steel Co., Coatesville, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Youngstown Sheet & Tube Co., Youngstown, O.

PLATE COILS
(A—Ammonia; B—Other refrigerants)

(B) Ace Cabinet Corp., New Bedford, Mass.
(B) Allied Freezer Corp., 2822 Park Ave., N.Y.C. 51
(B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(B) Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 87)*
(A,B) Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill. *(p. 167)*
(A,B) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(B) Glenwood-Inglewood Co., Glenwood Ave. at Thomas, Minneapolis 5, Minn.
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
Stanley Knight Corp., 3430 N. Pulaski Rd., Chicago 41, Ill.
(A,B) Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 215)*
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
(B) Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
(B) Standard Refrigeration Co., 232 S. Hoyne Ave., Chicago 20, Ill.
(A,B) Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*
(B) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.
(B) Yoder Co., 1311 W. 80th St., Cleveland 2, O.

PLATE COILS WITH EUTECTIC
(A—Ammonia; B—Other refrigerants)

(B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(A,B) Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill. *(p. 167)*
(A,B) Frosty Foods Equip. Co., 305 Benson Blvd., Sioux City 15, Ia.
(A,B) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(A,B) Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 215)*
C. F. Moores Co., Inc., 1123 Ivy Hill Rd., Wyndmoor, Phila. 18, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*
(B) Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.

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L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich. *(p. 126)*

PLYWOOD

Aetna Plywood & Veneer Co., 1731 Elston Ave., Chicago 22, Ill.

PORCELAIN, ELECTRICAL

General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Red Spot Elec. Co., Tacoma, Wash.

PORCELAIN ENAMEL MATERIALS

Chicago Vitreous Enamel Product Co, 1407 S. 55th Court, Cicero 50, Ill.
Ferro Enamel Corp., Harvard & 56th St., Cleveland 5, O.
Pemco Co., 5601 Easter Ave., Batimore 23, Md.

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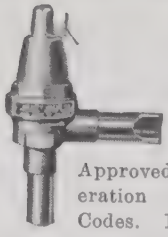

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Fox Co., Fox Lane, Cin'ti. 23, O

POROUS METAL (See POWDERED METAL PRODUCTS)

POULTRY REFRIGERATORS (See DISPLAY CASES)

POWDERED METAL PRODUCTS

Amplex Div., Chrysler Corp., 6501 Harper Ave., Detroit 31, Mich.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Keystone Carbon Co., Inc., 1935 State St., St. Marys, Pa.
P. R. Mallory & Co., Inc., 3029 W. Washington St., Indpls., Ind.
Moraine Products Div., Gen'l. Motors Corp., Dayton 1, O.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J

POWER ELEMENTS

Clifford Mfg. Co., 564 E. 1st St., Boston 27, Mass. *(p. 29)*
Fulton Sylphon Co., Knoxville, Tenn.
Red Spot. Elec. Co., Tacoma, Wash.

PRESSURE RELIEF DEVICES, REFRIGERANT (See also PRESSURE RELIEF VALVES)

Black, Sivalls & Bryson, Kansas City 3, Mo.

PRESSURE RELIEF VALVES, REFRIGERANT (See also PRESSURE RELIEF DEVICES)

(A—Ammonia; B—Other refrigerants)

(A,B) Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
Black, Sivalls & Bryson, Kansas City 3, Mo.
(A,B) A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
(A) Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
(A) Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Farris Engrg. Corp., 400 Commercial Ave., Palisades Park, N.J.
(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(A,B) Henry Valve Co., Melrose Park, Ill. *(p. 168)*
(A,B) Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
(B) Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
(B) Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
(A,B) King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 158)*
(B) Mueller Brass Co., Port Huron, Mich.
(A,B) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
(A,B) Refrigerating Specialities Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
(B) Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Watts Regulator Co., 10 Embankment St., Lawrence, Mass.
(B) Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
(B) Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
(A) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

PRESSURE RELIEF VALVES, WATER

Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
D'Este Div., American Chain & Cable Co., Inc., Reading Pa.
Farris Engrg. Corp., 400 Commercial Ave., Palisades Park, N.J.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa

Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Mueller Co,. 512 W. Cerro Gordo St., Decatur 70, Ill.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Refrigerating Specialities Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
Watts Regulator Co., 10 Embankment St., Lawrence, Mass.

PRESSURE VESSELS

Alloy Products Corp., 1045 Perkins Ave., Waukesha, Wis.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 7
Bethlehem Steel Co., Bethlehem, Pa.
Black, Sivalls & Bryson, Kansas City 3, Mo.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Chattanooga Boiler & Tank Co., P.O. Box 110, Chattanooga, Tenn.
Colonial Iron Wks. Co., 17643 St. Clair Ave., Cleveland 10, O
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Frick Co., Waynesboro, Pa. *(p. 47)*
Fruehauf Trailer Co., 10940 Harper Ave., Detroit 32, Mich. (Transport)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Gustav Glaser Co., Inc., 2 Wait St., Paterson 4, N.J.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
R. Munroe & Sons Mfg. Corp., 23rd & Smallman Sts., Pittsburgh 22, Pa.
National Tube Co., U. S. Steel Corp. Subsidiary, P.O. Box 266, Pittsburgh, Pa.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Pittsburgh-Des Moines Construction Co., Neville Island, Pittsburgh 25, Pa.
Pressed Steel Tank Co., 1471 S. 66th St., Milwaukee 14, Wis. *(p. 194)*
Quaker City Iron Wks., Aramingo Ave. & E. Tioga St., Phila., Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 150)*
Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.
Stover Steel Tank & Mfg. Co., 100 S. Hancock St., Freeport, Ill.
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Walsh Holyoke Boiler Wks., Div. of Continental-United Industries Co., Inc., 110 Appleton St., Holyoke, Mass.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

PRESSURESTATS (See CONTROLS)

PRIMERS (See FINISHES)

PROGRAM CONTROLLERS (See CONTROLS, PROGRAM)

PROOF BOXES, BAKERY

Baker-Perkins, Inc., Saginaw, Mich.
Drying System, Inc., 1810½ Foster Ave., Chicago 40, Ill.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Fred D. Pfening Co., 1075 W. 5th Ave., Columbus 8, O.
Union Steel Products Co., 448 Pine St., Albion, Mich.

PROPANE

Carbide & Carbon Chemicals Corp., Unit. of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Sun Oil Co., 1608 Walnut St., Phila. 3, Pa. *(p. 161)*

PROPORTIONING UNITS

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
D. W. Haering & Co., 205 W. Wacker Dr., Chicago 6, Ill.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.

PROPYLENE GLYCOL

Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Dow Chemical Co., Midland, Mich.

PSYCHROMETERS

Henry J. Green, 1191 Bedford Ave., Brooklyn 16, N.Y.
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

PULLEYS

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American Pulley Co., 4200 Wissahickon Ave., Phila. 29, Pa.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Browning Mfg. Co., Inc., Maysville, Ky.
Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Dayton Rubber Mfg. Co., 2342 W. Riverview Ave., Dayton 1, O.
R. & J. Dick Co., Inc., Passaic, N.J.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Linderme Machine & Tool Co., 12241 Coyle Ave., Detroit 27, Mich.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Nice Ball Bearing Co., 30th & Hunting Park Ave., Phila. 40, Pa.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Chas. A. Schieren Co., 30 Ferry St., N.Y.C. 7
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
John M. Watt's Sons, 112 Walnut St., Phila., Pa.

PULLEYS, ADJUSTABLE

Browning Mfg. Co., Inc., Maysville, Ky.
Chicago Die Casting Mfg. Co., 2500 Monroe St., Chicago 12, Ill.
Jeffrey Mfg. 887 N. 4th St., Columbus 16, O.
Lau Blower Co., 2007 Home Ave., Dayton 7, O.
Linderme Machine & Tool Co., 12241 Coyle Ave., Detroit 27, Mich.
Scientiae Corp., 101 Pine St., Dayton 2, O.

PULLEYS, CONVEYOR

R. & J. Dick Co., Inc., Passaic, N.J.
Dodge Mfg. Corp., 505 S. Union St., Mishawaka, Ind.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.

PUMPS, AMMONIA

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.

Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Cold Control, Inc., 111 Broadway, N.Y.C. 6
M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Foster Pump Wks., Inc., 50 Washington St., Brooklyn 1, N.Y.
Goulds Pumps, Inc., Seneca Falls, N.Y.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Lawrence Machine & Pump Corp., 371 Market St., Lawrence, Mass.
Charles S. Lewis & Co., 2207 Pine St., St. Louis 3, Mo.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Union Steam Pump Co., Battle Creek, Mich.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

PUMPS, BRINE

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
American Well Wks., Aurora, Ill.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Chicago Pump Co., 2300 Wolfram St., Chicago 18, Ill.
M. T. Davison Co., 43 Keap St., Brooklyn 11, N.Y.
Dean Hill Pump Co., 4000 E. 16th St., Indpls. 7, Ind.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Goulds Pumps, Inc., Seneca Falls, N.Y.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Lawrence Machine & Pump Corp., 371 Market St., Lawrence, Mass.
Charles S. Lewis & Co., 2207 Pine St., St. Louis 3, Mo.
Union Steam Pump Co., Battle Creek, Mich.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Weinman Pump Mfg. Co., 290 Spruce, Columbus 8, O.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
Yeomans Bros. Co., 1433 N. Dayton St., Chicago 22, Ill.
York Corp., York, Pa. *(p. 163)*

PUMPS, CENTRIFUGAL

Aldrich Pump Co., Allentown, Pa.
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
American Well Wks., Aurora, Ill.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Beach-Russ Co., 50 Church St., N.Y.C. 7
Brown & Sharpe Mfg. Co., 235 Promenade St., Providence 1, R.I.
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Columbus Steam Pump Wks. Co., 724 W. Gay St., Columbus 8, O.
A. D. Cook, Inc., Lawrenceburg, Ind.
M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
Dean Hill Pump Co., 4000 E. 16th St., Indpls. 7, Ind.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Deming Co., 884 S. Broadway, Salem, O.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Goulds Pumps, Inc., Seneca Falls, N.Y.
Gray-Mills Corp., 1948 Ridge Ave., Evanston, Ill.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Lawrence Machine & Pump Corp., 371 Market St., Lawrence, Mass.
Layne & Bowler, Inc., P.O. Box 215, Hollywood Sta., Memphis 8, Tenn.
Charles S. Lewis & Co., 2207 Pine St., St. Louis 3, Mo.
Peerless Pump Div., Food Machinery Corp., 301 West Ave., 26, Los Angeles 31, Cal.
Ruthman Machinery Co., 1810 Reading Rd., Cin'ti., O.
F. J. Stokes Machine Co., Tabor Rd., Phila. 20, Pa.
Swaby Mfg. Co., 2330 W. Cermak Rd., Chicago 8, Ill.
Trane Co., La Crosse, Wis. *(p. 14)*
Union Steam Pump Co., Battle Creek, Mich.
Weinman Pump Mfg. Co., 290 Spruce, Columbus 8, O.
Western Engine Co., Div. of Vernon Tool Co., Ltd., 1101 Meridian Ave., Alhambra, Cal.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, CHEMICAL

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Beach-Russ Co., 50 Church St., N.Y.C. 7
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Foster Pump Wks., Inc., 50 Washington St., Brooklyn 1, N.Y.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Haveg Corp., Marshallton, Del.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Jabsco Pump Co., 2031 N. Lincoln St., Burbank, Cal.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Lawrence Machine & Pump Corp., 371 Market St., Lawrence, Mass.
Manzel Bros. Co., Div. of Frontier Industries, Inc., 309 Babcock St., Buffalo 10, N.Y.
Nash Engrg. Co., 358 Wilson Rd., S. Norwalk, Ct.
National Lead Co., 111 Broadway, N.Y.C. 6
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Milton Roy Co., 1401 E. Mermaid Ave., Phila. 18, Pa.
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
Yeomans Bros. Co., 1433 N. Dayton St., Chicago 22, Ill

PUMPS, COOLANT

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Gray-Mills Corp., 1948 Ridge Ave., Evanston, Ill.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Jabsco Pump Co., 2031 N. Lincoln St., Burbank, Cal.
National Lead Co., 111 Broadway, N.Y.C. 6
Ruthman Machinery Co., 1810 Reading Rd., Cin'ti., O.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Whittington Pump & Engrg. Corp., 245 S. Meridian St., Indpls. 4, Ind.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, CORROSION RESISTING

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.

M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
Duriron Co., Inc., Dayton 1, O.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Haveg Corp., Marshallton, Del.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Jabsco Pump Co., 2031 N. Lincoln St., Burbank, Cal.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Lawrence Machine & Pump Corp., 371 Market St., Lawrence, Mass.
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Milton Roy Co., 1401 E. Mermaid Ave., Phila. 18, Pa. (Chemical)
Weinman Pump Mfg. Co., 290 Spruce, Columbus 8, O.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, DEEP WELL

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
American Well Wks., Aurora, Ill.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
A. D. Cook, Inc., Lawrenceburg, Ind.
M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
Deming Co., 884 S. Broadway, Salem, O.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Goulds Pumps, Inc., Seneca Falls, N.Y.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Layne & Bowler, Inc., P.O. Box 215, Hollywood Sta., Memphis 8, Tenn.
Peerless Pump Div., Food Machinery Corp., 301 West Ave., 26, Los Angeles 31, Cal.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, FREON & METHYL

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Goulds Pumps, Inc., Seneca Falls, N.Y.
Lawrence Machine & Pump Corp., 371 Market St., Lawrence, Mass.
Charles S. Lewis & Co., 2207 Pine St., St. Louis 3, Mo.
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 15, N.Y.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, GEAR

Deming Co., 884 S. Broadway, Salem, O.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Gray-Mills Corp., 1948 Ridge Ave., Evanston, Ill.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Kinney Mfg. Co., 3529 Washington St., Boston 30, Mass. *(p. 173)*
Peerless Pump Div., Food Machinery Corp., 301 West Ave., 26, Los Angeles 31, Cal.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Watson Flagg Machine Co., 845 E. 25th St., Paterson 3, N.J.
Western Engine Co., Div. of Vernon Tool Co., Ltd., 1101 Meridian Ave., Alhambra, Cal.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, HAND

Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
Deming Co., 884 S. Broadway, Salem, O.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, HIGH PRESSURE

Aldrich Pump Co., Allentown, Pa.
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Austin Mason Co., 15 Park Row, N.Y.C.
Cooper-Bessemer Corp., Mt. Vernon, O.
M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Eisler Engrg. Co., Inc., 740 S. 13th St., Newark 3, N.J.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Nathan Mfg. Co., 416 E. 106th St., N.Y.C. 29
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
Yeomans Bros. Co., 1433 N. Dayton St., Chicago 22, Ill.

PUMPS, JET

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Cold Control, Inc., 111 Broadway, N.Y.C. 6
Peerless Pump Div., Food Machinery Corp., 301 West Ave., 26, Los Angeles 31, Cal.
Tranter Mfg. Co., 105 Water St., Pittsburgh 22, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, MIDGET

Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, OIL

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p.99)*
Beach-Russ Co., 50 Church St., N.Y.C. 7
Bowser, Inc., 1302 E. Creighton Ave., Ft. Wayne 2, Ind.
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Foster Pump Wks., Inc., 50 Washington St., Brooklyn 1, N.Y.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Kinney Mfg. Co., 3529 Washington St., Boston 30, Mass. *(p. 173)*
Nathan Mfg. Co., 416 E. 106th St., N.Y.C. 29
Racine Tool & Machine Co., 1000 Carlisle Ave., Racine, Wis.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Union Steam Pump Co., Battle Creek, Mich.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, POSITIVE PRESSURE

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E. Battle Creek, Mich.
M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Gray-Mills Corp., 1948 Ridge Ave., Evanston, Ill.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Nash Engrg. Co., 358 Wilson Rd., S. Norwalk, Ct.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Milton Roy Co., 1401 E. Mermaid Ave., Phila. 18, Pa. (Chemical)
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, SHALLOW WELL

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Deming Co., 884 S. Broadway, Salem, O.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Jabsco Pump Co., 2031 N. Lincoln St., Burbank, Cal.
Peerless Pump Div., Food Machinery Corp., 301 West Ave., 26, Los Angeles 31, Cal.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, SPECIAL

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Lawrence Machine & Pump Corp., 371 Market St., Lawrence, Mass.
Milton Roy Co., 1401 E. Mermaid Ave., Phila. 18, Pa. (Chemical)
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, SUBMERSIBLE

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Lawrence Machine & Pump Corp., 371 Market St., Lawrence, Mass.
Star Elec. Motors Co., 200 Bloomfield Ave., Bloomfield, N.J.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, SUMP

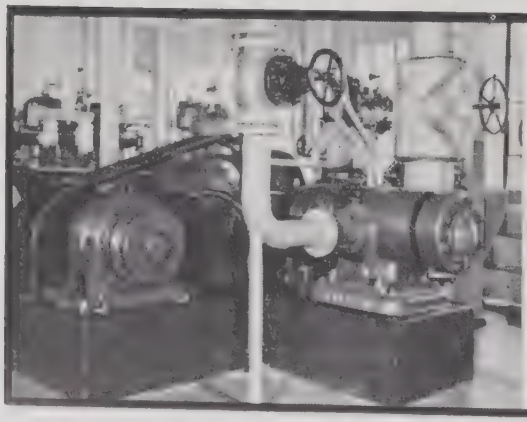
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Chicago Pump Co., 2300 Wolfram St., Chicago 18, Ill.
Columbus Steam Pump Wks. Co., 724 W. Gay St., Columbus 8, O.
A. D. Cook, Inc., Lawrenceburg, Ind.
M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Independent Pneumatic Tool Co., 600 W. Jackson Blvd., Chicago 6, Ill.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Jabsco Pump Co., 2031 N. Lincoln St., Burbank, Cal.
Byron Jackson Co., P.O. Box 2017 Terminal Annex, Los Angeles 54, Cal.
Lawrence Machine & Pump Corp., 271 Market St., Lawrence, Mass.
Peerless Pump Div., Food Machinery Corp., 301 West Ave., 26, Los Angeles 31, Cal.
Swaby Mfg. Co., 2330 W. Cermak Rd., Chicago 8, Ill.
Weinman Pump Mfg. Co., 290 Spruce, Columbus 8, O.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
Yeomans Bros. Co., 1433 N. Dayton St., Chicago 12, Ill.

PUMPS, VACUUM

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Austin Mason Co., 15 Park Row, N.Y.C.
Beach-Russ Co., 50 Church St., N.Y.C. 7
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
Columbus Steam Pump Wks. Co., 724 W. Gay St., Columbus 8, O.
M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Distillation Products, Inc., 755 Ridge Rd., W., Rochester 13, N.Y.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eisler Engrg. Co., Inc., 7430 S. 13th St., Newark 3, N.J.
Foster Pump Wks., Inc., 50 Washington St., Brooklyn 11, N.Y.
Fuller Co., Box 42, Catasauqua, Pa. *(p. 172)*
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Jabsco Pump Co., 2031 N. Lincoln St., Burbank, Cal.
Kinney Mfg. Co., 3529 Washington St., Boston 30, Mass. *(p. 173)*
Lummus Co., 420 Lexington Ave., N.Y.C.
Nash Engrg. Co., 358 Wilson Rd., S. Norwalk, Ct.
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Ernest Scott & Co., P.O. Box 82, Fall River, Mass.
Union Steam Pump Co., Battle Creek, Mich.
C. H. Wheeler Mfg. Co., 1741 Sedgley Ave., Phila. 32, Pa.
Whittington Pump & Engrg. Corp., 245 S. Meridian St., Indpls. 4, Ind.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

PUMPS, VANE

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
American Well Wks., Aurora, Ill.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Brown & Sharpe Mfg. Co., 235 Promenade St., Providence 1, R.I.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Foster Pumps., Inc., 50 Washington St., Brooklyn 1, N.Y.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Peerless Pump Div., Food Machinery Corp., 301 West Ave., 26, Los Angeles 31, Cal.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*



FULLER ROTARY COMPRESSORS AND VACUUM PUMPS

Built in single and two stage, for capacities on air to 3300 c. f. m., 125-lb. pressure. For vacuums to 29.90-in. (referred to 30-in. barometer.) Write for Bulletin C-5

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SAN FRANCISCO 7
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KINNEY HIGH VACUUM PUMPS

Mechanical, Oil Sealed

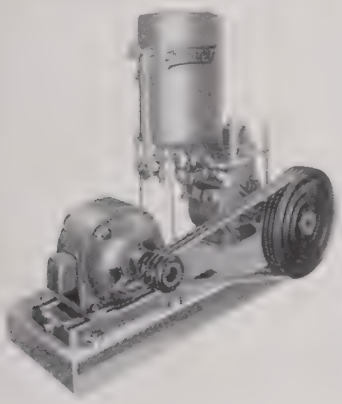
Kinney Dry Vacuum Pumps are standard equipment in the plants of the principal manufacturers of household refrigerators and quick freeze units for drying coils, compressor units and drying oil. They are also standard equipment in the plants producing temperature control units. Good mechanical performance coupled with high volumetric efficiency at the high vacuums needed to insure moisture-free and air-free units make Kinney Vacuum Pumps a necessity for such work. The same high performance is desirable for repair shops.

Kinney Vacuum Tight Valves insure tight systems.

We recommend Kinney heated separator tanks when excessive moisture is present in the system to be exhausted.

SINGLE STAGE VACUUM PUMPS

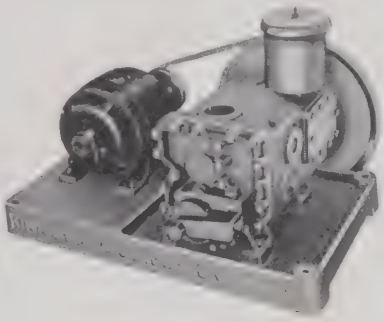
Models VSD and DVD



Eight pump sizes are available (13, 27, 46, 110, 217, 311, 486 and 702 cu. ft. per min.) with motors from ½ to 40 h.p. The three smallest sizes are furnished water cooled or air cooled; all larger sizes are furnished water cooled. Single Stage Pumps on a blank test will produce absolute pressure readings of 10 microns (.01 mm Hg.) or better.

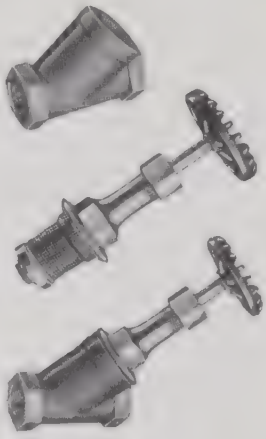
COMPOUND VACUUM PUMP

Model CVD



Two pump sizes are available (15 and 46 cu. ft. per min.) with 1 and 3 h.p. motors. The working mechanism is similar to the Single Stage but the two cylinders are connected in series. The Compound Pump has an air cooled casing of unique design and special provision for oil sealing and lubricating the high vacuum working parts. Low absolute pressures to 0.5 micron (.0005 mm Hg.) are regularly maintained.

VACUUM TIGHT VALVES



Kinney Vacuum Tight Valves greatly facilitate testing installations. They are constructed with a specially designed metal bellows, eliminating the usual stuffing box. Leakage has been reduced to .000,000,007 cu. ft. per second. Available in sizes 1", 1½", 2" and 3".

Ask for Kinney Vacuum Pump Bulletin V45.

PUMPS, WATER

American-Marsh Pumps, Inc., 1948 Capitol Ave., N.E., Battle Creek, Mich.
Aurora Pump Co., 178 Loucks St., Aurora, Ill.
Thomas Beckett & Co., Inc., 2118 Griffin St., Dallas 2, Tex.
Buffalo Pumps, Inc., 465 Broadway, Buffalo 4, N.Y.
Chicago Pump Co., 2300 Wolfram St., Chicago 18, Ill.
M. T. Davidson Co., 43 Keap St., Brooklyn 11, N.Y.
Walter H. Eagan Co., Inc., 2336 Fairmount Ave., Phila. 30, Pa.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Fairbanks, Morse & Co., 600 S. Michigan Ave., Chicago 5, Ill.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Foster Wheeler Corp., 165 Broadway, N.Y.C.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Jabsco Pump Co., 2031 N. Lincoln St., Burbank, Cal.
Byron Jackson Co., P.O. Box 2017, Terminal Annex, Los Angeles 54, Cal.
Lawrence Machine & Pump Corp., 371 Market St., Lawrence, Mass.
Layne & Bowler, Inc., P.O. Box 215, Hollywood Sta., Memphis 8, Tenn.
Charles S. Lewis & Co., 2207 Pine St., St. Louis 3, Mo.
National Engrg. & Mfg. Co., 213 W. 19th St., Kansas City 8, Mo. (Evaporator Cooler Pumps)
Peerless Pump Div., Food Machinery Corp., 301 West Ave., 26, Los Angeles 31, Cal.
Union Steam Pump Co., Battle Creek, Mich.
Viking Pump Co., 4th & Squire Sts., Cedar Falls, Ia.
Weinman Pump Mfg. Co., 290 Spruce, Columbus 8, O.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
Yeomans Bros. Co., 1433 N. Dayton St., Chicago 22, Ill.

PURGERS, REFRIGERANT

Armstrong Machine Wks., 846 Maple St., Three Rivers, Mich.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Guild & Garrison, Inc., 43 Keap St., Brooklyn 11, N.Y.
Mueller Brass Co., Port Huron, Mich.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Rex Engrg. & Sales Co., Box 4151, Oklahoma City, Okla.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

PURIFIERS, WATER (See WATER TREATING)

QUICK FREEZE CABINETS (See FARM & HOME FREEZERS)

QUICK FREEZERS (See also BLAST FREEZERS; also SHARP FREEZERS)

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Burge Ice Machine Co., 218 N. Jefferson St., Chicago 6, Ill.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
W. J. Finnegan Co., 7402 Santa Monica Blvd., Los Angeles 46, Cal. *(p. 116)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
York Corp., York, Pa. *(p. 163)*

RACK, GEAR & PINION

Beverage Engrg. & Equip. Co., 13301 Lakewood Hts. Blvd., Cleveland 7, O.
Link-Belt Co., 300 Pershing Rd., Chicago 9, Ill.
Standard Steel Specialty Co., Beaver Falls, Pa.

RACKS, WIRE (See also WIRE FORMING; also SHELVES)

Bethlehem Steel Co., Bethlehem, Pa.
Kennedy Metal Products Co., Preston St., & Audubon Park, Louisville 4, Ky.
Locker Engrg. Co., 521 N. La Cienga Blvd., Los Angeles 36, Cal.
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 15, Ill.

RADIATION, ULTRA-VIOLET (See LAMPS, BACTERICIDAL)

RAILROAD REFRIGERATION SYSTEMS

Cold Control, Inc., 111 Broadway, N.Y.C. 6
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Propane Development Corp., 41 Murray St., N.Y.C. 7

REACH-IN REFRIGERATORS (See REFRIGERATORS, REACH-IN)

RECEIVERS

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 124)*
Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Cold Control, Inc., 111 Broadway, N.Y.C. 6
Downington Iron Wks., Downingtown, Pa. *(p. 59)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Oakes North Chicago Div., Houdaille-Hershey Corp., 1900 Foss Park Ave., N. Chicago, Ill. *(p. 174)*
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Reynolds Mfg. Co., Inc., Springfield, Mo.
Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 19, Va.
Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

RECEPTACLES, ELECTRICAL

General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.
Pyle Nat'l. Co., 1371 W. 37th St., Chicago 9, Ill. *(p. 17)*

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Hydrogen brazed, internally clean receiver tanks in sizes to meet your particular requirements. Standard diameters range from 2½" to 6" O.D. Spacing of tapped spuds and overall lengths can be varied within broad ranges.

All receivers approved by Underwriters' Laboratories.

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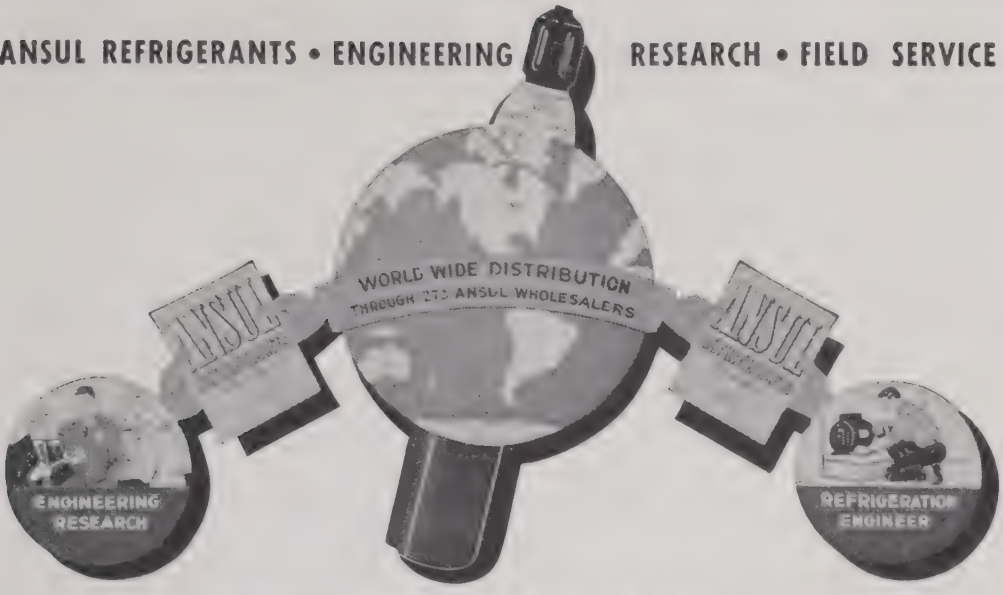

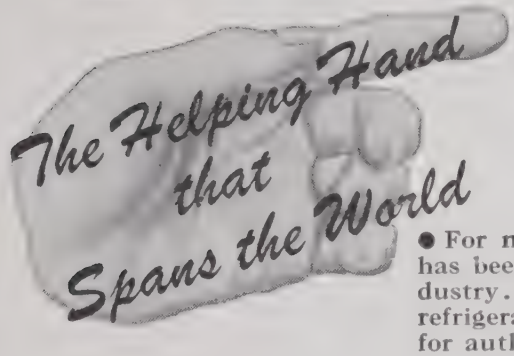

ANSUL TECHNICAL SERVICE

● For more than 30 years the name ANSUL has been well known in the refrigeration industry...not only as a producer of high grade refrigerants . . . but, also, as a reliable source for authentic up-to-the-minute technical information on refrigeration processes and problems-in-practice.

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ANSUL WHOLESALERS are ready and equipped to render an intelligent, co-operative service to refrigeration and service engineers on problems which arise from time-to-time in the operation of refrigerating systems.

LIQUID Methyl Chloride ANSUL

ANSUL REFRIGERANTS ARE AVAILABLE AT LEADING WHOLESALERS EVERYWHERE

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RECIRCULATING INJECTORS (See INJECTORS)

RECORDERS (See also particular type)

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Fischer & Porter Co., Hatboro, Pa. (Flow Rate)
Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Hays Corp., Michigan City, Ind.
Leeds & Northrup Co., 4970 Stenton Ave., Phila. 44, Pa.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12
Westinghouse Elec. Corp., Plane & Orange Sts., Newark 1, N.J.

RECORDERS, HUMIDITY

Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh 8, Pa.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Henry J. Green, 1191 Bedford Ave., Brooklyn 16, N.Y.
Leeds & Northrup Co., 4970 Stenton Ave., Phila. 44, Pa.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5 N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

RECORDERS, PRESSURE

Ashcroft Gauge Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Builders-Providence, Inc., Div., Builders Iron Foundry, 9 Codding St., Providence 1, R.I.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Fischer & Porter Co., Hatboro, Pa. (Flow Rate)
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Hays Corp., Michigan City, Ind.
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
F. J. Stokes Machine Co., Tabor Rd., Phila. 20, Pa.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16 Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Vacuum Equip. Div., Distillation Products, Inc., 755 Ridge Rd., W. Rochester 13, N.Y.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

RECORDERS, TEMPERATURE (See THERMOMETERS, RECORDING)

RECTIFIERS

Louis Allis Co., 427 E. Stewart St., Milwaukee 7, Wis.
Fansteel Metallurgical Corp., N. Chicago, Ill.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

REDUCERS, SPEED (See SPEED REDUCERS)

REFRIGERANTS (See also particular refrigerant)

Ansul Chemical Co., Marinette, Wis. *(p. 175)*
Henry Bower Chemical Mfg. Co., Gray's Ferry Rd. & 29th St., Phila. 46, Pa.
Carbide & Carbon Chemicals Corp., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Electrochemicals Dept., E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del. *(p. 153)*
Eston Chemicals, Inc., 3100 E. 26th St., Los Angeles 23, Cal. *(p. 152)*
Kinetic Chemicals, Inc., 10th & Market Sts., Wilmington, Del. *(p. 117)*
Mathieson Alkali Wks., Inc., 60 E. 42nd St., N.Y.C. 17
Virginia Smelting Co., W. Norfolk, Va. *(p. 177)*

REFRIGERATORS (See particular type; also DISPLAY CASES; also WALK-IN COOLERS, etc.)

REFRIGERATORS, BIOLOGICAL

Bowser, Inc., Terryville, Ct.
Gennett & Sons, Inc., 1 Main St., Richmond, Ind.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
E. J. McAleer & Co., Inc., 1422 N. 8th St., Phila. 22, Pa.
McCall Refrigerator Corp., Hudson, N.Y.
Refrigeration Corp. of America, Div. of Noma Elec Corp., 55 W. 13th St., N.Y.C. 11

REFRIGERATORS, FLORISTS

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(C) Cleveland Refrigerator Co., 2901 E. 65th St., Cleveland 4, O.
(C) Corbin Cabinet Lock Co., Div. of American Hardware Corp., New Britain, Ct.
(B) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
(A,B) Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
(A,B) Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 6, Tex.
(A,B,C) Gem Refrigerator Co., 2539 Germantown Ave., Phila. 33, Pa.
(A,B,C) General Refrigerator & Store Fixtures Co., 856 N. Broad St., Phila., Pa.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
(B) John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B,C) Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 179)*
(B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
(A,B,C) Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
(B) Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
(A,B,C) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
McCall Refrigerator Corp., Hudson, N.Y.
(B) McCray Refrigerator Co., Kendallville, Ind.
(A,B) Masterfreeze Corp., Sister Bay, Wis.
(A,B,C) Minneapolis Show Case & Fixture Co., 1009 Washington Ave., S., Minneapolis, Minn.
(A,B,C) Modern Appliance Co., 111 Ellsworth, San Mateo, Cal.
(A,C) Modern Refrigerator Wks., 823 Milford St., Glendale 3, Cal.
(B) Morton Show Cases, Inc., Washington Courthouse, O.

(Continued)

Virginia Smelting Co.

West Norfolk, Virginia

or F. A. Eustis, 131 State Street, Boston 4, Massachusetts—76 Beaver Street, New York 5, New York

Factory stocks: West Norfolk, Virginia; Jersey City, New Jersey; Boston, Massachusetts and other principal cities.

VIRGINIA *Refrigerants*

"Virginia" Refrigerants are produced under the direction of skilled technicians who recognize the importance of high purity standards for refrigeration use.

"EXTRA DRY ESOTOO" ("Virginia" Sulfur Dioxide, SO_2)

A water-white, oil-free product guaranteed to contain less than 50 P.P.M. moisture. Shipped in cylinders containing 10, 35 and 150 lbs. net weight; drums containing 1965 lbs. net C.L. or L.C.L.

"V-METH-L" ("Virginia" Methyl Chloride, CH_3Cl)

Low moisture and low acidity specifications, freedom from dirt and oil characterize this product—an ideal refrigerant for commercial applications.

Shipped in cylinders containing 18, 24, 65, 100, 145, and 300 lbs. net weight and in drums containing 1,340 lbs. net, C.L. or L.C.L.

METHYLENE CHLORIDE (CH_2Cl_2)

Highly refined to conform to strictest refrigeration purity standards—for household refrigerators and centrifugal air conditioning compressors.

Shipped in steel drums containing 550 lbs. net weight.

Write for free Booklet #36 "Tabulated Properties of Various Refrigerants"

AVAILABILITY

Prompt shipment from "Virginia" manufacturing plant; from emergency stocks located in key distribution centers and from refrigeration supply jobbers in all principal cities from coast to coast.

DISTRIBUTORS FOR KINETICS' "FREON-12," "FREON-22," "FREON-11," "FREON-21" AND "FREON-113"

REFRIGERATORS, FLORISTS (Continued)

(A,B,C) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B,C) Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
(A) Perfecold, Inc., 1940 S. Main St., Los Angeles 7, Cal.
(A,B) Puffer-Hubbard Mfg. Co., Grand Haven, Mich.
(B) Royal Store Fixture Co., 847 N. Broad St., Phila. 23, Pa.
(B) St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
(B) Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.
(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
(B) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 178)*
Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.
(B) Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.

REFRIGERATORS, FOOD SERVICE (See REFRIGERATORS, RESTAURANT)

REFRIGERATORS, HOUSEHOLD, ELECTRIC

Acme-Nat'l. Refrigeration Co., Inc., 634 Dean St., Brooklyn 17, N.Y.
Admiral Corp., 3800 Cortland St., Chicago 47, Ill.
Coolerator Co., 50 Ave. W. and Wadena St., Duluth 1, Minn.
Copeland Refrigeration Corp., Sidney, O. *(p. 64)*
Crosley Div., Aviation Corp., Cin'ti. 25, O.

RANNEY REFRIGERATOR COMPANY

"Three Generations of Better Refrigeration"

Manufacturers of

"Private Brand" Household Electric Refrigerators

Styling

Designing

Engineering

Testing

Production

Greenville, Michigan
Since 1892

Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Frostair Div., General Tire & Rubber Co., 332 S. Michigan Ave., Chicago 4, Ill.
General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
Gibson Refrigerator Co., 515 W. Williams St., Greenville, Mich.
Glascock Bros. Mfg. Co., Muncie, Ind.
Hotpoint, Inc., 5600 W. Taylor St., Chicago 44, Ill.
International Harvester Co., 180 N. Michigan Ave., Chicago 1, Ill.
Leonard Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
Monitor Equip. Corp., Riverdale, N.Y.C. 63
Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
National Cooperatives, Inc., 343 S. Dearborn St., Chicago 4, Ill.
Norge Div., Borg-Warner Corp., 670 E. Woodbridge St., Detroit 26, Mich.
Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.
Philco Corp., Tioga & C Sts., Phila. 34, Pa.
Rex Mfg. Co., Inc., Western Ave., Connersville, Ind.
Sanitary Refrigerator Co., Fond du Lac, Wis. *(p. 127)*
Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.
Westinghouse Elec. Corp., Mansfield, O.

REFRIGERATORS, HOUSEHOLD, GAS

Clayton & Lambert, 1701 Dixie Highwy., Louisville, Ky
Servel, Inc., Evansville 20, Ind. *(p. 69)*

REFRIGERATORS, HOUSEHOLD, ICE

American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
Coolerator Co., 50 Ave. W. and Wadena St., Duluth 1, Minn.

TYLER

FOR FOOD REFRIGERATION

- MEAT DISPLAY CASES
  CONVENTIONAL—OPEN SELF-SERVICE
- DAIRY CASES
  CONVENTIONAL—OPEN SELF-SERVICE
- FROZEN FOODS DISPLAY CASES
- WALL DAIRY REFRIGERATORS
- VEGETABLE CASES
- WALK-IN COOLERS
- REACH-IN BOXES
- DRY BEVERAGE COOLERS
- HARDER-FREEZ HOME FREEZERS
- SHELVING COUNTERS, ETC.

TYLER FIXTURE CORP.
NILES, MICHIGAN

Ice Cooling Appliance Corp., 33 S. Clark St., Chicago 3, Ill.
Modern Refrigerator Wks., 823 Milford St., Glendale 3, Cal.
Sanitary Refrigerator Co., Fond du Lac, Wis. *(p. 127)*

REFRIGERATORS, HOUSEHOLD, PRIVATE LABEL

Crosley Div., Aviation Corp., Cin'ti. 25, O.
Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.
Ranney Refrigerator Co., Greenville, Mich. *(p. 178)*

REFRIGERATORS, LABORATORY & INDUSTRIAL

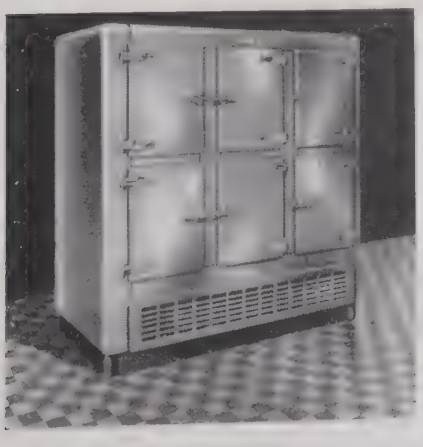
Ace Cabinet Corp., New Bedford, Mass.
American Instrument Co., 8010 Georgia Ave., Silver Spring, Md.
Bowser, Inc., Terryville, Ct.
Brunswicke-Balke-Collender Co., 623 S. Wabash Ave., Chicago 5, Ill.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Cleveland Refrigerator Co., 2901 E. 65th St., Cleveland 4, O.
Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
Frick Co., Waynesboro, Pa. *(p. 47)*
Frostrode Products, Inc., 19929 Exeter Rd., Detroit 3, Mich.
Gennett & Sons, Ind., 1 Main St., Richmond, Ind.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 179)*
Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
McCray Refrigerator Co., Kendallville, Ind.
J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
Precision Scientific Co., 3737 W Cortland St., Chicago 47, Ill.
Charles Q. Sherman Corp , 149 Broadway, N.Y.C. 6
Emil Steinhorst & Sons, Inc., 612 South St., Utica 3, N.Y.
York Corp., York, Pa. *(p. 163)*

REFRIGERATORS, MORTUARY

Audiffren Refrigerating Co., Proctor, Vt.
Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 179)*
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 129)*
C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.

REFRIGERATORS, REACH-IN (See also particular type, i.e., REFRIGERATORS, RESTAURANT; also REFRIGERATORS, FLORISTS)
(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit; D—Ice refrigerated)

(A) Airtemp Div., Chrysler Corp., 1119 Leo St., Dayton 1, O. *(p. 9)*
(A,B,C,D) American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(B) Cleveland Refrigerator Co., 2901 E. 65th St., Cleveland 4, O.
(A,B,D) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
(A,B) Duparquet Refrigerators, 33 E. 17th St., N.Y.C. 3
Electro-Kold, Div. of E. S. Matthews, Inc., S. 151 Post St., Spokane 8, Wash.
(A,B) Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
(A,B,C,D) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
(A,B,D) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 6, Tex.
(A) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(A,B,C,D) Gem Refrigerator Co., 2539 Germantown Ave. Phila. 33, Pa.
(A) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
(A,B,C,D) General Refrigerator & Store Fixtures Co., 856 N. Broad St., Phila., Pa.
(A,B,C,D) Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
(B) John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B,C,D) Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 179)*
(A,B) C. V. Hill & Col, Inc., 360 Pennington Ave., Trenton 1, N.J.
(A) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
(A,B,C,D) Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
(A,B,C) La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
(A,B,C) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
McCall Refrigerator Corp., Hudson, N.Y.
(A,B) McCray Refrigerator Co., Kendallville, Ind.
(A,B) Masterfreeze Corp., Sister Bay, Wis.
Matthews Refrigerator & Door Co., 5103 S.E. Powell Blvd., Portland 6, Ore.
(A,B,C) Minneapolis Show Case & Fixture Co., 1009 Washington Ave., S., Minneapolis, Minn.



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Standard and Custom Built Reach-in Refrigerators and Walk-in Coolers.

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1018 Commercial St., Waterloo, Iowa

(A,B,C,D) Modern Refrigerator Wks., 823 Milford St., Glendale 3, Cal.
(B) Morton Show Cases, Inc., Washington Courthouse, O.
(A,B) Nanticoke Refrigerator Manufacturers, Corner Hill & Slope Sts., Nanticoke, Pa.
(A) Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
(A,B,C,D) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B,C) Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
(A) Perfecold, Inc., 1940 S. Main St., Los Angeles 7, Cal.
(A,B,C) J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
(A,B) Puffer-Hubbard Mfg. Co., Grand Haven, Mich.
(A) Refrigeration Corp. of America, Div. of Noma Elec. Corp., 55 W. 13th St., N.Y.C. 11
(B) Roessing Mfg. Co., Sharpsburg Sta., Pittsburgh, Pa.
(A) W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
Royal Store Fixture Co., 847 N. Broad St., Phila. 23, Pa.
(B,C) St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.
(A,B) C. Schmidt Co., John & Livingston Sts., Cin'ti. 14, O.
(B) Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.
(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
(A,B) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall Mich.
(A) Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
(B) Star Metal Mfg. Co., Trenton Ave. & Ann St., Phila. 34, Pa.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 178)*
(A,B) United Refrigerator Mfg. Co., Inc., 350 Robert St., St. Paul 1, Minn.
(A,B) Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.
Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.
(A,B) Warren Co., Inc., P.O. Box 1435, Atlanta 1, Ga.
(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.
(A,B,C) Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.

REFRIGERATORS, RESTAURANT (See also REACH-IN REFRIGERATORS)

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

(A,B,C) American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
(A,B) Bemco Mfg. Corp., 1504 Minor at Pike, Seattle 1, Wash.
(A) Brunswick-Balke-Collender Co., 623 S. Wabash Ave., Chicago 5, Ill.
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(A) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
(A,B) Duparquet Refrigerators, 33 E. 17th St., N.Y.C. 3
(A,B,C) Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
(A,B) Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 6, Tex.
(A) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(A,B,C) Gem Refrigerator Co., 2539 Germantown Ave., Phila. 33, Pa.
(A) General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
(A,B,C) General Refrigerator & Store Fixtures Co., 856 N. Broad St., Phila., Pa.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
(B) John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B,C) Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 179)*
(A,B) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(A) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
(A,B,C) Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
(A,B) Koch Butchers' Supply Co., 600 E. 14th Ave., N Kansas City 16, Mo.
(A,B,C) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
McCall Refrigerator Corp., Hudson, N.Y.
(A,B) Masterfreeze Corp., Sister Bay, Wis.
(A,B,C) Modern Appliance Co., 111 S. Ellsworth, San Mateo, Cal.
(B) Morton Show Cases, Inc., Washington Courthouse, O.
(A,B) Nanticoke Refrigerator Manufacturers, Corner Hill & Slope Sts., Nanticoke, Pa.
(A,B,C) Natinoal Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A) Perfecold, Inc., 1940 S. Main St., Los Angeles 7, Cal.
(C) J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
(A) W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
(B) Royal Store Fixture Co., 847 N. Broad St., Phila. 23, Pa.
(B,C) St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.
(A,B) C. Schmidt Co., John & Livingston Sts., Cint'i. 14, O.
(A,B) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
(A) Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
(B) Star Metal Mfg. Co., Trenton Ave. & Ann St., Phila. 34, Pa.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 178)*
(A,B) United Refrigerator Mfg. Co., Inc., 350 Robert St., St. Paul 1, Minn.
(A,B) Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.
(A,B) Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.
(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.
(B,C) Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.

REFRIGERATORS & SALAD COUNTERS

Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Stanley Knight Corp., 3430 N. Pulaski Rd., Chicago 41, Ill.
Star Metal Mfg. Co., Trenton Ave. & Ann St., Phila. 34, Pa.

REFRIGERATORS, SPECIAL

Ace Cabinet Corp., New Bedford, Mass.
Bowser, Inc., Terryville, Ct.
Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Gennett & Sons, Inc., 1 Main St., Richmond, Ind.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 179)*
Hussman Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 5, Mo. *(p. 87)*
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Piper Co., Govans Post Office, Baltimore 12, Md.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 129)*

REGISTERS (See also GRILLES & AIR DIFFUSERS)

A-J Mfg. Co., 2119 Washington, Kansas City 8, Mo.
Auer Register Co., 3608 Payne Ave., Cleveland, O.
Barber-Colman Co., Rockford, Ill.
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
Hastings Air Conditioning Co., Inc., Hastings, Neb.
Lockjoint Wood Products Co., 1721 Mildred Ave., Wichita 7, Kan.
Milcor Steel Co., Milwaukee, Wis.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Pyle-Nat'l. Co., 1371 W. 37th St., Chicago 9, Ill. *(p. 17)*
Titus Mfg. Corp., Waterloo, Ia.
Tuttle & Bailey, Inc., New Britain, Ct
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.

REGULATORS (See particular type following; also CONTROLS; also CONTROLLERS)

REGULATORS, AIR PRESSURE

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
D'Este Div., American Chain & Cable Co., Inc., Reading, Pa.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hays Corp., Michigan City, Ind.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Mueller Co., 512 W. Cerro Gordo St., Decatur 70, Ill.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Pittsburgh Equitable Meter Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
C. J. Tagliabue Co., 550 Park Ave., Brooklyn 5, N.Y
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Watts Regulator Co., 10 Embankment St., Lawrence, Mass.

REGULATORS, CARBON DIOXIDE

Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hays Corp., Michigan City, Ind.
Piper Co., Govans Post Office, Baltimore 12, Md.
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.

REGULATORS, DIFFERENTIAL PRESSURE

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hays Corp., Michigan City, Ind.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Moore Products Co., H & Lycoming Sts., Phila. 24, Pa.
Pittsburgh Equitable Meter Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*

REGULATORS, FLOW

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Fischer & Porter Co., Hatboro, Pa.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hays Corp., Michigan City, Ind.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
Moore Products Co., H & Lycoming Sts., Phila. 24, Pa.
Pittsburgh Equitable Meter Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*

REGULATORS, LIQUID LEVEL (See also FLOATS etc.)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
Hays Corp., Michigan City, Ind.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Moore Products Co., H & Lycoming Sts., Phila. 24, Pa.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*

REGULATORS, PRESSURE

Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 205)*
D'Este Div., American Chain & Cable Co., Inc., Reading, Pa.
Gas & Oil Industry Labs., Inc., 4 Paine Ave., Irvington 11, N.J. (Gas Pressure)
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I
Hays Corp., Michigan City, Ind.
Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 7, Minn. *(p. 76)*
Moore Products Co., H. & Lycoming Sts., Phila. 24, Pa.
Mueller Co., 512 W. Cerro Gordo St., Decatur 70, Ill.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.

(Continued)

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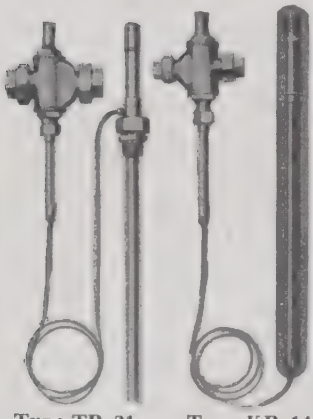

Type TR-21 Type KR-14

For ducts, reheaters, drinking water control, unit coolers and brine coils. They are simple, self-operated valves—using the irresistible force of liquid expansion. No stuffing boxes to leak, no auxiliary "power" required; all moving parts are *inside* the equipment. A type and size for every purpose—for steam, gas, oil, water or brine, for temperatures ranging from 0 to 300° F. Catalog No. RD600.

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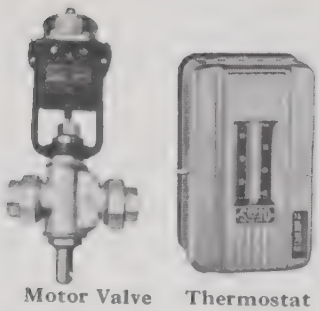
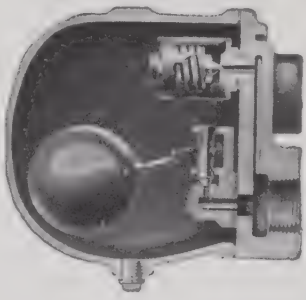


Motor Valve Thermostat

For refrigeration and air conditioning. When temperature control must be close, when wide and rapid fluctuations must be corrected instantly, when the thermostat is at a distance from the controlled unit, use electric regulators. They consist of room thermostats, aquastats, and limit controls for all heating and air conditioning needs; also motor valves for steam, water, brine or Freon. Catalog No. RD1000.

FLOAT-THERMOSTATIC TRAPS

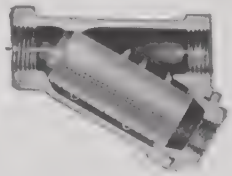


For blast coils, dripping ends of mains and risers, and for stack or blast heaters, large unit heaters and hot water generators. Automatic thermostatic air vents built in. Available in six sizes with connections ¾ to 2 in. Catalog No. RD450.

BLENDERS

For mixing warm and cold brine for cooling coils and unit coolers. Also mix very hot and cold water and deliver blended water at any temperature for which they have been adjusted. Valves are fully balanced so that control is not disturbed by differing or fluctuating water or brine pressures. Available in sizes ¾ in. to 4 in., for pressures up to 150 lb. Catalog Nos. RD700 and RD800.

SELF-CLEANING STRAINERS



For use in pipe lines carrying brine, steam, oil, gas, water, ammonia or air. Have large free screening area with minimum resistance to flow. Steam or air strainers can be cleaned by blowing through without disassembling. Made in cast iron, bronze or cast steel for pressures up to 500 lbs., with brass, iron or monel screens. Available in sizes ¼ to 8 in. Catalog No. RD1200.

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Manufacturing Distributors in Great Britain, Short & Mason, Ltd., London

Taylor Instruments for Indicating, Recording and Controlling Temperature, Pressure, Humidity, Flow and Liquid Level

The Taylor "Fulscope" Recording Controller

An air-operated instrument for accurate control of temperature and/or pressure on tanks, brine lines, compressors, or other refrigeration applications.

The "Fulscope" Controller gives practically any character of process control regardless of *time lag* in apparatus.

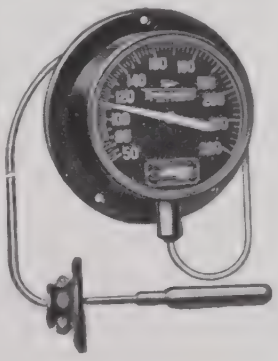
For the broad requirements of the Refrigeration Industry, Taylor offers Five Forms of "Fulscope" Control: *Fixed High Sensitivity*, for on-off control applications; *Adjustable Sensitivity*, for throttling control; *Automatic Reset*, for precision control; *Pre-Act*, a control effect based on *rate* of control point deviation; *Automatic Reset* and *Pre-Act*, for complex process problems.

The Taylor interchangeable unit construction permits one type of controller to be converted to another in the field.

For applications where a record is not needed, Taylor Indicating "Fulscope" Controllers are also available.

Taylor Dial Type Instruments for Temperature, Pressure, Rate of Flow, Liquid Level.

These instruments differ from the recording types only in their case construction and indicating mechanism. A heat-treated, chromium plated, alloy steel movement assure dependable long life. The large white numerals, and graduations on the 8″ black metal dial are easily read from a distance.



Taylor Recorders for Temperature and Pressure Measurement

These instruments are designed to give complete information regarding the processing requirements and operating conditions in refrigeration by supplying an accurate convenient record of temperature and pressure.

The simple design of these instruments insures long life and the universal case is drilled and tapped to receive any of the five standard Taylor "Fulscope" control mechanisms, if ever required.

Taylor Industrial Thermometers With Easy Reading BINOC Tubing

For cold water, brine, condenser, steam, and ammonia lines. The mercury-in-glass thermometers are essentially precise laboratory standards specially adapted to industrial requirements. Binoc tubing, because of its optically correct lens and background design, is three times easier to read; in fact, invites frequent reading.

Available with bulb forms of materials for all applications. Approximate temperature limits, minus 40 to plus 950 F.

Taylor BINOC Pocket Test Thermometer

Ideal for frequent testing of important temperatures. Taylor patented BINOC *Tubing* eliminates juggling and guesswork. High accuracy . . . Easier to Read.

REGULATORS, PRESSURE (Continued)

Pittsburgh Equitable Meter Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Schade Valve Mfg. Co., 2527 N. Bodine St., Phila. 33, Pa.
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos. 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
Watts Regulator Co., 10 Embankment St., Lawrence, Mass.
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

REGULATORS, TEMPERATURE

American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
D'Este Div., American Chain & Cable Co., Inc., Reading, Pa.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Eastern Industries, Inc., 296 Elm St., New Haven 6, Ct.
Fulton Sylphon Co., Knoxville, Tenn.
Hays Corp., Michigan City, Ind.
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Moore Products Co., H. & Lycoming Sts., Phila. 24, Pa.
Pyrometer Instrument Co., 103 Lafayette St., N.Y.C. 13
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 182)*
Sterling, Inc., 3738 N. Holton St., Milwaukee 12, Wis.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.

REGULATORS, VOLTAGE

American Transformer Co., 172 Emmet St., Newark 5, N.J.
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Sola Elec. Co., 4633 W. 16th St., Chicago 50, Ill.
Westinghouse Elec. Corp., E. Pittsburgh, Pa.

RELAYS, ELECTRICAL (See also STARTERS)

Allen-Bradley Co., Milwaukee 4, Wis.
Automatic Switch Co., 41 E. 11th St., N.Y.C. 3
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.
R. W. Cramer Co., Centerbrook, Ct.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Thomas A. Edison, Inc., W. Orange, N.J.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.
Jefferson Elec. Co., 25th Ave. & Madison St., Bellwood, Ill.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Paragon Elec. Co., 1600-12th St., Two Rivers, Wis.
Partlow Corp., 2 Campion Rd., New Hartford, N.Y.
Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
Photoswitch, Inc., 77 Broadway, Cambridge 42, Mass.
Potter & Brumfield Sales Co., 549 W. Washington Blvd., Chicago 6, Ill.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Roller-Smith Div., Realty & Industrial Corp., Bethlehem, Pa.
Signal Engrg. & Mfg. Co., 154 W. 14th St., N.Y.C. 11
Spencer Thermostat Co., Unit of Metals & Controls Corp. 34 Forest St., Attleboro, Mass.
Struthers-Dunn, Inc., 150 N. 13th St., Phila. 7, Pa.
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon, N.Y.
Westinghouse Elec. Corp., Plane & Orange Sts., Newark 1, N.J.
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

RELIEF VALVES (See PRESSURE RELIEF VALVES)

RESINS, PLASTIC (See PLASTIC MOLDING COMPOUNDS)

RESISTORS, ELECTRICAL

Clarostat Mfg. Co., Inc., 130 Clinton St., Brooklyn 2, N.Y.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
I-T-E Circuit Breaker Co., 19th & Hamilton Sts., Phila. 30, Pa.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls. Ind.
Ohio Carbon Co., 12508 Berea Rd., Cleveland 11, O.
Ohmite Mfg. Co., 4835 W. Flournoy St., Chicago 44, Ill.
Sprague Products Co., Marshall St., N. Adams, Mass.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.
Edwin L. Wiegand Co., 7506 Thomas Blvd., Pittsburgh 8, Pa.

RESPIRATORS (See GAS MASKS)

RETARDERS, CORROSION (See INHIBITORS)

RETARDERS, DOUGH (See DOUGH RETARDERS)

RETURN BENDS

James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Fitzsimons Mfg., 3775 E. Outer Dr., Detroit 12, Mich.
Grabler Mfg. Co., 6565 Broadway, Cleveland 5, O.
Henry Valve Co., Melrose Park, Ill. *(p. 106)*
Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
C. F. Moores Co., 1123 Ivy Hill Rd., Wyndmoor, Phila. 18, Pa.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17

REVOLUTION COUNTERS (See COUNTING & COMPUTING DEVICES)

RHEOSTATS

Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Ohmite Mfg. Co., 4835 W. Flournoy St., Chicago 44, Ill.
Westinghouse Elec. Co., 4454 Genesee St., Buffalo 5, N.Y.

RINGS, LUBRICANT RETAINER

Graphite Metallizing Corp., 1050 Nepperhan Ave., Yonkers 3, N.Y.
Messinger Bearings, Inc., D St. above Erie Ave., Phila. 1, Pa.

RINGS, PIPE WELDING

Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

RINGS, RUBBER, FOR REFRIGERATED FIXTURES

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Ball Bros. Co., Muncie, Ind.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)

RIVET TREATING CABINETS (See also REFRIGERATORS, LABORATORY & INDUSTRIAL)

Ace Cabinet Corp., New Bedford, Mass.
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
Sweden Freezer Mfg. Co., 1140 W. 53rd St., Seattle 7, Wash.

RIVETS

Aluminum Co. of America, Pittsburgh 19, Pa.
W. Ames & Co., Jersey City, N.J.
Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Cherry Rivet Co., 231 Winston St., Los Angeles 13, Cal.
Clark Bros. Bolt Co., Milldale, Ct.
Crescent Belt Fastener Co., 247 Park Ave., N.Y.C. 17
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del. (Explosive)
Fansteel Metallurgical Corp., N. Chicago, Ill. (Contact)
General Plate Div., Metals & Controls Corp., Attleboro, Mass. (Precious Metals)
International Nickel Co., 67 Wall St., N.Y.C. 5 (Nickel Alloy)
Interstate Drop Forge Co., 4035 N. 27th St., Milwaukee, Wis.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind. (Precious Metals)
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

ROD, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Colonial Alloys Co., Ridge Ave. & Crawford St., Phila. 29, Pa.
Mueller Brass Co., Port Huron, Mich.
Permanente Products Co., 1924 Broadway, Oakland 12, Cal.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.

ROD, BRASS, BRONZE & COPPER

American Brass Co., Waterbury 88, Ct.
Bristol Brass Corp., Bristol, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Lewin Metals Div., Lewin-Mathes Co., 1111 Chouteau, St. Louis, Mo.
Mueller Brass Co., Port Huron, Mich.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Titan Metal Mfg. Co., Bellefonte, Pa.

ROD, DRILL

Bethlehem Steel Co., Bethlehem, Pa.
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Jessop Steel Co., Green St., Washington, Pa.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Vanadium-Alloys Steel Co., Latrobe, Pa.

ROD, NICKEL & NICKEL ALLOY

Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C.
Jessop Steel Co., Green St., Washington, Pa.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.

ROD, STAINLESS STEEL

Arcos Corp., 1515 Locust St., Phila. 2, Pa.
Audubon Wire Cloth Corp., Richmond St. & Castor Ave., Phila. 34, Pa.
Bethlehem Steel Co., Bethlehem, Pa.
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Jessop Steel Co., Green St., Washington, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.

ROD, THREADED

Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Columbus Bolt Wks. Co., 291 Marconi Blvd., Columbus 16, O.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 51, Mo.

ROD, TIE

W. Ames & Co., Jersey City, N.J.
Bethlehem Steel Co., Bethlehem, Pa.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.

ROLLERS, RUBBER

Bushings, Inc., 4358 Coolidge Highwy., Royal Oak, Mich.
Dayton Rubber Mfg. Co., 2342 W. Riverview Ave., Dayton 1, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos Manhattan, Inc., 61 Willett St., Passaic, N.J.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)

ROLLERS, SLIDING DOOR

Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.

ROOF COOLING SYSTEMS

April Showers Co., 4126-8th St., N.W., Washington 11, D.C. *(p. 186)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Spraying Systems Co., 4021 W. Lake St., Chicago 24, Ill.
Water Cooling Corp., 71Nassau St., N.Y.C. 7
Water Cooling Equip. Corp., Afton Sta., St. Louis 23, Mo.

RUBBER, SYNTHETIC, & RUBBER-LIKE PLASTICS

Buffalo Weaving & Belting Co., 262 Chandler St., Buffalo 7, N.Y.
Canfield Rubber Co., 708 Railroad Ave., Bridgeport 5, Ct.
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
E. I. du Pont de Nemours & Co., Inc., Wilmington 98, Del.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Haveg Corp., Marshallton, Del.
Inland Mfg. Div., Gen'l Motors Corp., Dayton 1, O.
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Mack Molding Co., Ryerson Ave., Wayne, N.J.
National Motor Bearing Co., Inc., Redwood City, Cal.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Presstite Engrg. Co., 3900 Chouteau Ave., St. Louis 10, Mo.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

RUBBER PRODUCTS, CELLULAR

Dayton Rubber Mfg. Co., 2342 W. Riverview Ave., Dayton 1, O.
Dryden Rubber Co., 1014 S. Kildare Ave., Chicago 24, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Sponge Rubber Products Co., 106 Derby Place, Shelton, Ct. *(p. 188)*
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.

RUBBER PRODUCTS, EXTRUDED

Ball Bros. Co., Muncie, Ind.
Canfield Rubber Co., 708 Railroad Ave., Bridgeport 5, Ct.
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
Dryden Rubber Co., 1014 S. Kildare Ave., Chicago 24, Ill.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. *(p. 89)*
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
National Motor Bearing Co., Inc., Redwood City, Cal.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

April Showers
PATENTED AUTOMATIC ROOF COOLING

SCIENTIFICALLY CONTROLLED ROOF SPRAY SYSTEMS

- Eliminate solar heat effect
- Preserve roofing materials
- Reduce air conditioning load
- Minimize water requirements

Specially designed spray nozzles, thermostats and piping of non-ferrous construction keep the roof wetted as required to hold its temperature within a few degrees of the wet bulb.

APRIL SHOWERS COMPANY, INC.
4126 Eighth St., N.W. Washington 11, D.C.
Write for dealer plan and discounts

RUBBER— GORSYN —SYNTHETIC
Molded and Lathe Cut Specialties to Individual Specification.
Grommets
Motor Mounts
"O" Ring Packings
Terminal Insulators
Gaskets and Washers
Featuring odorless and tasteless compounds and compounds developed to resist high and low temperatures.
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Our Machine Shop Available For Your Special Mold Designs
GOSHEN RUBBER AND MANUFACTURING CO.
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THE GENERAL TIRE & RUBBER CO.

MAKERS OF AMERICA'S TOP QUALITY TIRE

Mechanical Goods Division • Wabash, Indiana

GENERAL *Engineered Rubber Parts*

Rubber parts for refrigerating and air-conditioning equipment, made to your specification and General quality standards. General engineers offer skilled help in selecting the right rubber and design for top efficiency.

RUBBER DOOR GASKETS

Extruded rubber gaskets in any design and size, solid or hollow, made accurately to specification. Natural or synthetic rubber. Our method of compounding produces gaskets that are odorless, tasteless and non-staining, with excellent compressibility. Shown above are gaskets supplied by us to leading refrigerator makers.

BONDED-TO-METAL PARTS

General is experienced in making rubber-bonded-to-metal parts of all kinds—hard and soft rubber knobs, cooler lids, fountain fittings, bumpers, and other parts.

MOLDED RUBBER PARTS

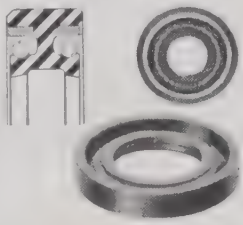
General is equipped to supply all types of molded rubber parts in any size and design, made to specification, of hard and soft rubber. Dust seal and rubber bushing shown are typical of many refrigerator parts molded by General.

DIE-CUT RUBBER PARTS

General is a reliable source of supply for die-cut rubber parts—washers, gaskets, etc. Made to your specification, to close tolerances.

OIL AND REFRIGERANT SEALS

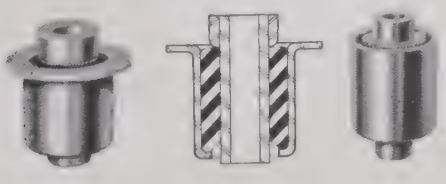
Shown are single and double-lip oil seals designed by General for highest efficiency and service. General engineers can help you design seals for lubricants and refrigerants to meet your special needs.



Precision Vibration Control with

GENERAL SILENTBLOC MOUNTINGS

GENERAL Silentbloc Mountings offer precision control of vibration and noise in motors, compressors, fans and pumps. Engineered to conform to any curve of deflection, they give complete control of vibration through starting and operating cycles. Silentbloc performance is due to its patented construction—elongation and confinement of rubber between metal sleeves. Radial compressive force, with even distribution of stress, assures indestructible rubber-to-metal adhesion and long service.



Silentbloc Mountings can solve your problems of vibration and noise in domestic and commercial equipment. Made of any metal, in any size to carry loads of ounces to tons. Simple to incorporate in your designs, easy to install. Write for Silentbloc booklet.

Temperature Seals--Gasketing--Cushioning--Insulation

• Spongex*. **General Properties:** Spongex is a resilient, spongy, blown form of rubber. Flexible, elastic, compressible and light in weight, it has low thermal conductivity. Noise absorptive, it is an efficient medium for sealing, gasketing, insulating, vibration dampening and cushioning.

Cell-Tite*. **General Properties:** Cell-Tite is a fine-cell-structure, cellular rubber material different from Spongex in that its cells are non-connecting. It is a medium proved excellent for insulating against heat, cold, air, moisture and sound and is manufactured in both soft and hard forms. Soft Cell-Tite, like Spongex in feel and appearance, is particularly adaptable to uses which utilize its non-absorbent properties. Hard Cell-Tite is as light as cork and possesses superior insulating and wearing qualities.

**Trade Mark Reg. U. S. Pat. Off*

Sponge Rubber Products Co.

267 Derby Place, Shelton, Connecticut

Plants in Derby and Shelton, Conn.

Sales Offices: New York • Chicago • Detroit • Washington

WORLD'S LARGEST MANUFACTURER
OF CELLULAR RUBBER AND BONDED FIBRE PRODUCTS

RUBBER PRODUCTS, HARD

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Luzerne Rubber Co., Trenton 9, N.J.
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.

RUBBER PRODUCTS, MECHANICAL BONDED

Apex Molded Products Co., 3574 Ruth St., Phila. 34, Pa.
Bushings, Inc., 4358 Coolidge Highwy., Royal Oak, Mich.
Dryden Rubber Co., 1014 S. Kildare Ave., Chicago 24, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Korfund Co., Inc., 48-51-32nd Place, Long Island City 1, N.Y. *(p. 237)*
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
National Motor Bearing Co., Inc., Redwood City, Cal.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
John M. Watt's Sons, 112 Walnut St., Phila., Pa.

RUBBER PRODUCTS, MOLDED

Apex Molded Products Co., 3574 Ruth St., Phila. 34, Pa.
Ball Bros. Co., Muncie, Ind.
Buffalo Weaving & Belting Co., 262 Chandler St., Buffalo 7, N.Y.
Canfield Rubber Co., 708 Railroad Ave., Bridgeport 5, Ct.
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
Dryden Rubber Co., 1014 S. Kildare Ave., Chicago 24, Ill.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
Garlock Packing Co., 402 E. Main St., Palymra, N.Y.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
National Motor Bearing Co., Inc., Redwood City, Cal.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Standard Products Co., 505 Blvd., Bldg., Detroit 2, Mich.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.

RUPTURE MEMBERS (See PRESSURE RELIEF DEVICES)

SADDLES, PIPE (See also FITTINGS)

Thomas Beckett & Co., Inc., 2118 Griffin St., Dallas 2, Tex.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.

SAFETY SHUT-OFFS FOR GAS REFRIGERATORS

Spencer Thermostat Co., Unit of Metals & Controls Corp., 34 Forest St., Attleboro, Mass.

SAFETY VALVES (See PRESSURE RELIEF VALVES)

SALAD REFRIGERATORS (See REFRIGERATORS, SALAD)

SCALE & CORROSION INHIBITORS (See INHIBITORS)

SCALE TRAPS (See also STRAINERS; also FILTERS, etc.)

Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. *(p. 203)*
Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Remco, Inc., 49th St. & A.V.R.R., Pittsburgh 1, Pa.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 165)*

SCREENS, LOUVRE

Ingersoll Steel Div., Borg-Warner Corp., 310 S. Michigan, Chicago, Ill.
Kentucky Metal Products Co., Preston St. & Audubon Park, Louisville 4, Ky.

SCREENS, PERFORATED METAL (See METAL, PERFORATED)

SCREENS, SPECIAL

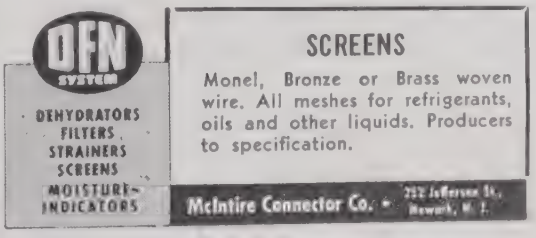
Audubon Wire Cloth Corp., Richmond St. & Castor Aye., Phila. 34, Pa.
Bonded Scale Co., 2176 S. 3rd St., Columbus 7, O.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 189)*
Western Engine Co., 1101 Meridian Ave., Alhambra, Cal.

SCREENS, VIBRATING

Bonded Scale Co., 2176 S. 3rd St., Columbus 7, O.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.

SCREENS, WIRE (See also WIRE CLOTH)

Audubon Wire Cloth Corp., Richmond St. & Castor Ave., Phila. 34, Pa.
Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp. Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 189)*
Michigan Wire Cloth Co., 2098 Howard St., Detroit 16, Mich.
Newark Wire Cloth, 351 Verona Ave., Newark 4, N.J.



SCREW MACHINE PARTS (See also THREADED SPECIALTIES)

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Aircraft Screw Products Co., Inc., 47-23-35th St., Long Island City 1, N.Y.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Corbin Screw Corp., New Britain, Ct.
Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia.
Eastern Machine Screw Corp., Truman & Barclay Sts., New Haven 6, Ct.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Mueller Brass Co., Port Huron, Mich.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
William H. Ottemiller Co., Pattison St. & M&P R.R., York, Pa.
Ross Sprinkler Co., 34 Roberts St., Pasadena 3, Cal.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Standard Valve Mfg. Co., 817 Albany St., Boston 19, Mass.

SCREWS (See also particular type)

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Clark Bros. Bolt Co., Milldale, Ct.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.J.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREW, CAP

Aircraft Screw Products Co., Inc., 47-23-35th St., Long Island City 1, N.Y.
Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bristol Co., Waterbury 91, Ct.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Clark Bros. Bolt Co., Milldale, Ct.
Corbin Screw Corp., New Britain, Ct.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Wm. H. Haskell Mfg. Co., 24 Commerce St., Pawtucket, R.I.
Holo-Krome Screw Corp., Hartford 10, Ct.
William H. Ottemiller Co., Pattison St. & M&P R.R., York, Pa.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREWS, DRIVE

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREWS, LAG

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Corbin Screw Corp., New Britain, Ct.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Wm. H. Haskell Mfg. Co., 24 Commerce St., Pawtucket, R.I.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREW, LOCK; LOCKING; LOCK WASHER

Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Shakeproof, Inc., 2401 N. Keeler Ave., Chicago 39, Ill.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREWS, MACHINE

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Corbin Screw Corp., New Britain, Ct.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Holo-Krome Screw Corp., Hartford 10, Ct.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREWS, PHILLIPS HEAD

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

To Get Satisfied Customers . . . BE ONE YOURSELF!

Specify Baker

FAMOUS FOR 42 YEARS AND IN 68 COUNTRIES

NEW MODELS

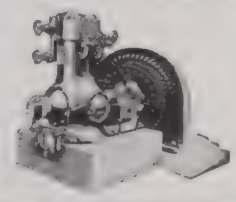
Scheduled for Production in 1947

Baker Models F-82, F8A, F69

4-Cylinder Ammonia Compressors
30 to 80 tons capacity.
Write for details

PRODUCTS: Ammonia Compressors; Self-contained Ammonia Units; Self-Contained "Freon-12" Units; Shell and Tube Condensers, Vertical or Horizontal; Refrigeration Valves and Fittings; Water or Brine Coolers, Shell and Tube or Multi-Unit Type; Coils and Cooler Units of all sizes; Automatic Refrigeration Controls and Accessories; Evaporative Type Condensers.

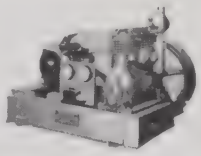
BAKER AMMONIA COMPRESSORS



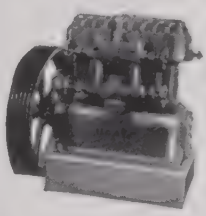
From 1 to 100 tons capacity. V-belt drive or direct connection to motors or engines. Vertical enclosed single-acting type. Duplex or multiple installations to obtain any desired capacity.

"FREON-12" COMPRESSOR UNITS

Arranged for use with evaporative type condenser or separately mounted shell and tube condensers. Sizes from 3 h.p. to 20 h.p. Two and four cylinder types. Automatic controls.

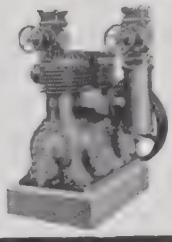


BAKER AMMONIA COMPRESSOR UNITS



Model F6B from 7½ to 20 h.p. Bore and stroke 3½"x 3½". Four cylinders, reciprocating single acting type. Force feed lubrication. Timken Roller Bearings. This model available in single or multiple compressor or condensing units as desired.

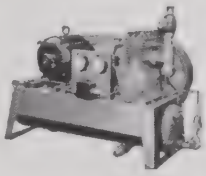
BAKER AMMONIA BOOSTER COMPRESSORS



For sub-zero temperature work. Lubrication under pressure. Single or multiple compressor installations. V-belt or direct drive. Large gas manifold at compressor suction ports allows complete filling of cylinder at variable speeds.

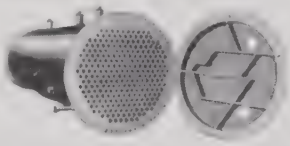
BAKER "FREON-12" UNITS

"Freon-12" Units of the self-contained automatic type from ¼ to 20 h.p. capacity in single units. Furnished in air-cooled type ½ to 20 h.p. and water-cooled type 3 to 20 h.p.



BAKER SHELL AND TUBE CONDENSERS AND COOLERS

ASRE or National Board Code welded single units to 150 tons, multiple units any capacity. Shells 8" or 50" diameter. Removable heads. Vertical or horizontal, single pass or multipass. Brine or water.



FOR FURTHER INFORMATION WRITE TO

BAKER ICE MACHINE CO., INC.

SOUTH WINDHAM, MAINE, and OMAHA, NEBRASKA

MANUFACTURERS OF REFRIGERATION AND AIR CONDITIONING EQUIPMENT SINCE 1905

SCREWS, SELF-TAPPING

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Corbin Screw Corp., New Britain, Ct.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Shakeproof, Inc., 2501 N. Keeler Ave., Chicago 39, Ill.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREWS, SET

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Bristol Co., Waterbury 91, Ct.
Clark Bros. Bolt Co., Milldale, Ct.
Corbin Screw Corp., New Britain, Ct.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Holo-Krome Screw Corp., Hartford 10, Ct.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
William H. Ottemiller Co., Pattison St. & M&P R.R., York, Pa.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREWS, SHEET METAL

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Corbin Screw Corp., New Britain, Ct.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREWS, SPECIAL

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Bethlehem Steel Co., Bethlehem, Pa.
Bristol Co., Waterbury 91, Ct.
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Clark Bros. Bolt Co., Milldale, Ct.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Holo-Krome Screw Corp., Hartford 10, Ct.
Mound Tool Co., 1203 S. 7th St., St. Louis 4, Mo.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
William H. Ottemiller Co., Pattison St., & M&P R.R., York, Pa.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Russell, Burdsall & Ward Bolt & Nut Co., Port Chester, N.Y.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREWS, THUMB

Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St. Chicago 10, Ill.

SCREWS, WELD

Atlas Bolt & Screw Co., 1108 Ivanhoe Rd., Cleveland 10, O. *(p. 33)*
Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCREWS, WOOD

Central Screw Co., 3501 Shields Ave., Chicago 9, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
Pheoll Mfg. Co., 5700 Roosevelt Rd., Chicago 50, Ill.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.

SCROLLS, BLOWER (See BLOWER SCROLLS)

SCUTTLE BUTT TANKS

Richard M. Armstrong Co., Box 188, W. Chester, Pa *(p. 125)*
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*

SEALING COMPOUNDS (See COMPOUNDS)

SEALS, DOOR BOTTOM (See also DOOR GASKETS)

Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
W. J. Dennis & Co., 1732 N. Kolmar Ave., Chicago 39, Ill.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Jarrow Products, 420 N. La Salle St., Chicago 10, Ill. *(p. 89)*
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

SEALS, OIL

Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
National Motor Bearing Co., Inc., Redwood City, Cal.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Sealol Corp., 45 Willard Ave., Providence 5, R.I.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.

SEALS, RUBBER

Apex Molded Products Co., 3574 Ruth St., Phila. 34, Pa.
Ball Bros. Co., Muncie, Ind.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind ("O" Ring) *(p. 186)*
Inland Mfg. Div., Gen'l. Motors Corp., Dayton 1, O.
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
National Motor Bearing Co., Inc., Redwood City, Cal.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

SEALS, SHAFT (See also PACKING)

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Bearium Metals Corp., 268 State St., Rochester, N.Y.
Bridgeport Thermostat Co., Inc., 1225 Connecticut Ave., Bridgeport 1, Ct.
Chicago Seal Co., 232 S. Hoyne Ave., Chicago 20, Ill.
Clifford Mfg. Co., 564 E. 1st St., Boston 27, Mass. (*p. 2nd*)
Crane Packing Co., 1800 Cuyler Ave., Chicago 13, Ill.
Durametallic Corp., 2104 Factory St., Kalamazoo 24F, Mich.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Tire & Rubber Co., Garfield St., Wabash, Ind. (*p. 187*)
Johns-Manville, 22 E. 40th St., N.Y.C. 16 (*p. 145*)
Linear, Inc., State Rd. & Levick St., Phila. 35, Pa.
National Motor Bearing Co., Inc., Redwood City, Cal.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Rotary Seal Co., 2020 N. Larrabee St., Chicago 14, Ill.
Sealol Corp., 45 Willard Ave., Providence 5, R.I.
Speer Carbon Co., St. Marys, Pa.
U. S. Graphite Co., Saginaw, Mich.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.

SEAMLESS TUBING (See TUBING)

SEATS, COMPRESSOR VALVE

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Chicago Seal Co., 232 S. Hoyne Ave., Chicago 20, Ill.

SELF-SERVICE REFRIGERATORS (See DISPLAY CASES)

SEPARATORS (See particular type, i.e., OIL; also STEAM, etc.)

SHAFTS & SHAFTING

AC Spark Plug Div., Gen'l. Motors Corp., Flint 2, Mich.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Link-Belt Co., 519 Holmes Ave., Indpls. 6, Ind.
Modern Machine Wks., Inc., 5350 S. Kirkwood Ave., Cudahy, Wis. (*p. 193*)
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Spicer Mfg. Co., Div. of Dana Corp., 4100 Bennett Rd., Toledo 1, O.

SHAFT COUPLINGS (See also FLEXIBLE COUPLINGS)

Ajax Flexible Coupling Co., Inc., Westfield, N.Y.
Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Bushings, Inc., 4358 Coolidge Highwy., Royal Oak, Mich.
Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Crocker-Wheeler Div., Joshua Hendy Iron Wks., Ampere, N.J. (Flexible)
Diamond Chain Co., Inc., 402 Kentucky Ave., Indpls. 7, Ind.

SHAFTS

Machined to your specification on a production basis. Send us your prints. Specialized in this work.

MODERN MACHINE WORKS INC.
5350 S. Kirkwood Ave.
CUDAHY, WIS.

Guardian Products Corp., 1221 E. 2nd St., Michigan City, Ind.
Jeffrey Mfg. Co., 887 N. 4th St., Columbus 16, O.
Link-Belt Co., 220 S. Belmont Ave., Indpls. 6, Ind.
Link-Belt Co., 519 Holmes Ave., Indpls. 6, Ind.
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
Morse Chain Co., Div. of Borg-Warner Corp., Ithaca, N.Y.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Ramsey Chain Co., Inc., 900 Broadway, Albany 1, N.Y.
Twin Disc Clutch Co., Racine, Wis.
J. A. Zurn Mfg. Co., Erie, Pa.

SHAFT SEALS (See SEALS)

SHAPES (See also particular type)

American Brass Co., Waterbury 88, Ct.
Bethlehem Steel Co., Bethlehem, Pa.
Brasco Mfg. Co., Harvey, Ill.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown N.Y.
Henry Disston & Sons, Inc., Tacony, Phila. 35, Pa. (Hot Rolled only)
Charles W. Krieg Co., 48 Dickerson St., Newark 4, N.J.
Pressed Steel Tank Co., 1471 S. 66th St., Milwaukee 14, Wis. (*p. 194*)
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Standard Steel Specialty Co., Beaver Falls, Pa.

SHAPES, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Brasco Mfg. Co., Harvey, Ill.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown N.Y.
Charles W. Krieg Co., 48 Dickerson St., Newark 4, N.J.
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.

SHAPES, BRASS, BRONZE & COPPER

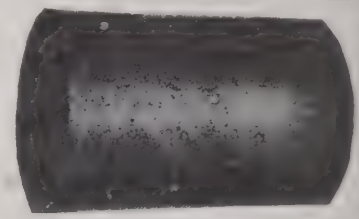
American Brass Co., Waterbury 88, Ct.
Bohn Aluminum R Brass Corp, E. Maumee, Adrian, Mich.
Brasco Mfg. Co., Harvey, Ill.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32 Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Charles W. Krieg Co., 48 Dickerson St., Newark 4, N.J.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Titan Metal Mfg. Co., Bellefonte, Pa.

SHAPES, DRAWN OR PRESSED (See Also STAMPINGS; also SHELLS)

Ackermann Mfg. Co., Wheeling, W. Va.
Bingham Stamping Co., 1062 Post St., Toledo, O.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Charles W. Krieg Co., 48 Dickerson St., Newark 4, N.J.
Lansing Stamping Co., 1159 S. Pennsylvania Ave., Lansing 2, Mich.
Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17 (*p. 197*)
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Powell Pressed Steel Co., Hubbard, O.
Pressed Steel Tank Co., 1471 S. 66th St., Milwaukee 14, Wis. (*p. 194*)
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Stanley Wks., 195 Lake St., New Britain, Ct.
Toledo Stamping & Mfg. Co., 99 Fearing Blvd., Toledo 7, O.
Worcester Pressed Steel Co., 100 Barber Ave., Worcester 6, Mass.

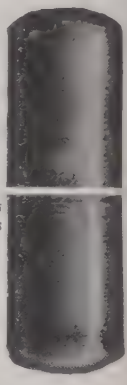
DEEP DRAWN SHAPES AND SHELLS AND CONTAINERS

Hackney Seamless Drawn Shells are ideal for use as condenser shells and refrigerant gas receivers. Efficient performance is due to their smooth surface, elimination of scale, neat appearance, comparative light weight and economy. Pioneers in the cold drawing of seamless containers from metal plates, Pressed Steel Tank Company has had more than 45 years' experience in designing, engineering and manufacturing containers for gases, liquids and solids. Our engineers will gladly work with you in developing containers or special shapes to your individual needs. For full details, write us today.



An entirely seamless shell 12" x 20" for 300 lbs. W.P.

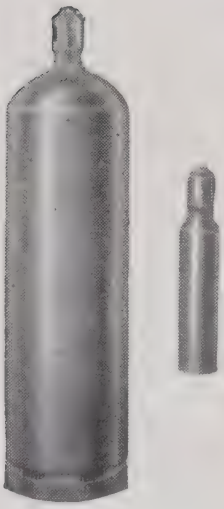
Tapered Diffuser tube, deep-drawn, seamless, 4" x 24"



Condenser shells or liquid receivers advantageously made by the Hackney Process

A seamless tapered shell

REFRIGERANT COMPRESSED GAS CYLINDERS



Hackney light tare weight cylinders are widely used for anhydrous ammonia, freon, sulphur dioxide, methyl chloride, iso-butane, propane, oxygen, etc.—in capacities from 25 pounds up. They have been designed with the users' requirements in mind—constructed to give full protection to product, ease of handling, utmost resistance to transportation and handling abuse and lowest transportation costs.

Hackney refrigeration cylinders are of two-piece construction or are entirely seamless—formed by a series of cupping cold drawing operations—and subjected to special laboratory controlled heat-treatment, after complete frabrication, to further improve physical qualities. Hackney cold drawing process provides uniform sidewall thickness and eliminates excess material.

All Hackney ICC Cylinders are manufactured to specifications that more than comply with the minimum requirements set up under Interstate Commerce Commission Regulations. Write Pressed Steel Tank Company for full information.

Hackney MILWAUKEE

Pressed Steel Tank Company

Manufacturers of Hackney Products

1483 S. 66th St., Milwaukee 14
208 S. La Salle St., Room 2083, Chicago 4
1389 Vanderbilt Concourse Bldg., New York 17
566 Roosevelt Bldg., Los Angeles 14
231 Hanna Bldg., Cleveland 15

SHAPES, EXTRUDED

Alpha Metals, Inc., 363 Hudson Ave., Brooklyn 1, N.Y. (Lead)
American Brass Co., Waterbury 88, Ct.
Bohn Aluminum & Brass Corp., E. Maumee, Adrian, Mich.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.
Titan Metal Mfg. Co., Bellefonte, Pa.
R. D. Werner Co., Inc., 295-5th Ave., N.Y.C. 16

SHAPES, STRUCTURAL (See also ANGLES, etc.)

Bethlehem Steel Co., Bethlehem, Pa.
Brasco Mfg. Co., Harvey, Ill.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Jones & Laughlin Steel Co., Pittsburgh, Pa.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
Weirton Steel Co., Weirton, W. Va.

SHAPES, VULCANIZED FIBRE (See FIBRE)

SHARP FREEZERS (See also BLAST FREEZERS; also QUICK FREEZERS)

W. J. Finnegan Co., 7402 Santa Monica Bldg., Los Angeles 46, Cal. *(p. 116)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
King Co., 902 N. Cedar St., Owatonna, Miun.
Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*
Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.

SHEAVES (See PULLEYS)

SHEET METAL ASSEMBLY & MANUFACTURING

E. H. Allen Co., 22 Dorrance St., Boston 29, Mass.
American Rolling Mill Co., Middletown, O.
Burt Mfg. Co., 926 S. High St., Akron 11, O.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Erie Art Metal Co., 1602 E. 18 St., Erie, Pa. (Parts)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich. *(p. 126)*
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Mullins Mfg. Corp., S. Ellsworth St., Salem, O.
New York Iron Roofing & Corrugating Co., Inc., 94-1st St., Jersey City 2, N.J. *(p. 195)*
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Stanley Wks., 195 Lake St., New Britain, Ct.
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.
Wirt & Knox Mfg. Co., 23rd & York Sts., Phila. 32, Pa.

SHEET, ALLOY

Alpha Metals, Inc., 363 Hudson Ave., Brooklyn 1, N.Y. (Lead)
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Henry Disston & Sons, Inc., Tacony, Phila. 35, Pa.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Illinois Zinc Co., 2959 W. 47th St., Chicago 32, Ill.
Jessop Steel Co., Green St., Washington, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.

SHEET, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chlogo 32, Ill.
Permanente Products Co., 1924 Broadway, Oakland 12, Cal.
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.
United Smelting & Aluminum Co., Inc., New Haven, Ct.
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.

SHEET, ASBESTOS

Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Belmont Packing & Rubber Co., Butler & Sepviva Sts., Phila. 7, Pa.
Philip Carey Mfg. Co., Lockland Cin'ti. 15, O.
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Ehret Magnesia Mfg. Co., Valley Forge, Pa.
Felt Products Mfg. Co., 1508 W. Carroll Ave., Chicago 7, Ill.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Hollow Center Packing Co., 6523 Euclid Ave., Cleveland 3, O.
Janos Asbestos Co., 39 Cortlandt St., N.Y.C. 7
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
Keasbey & Mattison Co., Ambler, Pa.
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Linear, Inc., State Rd., & Levick St., Phila. 35, Pa.
Norristown Magnesia & Asbestos Co., Washington St. below Ford St., Norristown, Pa.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Ruberoid Co., 500-5th Ave., N.Y.C. 18
Standard Asbestos Mfg. Co., 820 W. Lake St., Chicago 7, Ill.
Victor Mfg. & Gasket Co., 5750 Roosevelt Rd., Chicago 90, Ill.

SHEET, BRASS, BRONZE, COPPER

American Brass Co., Waterbury 88, Ct.
Bristol Brass Corp., Bristol, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.

The New York Iron Roofing and Corrugating Co., Inc.
94 1st Street, Jersey City 2, N.J.
SHEET METAL FABRICATORS
Rector 2-7519 Jo. Sq. 2-3370

SHEET, ENAMELING

American Rolling Mill Co., Middletown, O.
Carnegie-Illinois Steel Corp., U. S. Steel Corp., Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 300 W. 51st St., Chicago 32, Ill.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.

SHEET, INCONEL, MONEL & NICKEL

Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C. 5

SHEET, STEEL (See also particular type)

American Rolling Mill Co., Middletown, O.
Bethlehem Steel Co., Bethlehem, Pa.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Jessop Steel Co., Green St., Washington, Pa.
Jones & Laughlin Steel Co., Pittsburgh, Pa.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
New York Iron Roofing & Corrugating Co., Inc., 94-1st St., Jersey City 2, N.J. *(p. 195)*
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
Weirton Steel Co., Weirton, W. Va.
Wheeling Corrugating Co., Wheeling, W. Va.
Alan Wood Steel Co., Steel Mill Rd., Conshohocken, Pa.
Youngstown Sheet & Tube Co., Youngstown, O.

SHEET, STEEL, BONDERIZED

American Rolling Mill Co., Middletown, O.
Bethlehem Steel Co., Bethlehem, Pa.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.

SHEET, STEEL, COATED

American Rolling Mill Co., Middletown, O.
Bethlehem Steel Co., Bethlehem, Pa.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
National Lead Co., 111 Broadway, N.Y.C. 6
New York Iron Roofing & Corrugating Co., Inc., 94-1st St., Jersey City 2, N.J. *(p. 195)*
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Weirton Steel Co., Weirton, W. Va.
Wheeling Corrugating Co., Wheeling, W. Va.

SHEET, STEEL, ELECTRICAL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Rolling Mill Co., Middletown, O.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.

SHEET, STEEL, GALVANIZED

American Rolling Mill Co., Middletown, O.
Bethlehem Steel Co., Bethlehem, Pa.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bdlg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Jones & Laughlin Steel Co., Pittsburgh, Pa.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
New York Iron Roofing & Corrugating Co., Inc., 94-1st St., Jersey City 2, N.J. *(p. 195)*
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
Wheeling Corrugating Co., Wheeling, W. Va.

SHEET, STEEL, STAINLESS

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Rolling Mill Co., Middletown, O.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Jessop Steel Co., Green St., Washington, Pa.
New York Iron Roofing & Corrugating Co., Inc., 94-1st St., Jersey City 2, N.J. *(p. 195)*
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
Washington Steel Corp., 10 Woodland Ave., Washington, Pa.

SHEET, ZINC

New Jersey Zinc Co., 160 Front St., N.Y.C. 7

SHELLS, DRAWN OR SEAMLESS (See also SHAPES, DRAWN)

Alloy Products Corp., 1045 Perkins Ave., Waukesha, Wis.
American Brass Co., Waterbury 88, Ct.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17 *(p. 197)*
Metal Specialty Co., Este Ave. & B&O RR., Cin'ti., O.
Pressed Steel Tank Co., 1471 S. 66th St., Milwaukee 14, Wis. *(p. 194)*
Stanley Wks., 195 Lake St., New Britain, Ct.
Worcester Pressed Steel Co., 100 Barber Ave., Worcester 6, Mass.

SHELVES, REFRIGERATOR (See also RACKS; also WIRE FORMING)

Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
Hoosier Cardinal Corp., 601 W. Eichel Ave., Evansville 7, Ind.
Kentucky Metal Products Co., Preston St. & Audubon Park, Louisville 4, Ky.
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 16, Ill.
Sneath Glass Co., Hartford City, Ind.
Standard-Keil Hardware Mfg. Co., Inc., 2413 Atlantic Ave., Brooklyn 33, N.Y. *(p. 123)*
Union Steel Products Co., 448 Pine St., Albion, Mich.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
United Steel & Wire Co., Battle Creek, Mich.
Wall Wire Products Co., 11333 General Dr., Plymouth, Mich.
L. A. Young Spring & Wire Corp., 9200 Russell St., Detroit 11, Mich.

SHIELDS, EXPANSION (See also ANCHORS)

Chicago Expansion Bolt Co., 1338 W. Concord Place, Chicago 22, Ill.
U. S. Expansion Bolt Co., York, Pa.

The Linde Air Products Company

Unit of Union Carbide and Carbon Corporation

30 E. 42nd St., New York 17, N.Y. UCC Offices in Other Principal Cities

Prest-O-Lite

Compressed Gas Cylinders—Drawn Shells and Shapes—Cylinder Valves

Refrigerant Gas Cylinders

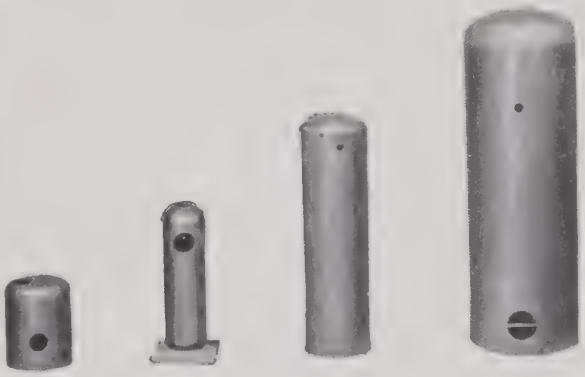
PREST-O-LITE cylinders combine the greatest strength with minimum weight commensurate with safe practice and conform in all respects to specifications of the Interstate Commerce Commission and other regulatory bodies. Fuse plugs approved by Bureau of Explosives as to type and location, are provided.

Cylinders can be supplied with McKAY diaphragm packless valves, or with PREST-O-LITE packed type valves.

PREST-O-LITE cylinders are manufactured precisely for the requirements of each type of service and are available in a variety of types and sizes of standard or special design. Typical cylinders for refrigerant gases are illustrated.

Deep-Drawn Shells and Shapes

PREST-O-LITE cold-drawn steel shells, receivers, and shapes can be made to practically any design and in a wide range of sizes. Linde engineers will be glad to help with the development of designs for your requirements. The shapes illustrated show typical receiver shells.



PREST-O-LITE Torches and Soldering Irons for brazing, heating, and soldering, are available separately or in convenient, moderately priced outfits for the refrigeration industry. These units are furnished complete for use with small PREST-O-LITE acetylene tanks.

PREST-O-LITE Halide Leak Detectors accurately and instantly locate leaks of non - combustible halide gases such as Freon and Carrene. This apparatus operates on PREST-O-LITE acetylene supplied in small tanks.

The words "McKay" and "Prest-O-Lite" are trade-marks of Units of Union Carbide and Carbon Corporation.

SHIMS

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Detroit Stamping Co., 350 Midland Ave., Detroit 3, Mich.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Laminated Shim Co., Inc., Union St., Glenbrook, Ct.
National Motor Bearing Co., Inc., Redwood City, Cal.

SHOCK ABSORBERS (See also VIBRATION ABSORBING BASES)

Apex Molded Products Co., 3574 Ruth St., Phila. 34, Pa.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
MB Mfg. Co., Inc., 1060 State St., New Haven 11, Ct.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.

SHUTTERS, AUTOMATIC (See also LOUVRES)

Arex Co., 333 N. Michigan Ave., Chicago 1, Ill.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Holcomb & Hoke Mfg. Co., Inc., 1545 Van Buren St., Indpls. 7, Ind.
Johnson Fan & Blower Corp., 1318 W. Lake St., Chicago 7, Ill.
Reed Unit-Fans, Inc., 1001 St. Charles Ave., New Orleans 8, La.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

SIGHT GLASSES (See LIQUID INDICATORS)

SIGNALS, ELECTRIC (See also ALARMS)

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon, N.Y.

SIGNS (See also NAME PLATES)

Acromark Co., 5 Morrell St., Elizabeth 4, N.J.
Artkraft Mfg. Corp., Kibby St. & D.T.&I. R.R., Lima, O.
Mine Safety Appliances Co., Braddock, Thomas & Meade Sts., Pittsburgh, Pa.

SILICA GEL (See DEHYDRANTS)

SILVER

Handy & Harman, 82 Fulton St., N.Y.C. 7
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.

SLEEVES, CYLINDER

McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.

SLIDES, DRAWER

Brasco Mfg. Co., Harvey, Ill.
Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.

SODA FOUNTAINS & FOUNTAINETTES

Ace Cabinet Corp., New Bedford, Mass.
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 4, Ill.
Horace A. Carter, Inc., 16 E. Marshall St., Richmond 19, Va.
Stanley Knight Corp., 3430 N. Pulaski Rd., Chicago 41, Ill.
Liquid Carbonic Corp., 3100 S. Kedzie Ave., Chicago 23, Ill.
C. Nelson Mfg. Co., 4016 N. Union St., St. Louis 15, Mo.
Piper Co., Govans Post Office, Baltimore 2, Md.
Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.

SODA FOUNTAIN CONVERSION UNITS

Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*

SODA FOUNTAIN EQUIPMENT

Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Heat-X-Changer Co., Inc., 415 Lexington Ave., N.Y.C. 17 *(p. 240)*
Hudson Products Co., Inc., 4400 St. Aubin, Detroit 7, Mich.
Stanley Knight Corp., 3430 N. Pulaski Rd., Chicago 41, Ill.
Liquid Carbonic Corp., 3100 Kedzie Ave., Chicago 23, Ill.
Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Spacarb, Inc., 311 E. 23rd St., N.Y.C. 10
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
Emery Thompson Machine & Supply Co., 1349 Inwood Ave., N.Y.C. 52

SODIUM CHLORIDE

Pennsylvania Salt Mfg. Co., 1000 Widener Bldg., Phila. 7, Pa.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.

SODIUM DICHROMATE (See WATER TREATING MATERIALS)

SOFTENERS, WATER (See WATER TREATING)

SOLDER (See also SOLDER, SILVER)

Alpha Metals, Inc., 363 Hudson Ave., Brooklyn 1, N.Y.
M. W. Dunton Co., 670 Eddy St., Providence 3, R.I.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Lenk Mfg. Co., Newton Lower Falls 62, Mass.
National Lead Co., 111 Broadway, N.Y.C. 6
Northern Indiana Brass Co., 935 Plum St., Elkart, Ind.
Ruby Chemical Co., 68 McDowell St., Columbus 8, O.

SOLDER, SILVER

Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
General Plate Div., Metals & Controls Corp., Attleboro, Mass.
Handy & Harman, 82 Fulton St., N.Y.C. 7
Chas W. Krieg Co., 48 Dickerson St., Newark 4, N.J.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls. Ind.
National Lead Co., 111 Broadway, N.Y.C. 6
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

SOLENOID VALVES, BRINE

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*

SOLENOID VALVES, REFRIGERANT LIQUID

(A—Ammonia; B—Other refrigerants)
(A,B) Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
(B) Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
(B) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
(A,B) Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill. *(p. 55)*
(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(A,B) General Controls Co., 801 Allen Ave., Glendale, Cal. *(p. 199)*
(B) Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
(A,B) Henry Valve Co., Melrose Park, Ill.
(A,B) Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
(B) Penn Elec. Switch Co., Goshen, Ind.
(A,B) Refrigerating Specialties Co., 728 S. [illegible] Blvd., Chicago 12, Ill.
(A,B) Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.

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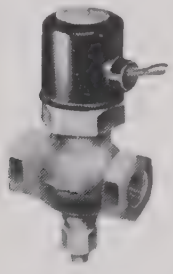

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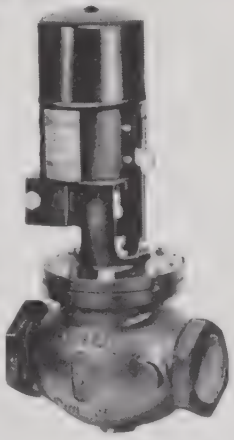

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SOLENOID VALVES, REFRIGERANT SUCTION

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(A,B) Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
(B) Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
(B) Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
(A,B) Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill. *(p. 55)*
(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(A,B) General Controls Co., 801 Allen Ave., Glendale, Cal. *(p. 199)*
(A,B) Henry Valve Co., Melrose Park, Ill. *(p. 203)*
(A,B) Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
(B) McQuay-Norris Mfg. Co., 2320 Marconi Ave., St. Louis 10, Mo.
(B) Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
(A,B) Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
(A,B) Sporlan Valve Co., 7525 Sussex Ave., St. Louis 17, Mo.

SOLENOID VALVES, WATER

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Central Automatic Sprinkler Co., 4th St. & Cannon Ave., Lansdale, Pa.
Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Hays Mfg. Co., 21st & Liberty Sts., Erie, Pa.
Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
Johnson Corp., 805 Wood St., Three Rivers, Mich.
Supreme Elec. Products Co., 194 Vassar St., Rochester 7 N.Y.

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DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Reliance Elec. & Engrg. Co., 1088 Ivanhoe Rd., Cleveland 10, O.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.
Twin Disc Clutch Co., Racine, Wis.
Westinghouse Elec. Corp., 200 McCandless Ave., Pittsburgh 1, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

SPEED NUTS & CLIPS (See FASTENERS)

SPEED REDUCERS (See also SPEED CHANGERS)

General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Janette Mfg. Co., 556 W. Monroe St., Chicago 6, Ill.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Palmer-Bee Co., 1701 Poland Ave., Detroit 12, Mich.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Phila. 20, Pa.
Star Elec. Motors Co., 200 Bloomfield Ave., Bloomfield, N.J.

SPONGE RUBBER PARTS (See RUBBER PRODUCTS, CELLULAR)

SPRAY ELIMINATORS

Farr Co., 2615 Southwest Dr., Los Angeles, Cal.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Harry Cooling Towers, Inc., West St., Doylestown, Pa

SPRAY HUMIDIFIERS (See also HUMIDIFIERS)

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Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill
Marley Co., Inc., 3001 Fairfax Rd., Kansas City 15, Kan. *(p. 79)*
Parks-Cramer Co., Box 444, Fitchburg, Mass.
Ross Sprinkler Co., 34 Roberts St., Pasadena 3, Cal.
Spraying Systems Co., 4021 W. Lake St., Chicago 24, Ill

SPRAY NOZZLES (See NOZZLES)

SPRAY PONDS (See COOLING PONDS)

SPRAY TYPE EVAPORATORS (See UNIT COOLERS)

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American Locomotive Co., 30 Church St., N.Y.C.
American Spring & Wire Specialty Co., 816 N. Spaulding Ave., Chicago 51, Ill.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Hunter Pressed Steel Co., 801 Maple St., Lansdale, Pa.
Korfund Co., Inc., 48-51-32nd Place, Long Island City 1, N.Y. *(p. 237)*
L. A. Young Spring & Wire Corp., 9200 Russell St., Detroit 11, Mich.

SPROCKETS

Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp. Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Diamond Chain Co., Inc., 402 Kentucky Ave., Indpls. 7, Ind.
Link-Belt Co., 519 Holmes Ave., Indpls. 6, Ind.
Morse Chain Co., Div. of Borg-Warner Corp., Ithaca, N.Y.

STAMPINGS (See also particular type; also SHAPES)

Ackermann Mfg. Co., Wheeling, W. Va.
Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 124)*
Alloy Products Corp., 1045 Perkins Ave., Waukesha, Wis.
American Central Mfg. Corp., Div. of Aviation Corp., 18th & Columbia Sts., Connersville, Ind.
American Emblem Co., Inc., 9 Genessee St., New Hartford, Utica 1, N.Y.
Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Bingham Stamping Co., 1062 S. Post St., Toledo, O.
A. S. Campbell & Co., Inc., E. Boston 28, Mass.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Columbia Metal Stamping Co., 1900 Harvard Ave., Cleveland, O.
Commercial Shearing & Stamping Co., 1775 Logan St., Youngstown, O.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Detroit Stamping Co., 350 Midland Ave., Detroit 3, Mich.
Erie Art Metal Co., 1602 E. 18 St., Erie, Pa.
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Fox Co., Fox Lane, Cin'ti. 23, O.
Gillian Mfg. Co., 7750 Dubois Ave., Detroit 11, Mich.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich. *(p. 126)*
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Hoosier Cardinal Corp., 601 W. Eichel Ave., Evansville 7, Ind.
Hunter Pressed Steel Co., 801 Maple St., Lansdale, Pa.
Lansing Stamping Co., 1159 S. Pennsylvania Ave., Lansing 2, Mich.
McAlear & Co., Inc., 1422 N. 8th St., Phila. 22, Pa.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.
Minerallac Elec. Co., 25 N. Peoria St., Chicago 7, Ill.
Motors Metal Mfg. Co., 5936 Milford Ave., Detroit 10, Mich.
Mullins Mfg. Corp., S. Ellsworth St., Salem, O.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Oakes North Chicago Div., Houdaille-Hershey Corp., 1900 Foss Park Ave., N. Chicago, Ill. *(p. 174)*
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.
Paul & Beekman, Inc., 1800 Courtland St., Phila. 40, Pa.
Powell Pressed Steel Co., Hubbard, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
S & S Products, Inc., Lima, O.
Shakeproof, Inc., 2501 N. Keeler Ave., Chicago 39, Ill.
Stanley Wks., 195 Lake St., New Britain, Ct.
Superior Spinning & Stamping Co., 4057 Fitch Rd., Toledo 12, O.
Toledo Stamping & Mfg. Co., 99 Fearing Blvd., Toledo 7, O.
Transue & Williams Steel Forging Corp., Alliance, O.
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.
E. R. Wagner Mfg. Co., 4001 N. 32nd St., Milwaukee, Wis.
Wall Wire Products Co., 11333 General Dr., Plymouth, Mich.
Whitehead Stamping Co., 1661 W. Lafayette Blvd., Detroit 16. Mich.
Worcester Pressed Steel Co., 100 Barber Ave., Worcester 6, Mass.
Wrought Washer Mfg. Co., 2253 S. Bay St., Milwaukee 7, Wis.

STAMPINGS, ALUMINUM

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 124)*
Aluminum Goods Mfg. Co., Manitowoc, Wis.
Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Fox Co., Fox Lane, Cin'ti. 23, O.
Gillian Mfg. Co., 7750 Dubois Ave., Detroit 11, Mich.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich. *(p.126)*
Lul Products, Inc., 2235 Sisson St., Baltimore 11, Md.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Minerallac Elec. Co., 25 N. Peoria St., Chicago 7, Ill.
Paul & Beekman, Inc., 1800 Courtland St., Phila. 40, Pa.
S & S Products, Inc., Lima, O.
Superior Spinning & Stamping Co., 4057 Fitch Rd., Toledo 12, O.
Toledo Stamping & Mfg. Co., 99 Fearing Blvd., Toledo 7, O.
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Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.
Whitehead Stamping Co., 1661 W. Lafayette Blvd., Detroit 16, Mich.
Wrought Washer Mfg. Co., 2253 S. Bay St., Milwaukee 7, Wis.

STAMPINGS, BRASS & BRONZE

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 124)*
American Brass Co., Waterbury 88, Ct.
Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Fox Co., Fox Lane, Cin'ti. 23, O.
Gillian Mfg. Co., 7750 Dubois Ave., Detroit 11, Mich.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich. *(p. 126)*
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Minerallac Elec. Co., 25 N. Peoria St., Chicago 7, Ill.
Paul & Beekman, Inc., 1800 Courtland St., Phila. 40, Pa.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
S & S Products, Inc., Lima, O.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Stanley Wks., 195 Lake St., New Britain, Ct.
Superior Spinning & Stamping Co., 4057 Fitch Rd., Toledo 12, O.
Toledo Stamping & Mfg. Co., 99 Fearing Blvd., Toledo 7, O.
Whitehead Stamping Co., 1661 W. Lafayette Blvd., Detroit 16, Mich.
Wrought Washer Mfg. Co., 2253 S. Bay St., Milwaukee 7, Wis.

STAMPINGS, CABINET DOOR

Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 97)*
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Paul & Beekman, Inc., 1800 Courtland St., Phila. 40, Pa.
Toledo Stamping & Mfg. Co., 99 Fearing Blvd., Toledo 7, O

STAMPINGS, EVAPORATOR

Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 167)*
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Metal Specialty Co., Este Ave. & B&O R.R., Cin'ti., O.
Paul & Beekman, Inc., 1800 Courtland St., Phila. 40, Pa.

STANDARDS, SHELF (See SUPPORTS)

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Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
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Electric Controller & Mfg. Co., 2700 E. 79th St., Cleveland 4, O.
Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Master Elec. Co., 126 Davis Ave., Dayton 1, O.
Spencer Thermostat Co., Unit of Metals & Controls Corp., 34 Forest St., Attleboro, Mass.
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C. H. Wheeler Mfg. Co., 1741 Sedgley Ave., Phila. 32, Pa.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

STEAM SEPARATORS

American District Steam Co., Bryant St., N. Tonawanda, N.Y.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Cochrane Corp., 17th St., below Allegheny Ave., Phila. 32, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Johnson Corp., 805 Wood St., Three Rivers, Mich.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

STEAM TRAPS

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Barnes & Jones, Inc., 128 Brookside Ave., Jamaica Plain 30, Mass.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
W. D. Cashin Co., 69 A St., S. Boston 27, Mass.
Cochrane Corp., 17th St., below Allegheny Ave., Phila. 32, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Cryer Trap & Valve Co., 366 Madison Ave., N.Y.C.
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
McAlear Mfg., Div of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Polar Hardware Co., 1631 S. Michigan Ave., Chicago 16, Ill.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 182)*
Sterling Engrg. & Mfg. Corp., 152 Business St., Hyde Park 36, Mass.
Sterling, Inc., 3738 N. Holton St., Milwaukee 12, Wis.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Trane Co., La Crosse, Wis. *(p. 14)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
Warren Webster & Co., Camden, N.J.
Williams Gauge Co., 1620 Pennsylvania Ave., Pittsburgh 12, Pa.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

STEAM TURBINES

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
DeLaval Steam Turbine Co., 853 Nottingham Way, Trenton 2, N.J.
Elliott Co., Jeannette, Pa.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Dean Hill Pump Co., 4000 E. 16th St., Indpls. 7, Ind.
Terry Steam Turbine Co., Hartford 1, Ct.
Westinghouse Elec. Corp., S. Phila. 1, Pa.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

STEEL (See particular mill forms, i.e., BAR, SHEET, etc.)

STEEL, COMPRESSOR VALVE

Athenia Steel Div., National-Standard Co., Clifton, N.J
Bethlehem Steel Co., Bethlehem, Pa.
Sandvik Steel, Inc., 111-8th Ave., N.Y.C. 11 (Swedish Steel) *(p. 230)*

STEEL PLATE FABRICATION

Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Bethlehem Steel Co., Bethlehem, Pa.
Chattanooga Boiler & Tank Co., P.O. Box 110, Chattanooga, Tenn.
Colonial Iron Wks. Co., 17643 St. Clair Ave., Cleveland 10, O.
Connery Construction Co., 2nd & Luzerne Sts., Phila. 40, Pa.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
R. Munroe & Sons Mfg. Corp., 23rd & Smallman Sts., Pittsburgh 22, Pa.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Pittsburgh-Des Moines Construction Co., Neville Island, Pittsburgh 25, Pa.
Quaker City Iron Wks., Aramingo Ave. & E. Tioga St., Phila., Pa.
Stover Steel Tank & Mfg. Co., 100 S. Hancock St., Freeport, Ill.
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walsh Holyoke Boiler Wks., Div. of Continental-United Industries Co., Inc., 110 Appleton St., Holyoke, Mass.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

STEEL, STRUCTURAL (See SHAPES, STRUCTURAL)

STERILIZERS, AIR (See AIR PURIFICATION EQUIPMENT)

STILLS, AMMONIA

H. K. Porter & Co., Inc., 49th & Harrison Sts., Pittsburgh, Pa.

STRAINERS (See also particular type & FILTERS; also SCALE TRAPS)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 159)*
W. D. Cashin Co., 69 A St., S. Boston 27, Mass.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill. *(p. 55)*
General Controls Co., 801 Allen Ave., Glendale, 1, Cal. *(p. 199)*
Henry Valve Co., Melrose Park, Ill. *(p. 203)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*

Industrial Wire Cloth Products Corp., Wayne, Mich.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Mueller Brass Co., Port Huron, Mich.
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 182)*
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Trane Co., La Crosse, Wis. *(p. 14)*
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
J. A. Zurn Mfg. Co., Erie, Pa.

STRAINERS, AIR

Alco Valve Co., 855 Kingsland Ave.,, St. Louis 5, Mo *(p. 98*
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Cryer Trap & Valve Co., 366 Madison Ave., N.Y.C.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
D'Este Div., American Chain & Cable Co., Inc., Reading, Pa.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
Sterling, Inc., 3738 N. Holton St., Milwaukee 12, Wis.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Water Cooling Corp., 71 Nassau St., N.Y.C. 7

STRAINERS, AMMONIA

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. *(p. 203)*
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 182)*
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

STRAINERS, OIL

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
W. D. Cashin Co., 69 A St., S. Boston 27, Mass.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Michigan Wire Cloth Co., 2098 Howard St., Detroit 16, Mich.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Rega Mfg. Co., 79 Mt. Hope Ave., Rochester, N.Y.
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
J. A. Zurn Mfg. Co., Erie, Pa.

STRAINERS, REFRIGERANT

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 159)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill. *(p. 55)*
General Controls Co., 801 Allen Ave., Glendale 1, Cal. *(p. 199)*
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. *(p. 203)*
Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Industrial Wire Cloth Products Corp., Wayne, Mich.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 189)*
Michigan Wire Cloth Co., 2098 Howard St., Detroit 16, Mich.
Mueller Brass Co., Port Huron, Mich.
Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Remco, Inc., 49th St. & A.V.R.R., Pittsburgh 1, Pa.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 182)*
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill.
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

HENRY

STRAINERS

Type 866

ANGLE STRAINER

Screen can be taken out for cleaning without removing strainer from line. Negligible pressure drop. Very large screen area. Will not trap oil.

Light in weight. Screen reinforced to prevent distortion. Spring tension provides seal of screen at inlet end. Flanged brass shell. Screen, 100 mesh. For other styles consult Catalog No. 98.

HENRY VALVE CO.

Melrose Park, Illinois, Suburb of Chicago.

APPROVED FOR USE BY ARMY • NAVY and MARITIME COMMISSION

STRAINERS, STEAM

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
W. D. Cashin Co., 69 A St., S. Boston 27, Mass.
Cochrane Corp., 17th St., below Allegheny Ave., Phila. 32, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Cryer Trap & Valve Co., 366 Madison Ave., N.Y.C.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
D'Este Div., American Chain & Cable Co., Inc., Reading, Pa.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 182)*
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
Sterling, Inc., 3738 N. Holton St., Milwaukee 12, Wis.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Trane Co., La Crosse, Wis. *(p. 14)*
Walworth Co., 60 E. 42nd St., N.Y.C. 17
J. A. Zurn Mfg. Co., Erie, Pa.

STRAINERS, WATER

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
American District Steam Co., Bryant St., N. Tonawanda, N.Y.
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
W. D. Cashin Co., 69 A St., S. Boston 27, Mass.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Cryer Trap & Valve Co., 366 Madison Ave., N.Y.C.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Delavan Mfg. Co., 3009-6th Ave., Des Moines 13, Ia.
D'Este Div., American Chain & Cable Co., Inc., Reading, Pa.
Dollinger Corp., 1 Centre Park, Rochester 3, N.Y.
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
O. C. Keckley Co., 400 W. Madison St., Chicago 6, Ill.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Jos. A. Martocello & Co., 229 N. 14th St., Phila. 7, Pa. *(p. 157)*
Monarch Mfg. Wks., Inc., Salmon & Westmorland Sts., Phila. 34, Pa.
Mueller Co., 512 W. Cerro Gordo St., Decatur 70, Ill.
Mueller Brass Co., Port Huron, Mich.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
Spray Engrg. Co., 114 Central St., Somerville 45, Mass.
Spraying Systems Co., 4021 W. Lake St., Chicago 24, Ill
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
Sterling, Inc., 3738 N. Holton St., Milwaukee 12, Wis.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Supreme Elec. Products Co., 194 Vassar St., Rochester 7, N.Y.
Trane Co., La Crosse, Wis. *(p. 14)*
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Water Cooling Corp., 71 Nassau St., N.Y.C. 7
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.
J. A. Zurn Mfg. Co., Erie, Pa.

STRAINERS, WELL

A. D. Cook, Inc., Lawrenceburg, Ind.
Cuno Engrg. Corp., 92 S. Vine St., Meriden, Ct.
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.

STRAPS, TUBE (See SUPPORTS, TUBE)

STRIP, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Permanente Products Co., 1924 Broadway, Oakland 12, Cal.
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.

STRIP, BERYLLIUM COPPER

Athenia Steel Div., National Standard Co., Clifton, N.J.
Beryllium Corp., P.O. Box 1462, Reading, Pa.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.

STRIP, BRASS, BRONZE, COPPER

American Brass Co., Waterbury 88, Ct.
Bristol Brass Corp., Bristol, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind. (Alloy)
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.

STRIP, INCONEL, MONEL, NICKEL

Central Steel & Wire Co., 3000 W. 51st St., Chicago 32. Ill.
Driver-Harris Co., Harrison, N.J.
International Nickel Co., 67 Wall St., N.Y.C. 5
Superior Steel Corp., Carnegie, Pa. (Monel & Nickel Clad)

STRIP, STEEL

Acme Steel Co., 2840 Archer Ave., Chicago 8, Ill.
American Rolling Mill Co., Middletown, O.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Athenia Steel Div., National-Standard Co., Clifton, N.J.
Bethlehem Steel Co., Bethlehem, Pa.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
Weirton Steel Co., Weirton W. Va.

STRIP, STEEL, COATED OR CLAD

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Rolling Mill Co., Middletown, O.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
National Lead Co., 111 Broadway, N.Y.C. 6
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Superior Steel Corp., Carnegie, Pa.
Weirton Steel Co., Weirton, W. Va.

STRIP, STEEL, GALVANIZED

Acme Steel Co., 2840 Archer St., Chicago 8, Ill.
American Rolling Mill Co., Middletown, O.

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.

STRIP, STEEL, STAINLESS

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Rolling Mill Co., Middletown, O.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Driver-Harris Co., Harrison, N.J.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Steel Corp., Carnegie, Pa.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
Washington Steel Corp., 10 Woodland Ave., Washington, Pa.

STRIP, ZINC

New Jersey Zinc Co., 160 Front St., N.Y.C. 7

STRIP HEATERS (See HEATER ELEMENTS)

STRUCTURAL SHAPES (See SHAPES, STRUCTURAL)

STRUCTURAL STEEL FABRICATION

Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Bethlehem Steel Co., Bethlehem, Pa.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Chattanooga Boiler & Tank Co., 1030 E. Main St., P.O. Box 110, Chattanooga, Tenn.
Colonial Iron Wks. Co., 17643 St. Clair Ave., Cleveland 10, O.
Gustav Glaser Co., Inc., 2 Wait St., Paterson 4, N.J.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
Pittsburgh-Des Moines Construction Co., Neville Island, Pittsburgh 25, Pa.
Stover Steel Tank & Mfg. Co., 100 S. Hancock St., Freeport, Ill.

STRUCTURE, PREFABRICATED

California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Lindsay Corp., 1740-25th Ave., Melrose Park, Ill.

STUDS (See BOLTS, STUD)

SUCTION PRESSURE REGULATING VALVES (See also TWO TEMPERATURE VALVES)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 159)*
Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
A. W. Cash Co., 540 N. 18th St., Decatur, Ill.
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
Electrimatic Div., Simoniz Co., 2100 Indiana Ave., Chicago 16, Ill. *(p. 55)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Hays Corp., Michigan City, Ind.
Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Staples & Pfeiffer, 528 Bryant St., San Francisco 7, Cal.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*

SULFUR DIOXIDE

Ansul Chemical Co., Marinette, Wis. *(p. 175)*
Eston Chemicals, Inc., 3100 E. 26th St., Los Angeles 23, Cal. *(p. 152)*
Great Western Div., Dow Chemical Co., 310 Sansome St., San Francisco 4, Cal
Virginia Smelting Co., W. Norfolk, Va. *(p. 177)*

SUPPORTS, SHELF (See also HOOKS; also HARDWARE SPECIALTIES)

Brasco Mfg. Co., Harvey, Ill.
Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
National Lock Co., 7th St. & 18th Ave., Rockford, Ill. *(p. 122)*
Standard Products Co., 505 Blvd. Bldg., Detroit 2, Mich.
Standard-Keil Hardware Mfg. Co., Inc., 2413 Atlantic Ave., Brooklyn 33, N.Y. *(p. 123)*
Wall Wire Products Co., 11333 General Dr., Plymouth, Mich.

SUPPORTS, TUBE (See also PIPE HANGERS)

Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Packless Metal Products Corp., 31 Winthrop Ave., New Rochelle, N.Y.
Paine Co., 2951 Carroll Ave., Chicago 12, Ill.

Since 1880

WOLF-LINDE AMMONIA COMPRESSORS
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WOLF-LINDE AMMONIA SUCTION PRESSURE CONTROLS

DERSCH, GESSWEIN & NEUERT, INC.
4845-47-49 West Grand Avenue, Chicago 39, Ill.

SURGE TANKS (See also ACCUMULATORS)

Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*

Downingtown Iron Wks., Downingtown, Pa. *(p. 59)*

Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*

Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*

W. J. Finnegan Co., 7402 Santa Monica Blvd., Los Angeles 35, Cal. *(p. 116)*

John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.

Pittsburgh-Des Moines Construction Co., Neveille Island, Pittsburgh 25, Pa.

Stover Steel Tank & Mfg. Co., 100 S. Hancock St., Freeport, Ill.

Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*

Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

SWING JOINTS

Barco Mfg. Co., 1801 Winnemac Ave., Chicago 40, Ill.

L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.

Dresser Mfg. Div., Dresser Industries, Inc., 490 Fisher Ave., Bradford, Pa.

Flexo Supply Co., Inc., 4649 Page Blvd., St. Louis 13, Mo.

Johnson Corp., 805 Wood St., Three Rivers, Mich.

Lunkenheimer Co., Beekman St., & Waverly Ave., Cin'ti. 14, O.

Walworth Co., 60 E. 42nd St., N.Y.C. 17

Water Cooling Corp., 71 Nassau St., N.Y.C. 7

SWITCHES, ELECTRIC (See also FLOAT SWITCHES; also STARTERS, MOTOR; also TIME SWITCHES)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*

Allen Bradley Co., Milwaukee 4, Wis.

Arrow-Hart & Hegeman Elec. Co., 103 Hawthorn St., Hartford 6, Ct.

Automatic Control Co., 1005 University Ave., St. Paul 4, Minn.

Automatic Switch Co., 41 E. 11th St., N.Y.C. 3

Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.

Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.

Bryant Elec. Co., Div. of Westinghouse Elec. Corp., 1421 State St., Bridgeport 2, Ct.

Burling Instrument Co., 253 Springfield Ave., Newark 3, N.J.

Clark Controller Co., 1146 E. 152nd St., Cleveland 10, O.

Clarostat Mfg. Co., Inc., 130 Clinton St., Brooklyn 2, N.Y.

Cook Elec. Co., 2700 Southport Ave., Chicago 14, Ill.

Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.

Drying System, Inc., 1810½ Foster Ave., Chicago 40, Ill.

Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.

General Elec. Co., 1285 Boston Ave., Bridgport 2, Ct.

General Elec. Co., 1 River Rd., Schenectady 5, N.Y.

Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.

McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.

P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.

Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*

Micro Switch Div., First Industrial Corp., Freeport, Ill.

Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*

Ohmite Mfg. Co., 4835 W. Fluornoy St., Chicago 44, Ill.

Ranco, Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 75)*

Soreng Mfg. Corp., 1901 Clybourn Ave., Chicago 14, Ill.

Spencer Thermostat Co., Unit of Metals & Controls Corp., 34 Forest St., Attleboro, Mass.

Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.

U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.

Westinghouse Elec. Corp., Beaver, Pa.

Western Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

SWITCHGEAR

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.

Automatic Switch Co., 41 E. 11th St., N.Y.C. 3

Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.

General Elec. Co., 1 River Rd., Schenectady 5, N.Y.

I-T-E Circuit Breaker Co., 19th & Hamilton Sts., Phila. 30, Pa.

Trumbull Elec. Mfg. Co., 999 Woodford Ave., Plainville, Ct.

Westinghouse Elec. Corp., E. Pittsburgh, Pa.

SWIVEL JOINTS (See SWING JOINTS)

SYNTHETIC RUBBER (See RUBBER PRODUCTS)

TACHOMETERS

Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.

Bristol Co., Waterbury 91, Ct. (Recording, Elec.)

Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.

Corbin Screw Corp., New Britain, Ct.

General Elec. Co., 1 River Rd., Schenectady 5, N.Y. (Elec.)

Ideal Industries, Inc., Sycamore, Ill.

Reliance Elec & Engrg. Co, 1088 Ivanhoe Rd., Cleveland 10, O.

Veeder-Root, Inc., Garden & Sargeant Sts., Hartford 2, Ct. (Laboratory only)

Westinghouse Elec. Corp., Plane & Orange Sts., Newark 1, N.J.

Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

TAGS (See NAME PLATES)

TANKS (See also particular type following)

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 124)*

American Rolling Mill Co., Middletown, O.

Arrow Tank Co., Inc., 16 Barnett St., Buffalo 15, N.Y. (Wood)

Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6

Bethlehem Steel Co., Bethlehem, Pa.

Calfornia Steel Products Co., Barrett & "A" Sts., Richmond, Cal.

Chattanooga Boiler & Tank Co., P.O. Box 110, Chattanooga, Tenn.

Colonial Iron Wks., 17643 St. Clair Ave., Cleveland 10, O.

Connery Construction Co., 2nd & Luzerne Sts., Phila. 40, Pa.

Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*

Falstrom Co., 13 Falstrom Court, Passaic, N.J.

Frick Co., Waynesboro, Pa. *(p. 47)*

Fruehauf Trailer Co., 10940 Harper Ave., Detroit 32, Mich. (All Transport Types)

Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.

Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.

Haveg Corp., Marshallton, Del. (Plastic)

Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.

Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.

National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17

John Nooter Boiler Wks. Co, 1426 S. 2nd St., St. Louis 4, Mo.

Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.

Pittsburgh-Des Moines Construction Co., Neville Island, Pittsburgh 25, Pa.

Quaker City Iron Wks., Aramingo Ave. & E. Tioga St., Phila., Pa.

Santa Fe Tank & Tower Co., Div. of Industrial Manufacturers, Ltd., 4820 Santa Fe Ave., Los Angeles 11, Cal. (Redwood, Douglas Fir) *(p. 80)*

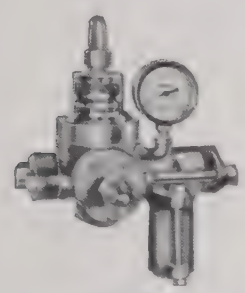
Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.

(Continued)

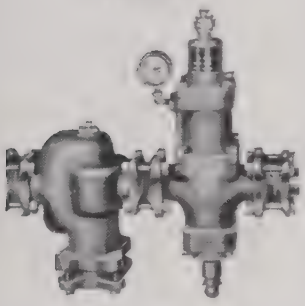
Hubbell Corporation

319 N. Albany Ave.
Chicago 12, Ill.

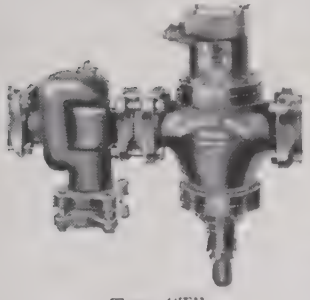
Designers and Manufacturers of Automatic Control Valves For All Refrigerants



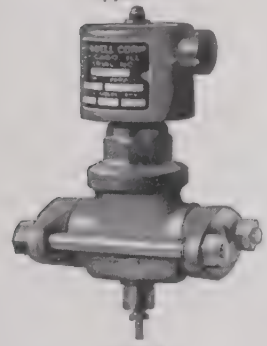
TYPE
SA-5, SF-5



TYPE
SA-6, SF-6



Type "T"



TYPE
A-2

Type SA-5, SA-6, SF-5 and SF-6 Back Pressure Regulating Valves are of the conventional type used to maintain a constant evaporator pressure.

Type SA-7, SA-8, SF-7 and SF-8 Combination Back Pressure Regulator and Stop Valves. This regulator is of the conventional type used to maintain a constant evaporator pressure and the addition of a small electric pilot valve built into the head, makes it a suction stop valve.

The DSA-9 and DSF-9 is a dual regulator which will control evaporators with two load conditions requiring different refrigerant temperatures. The diaphragms in the dual head may be set for any two evaporator pressures and will automatically change from one to the other by the opening or closing of the electric pilot valve which is built into the head.

The SAC-6 and SFC-6 valves are of the compensating type and are used where a constant temperature is desired in the medium being cooled. These valves will increase or decrease the evaporator pressure to compensate for the increase or decrease of the cooling load. These valves are made to operate with air or electricity.

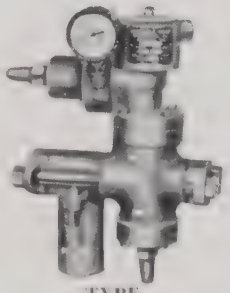
The Type "T" suction stop valve is used on installations where automatic suction line control is required. It is operated by high pressure gas and its construction makes a tight closing valve and its dependability far surpasses the conventional magnetic stop valve.

All of the valves have built-in opening stems, which eliminates the by-passes usually used for manual operation. Available with either screwed, welding type flanges or copper tube connections, in sizes from ¾″ to 8″ inclusive.

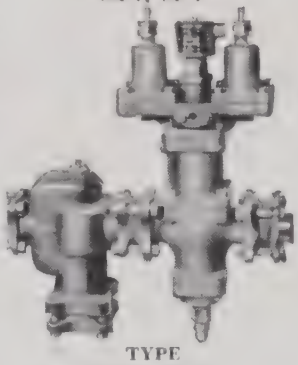
Solenoid Valves from ⅜″ to 2″ inclusive for liquids and gases with composition seat discs readily renewable, coils for any electrical characteristics, built-in lifting stem standard, available with either screwed, welding flanges or copper tube connections.

Strainers are available in all sizes for liquid and gas with very large screen areas and arranged to bolt directly to valve or with screwed, welding flange or copper tube connections for installation wherever strainer is necessary.

Write for complete information on these and our many additional Refrigeration Controls and Accessories.

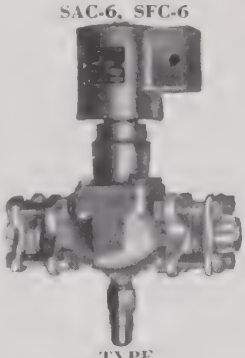


TYPE
SA-7, SF-7



TYPE
DSA-9, DSF-9

TYPE
SAC-6, SFC-6



TYPE
A-3, F-4

TANKS (Continued)

Emil Steinhorst & Sons, Inc., 612 South St., Utica 3, N.Y.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
Stover Steel Tank & Mfg. Co., 100 S. Hancock St., Freeport, Ill.
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
Walsh Holyoke Boiler Wks., Div. of Continental-United Industries Co., Inc., 110 Appleton St., Holyoke, Mass.
York Corp., York, Pa. *(p. 163)*

TANKS, BRINE, (See BRINE TANKS; also ICE TANKS)

TANKS, COPPER

Acklin Stamping Co., 1929 Nebraska Ave., Toledo 7, O. *(p. 124)*
Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
John Nooter Boiler Wks. Co., 1426 S. 2nd St. St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.

TANKS, HOT WATER

California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
R. Munroe & Sons Mfg. Corp., 23rd & Smallman Sts., Pittsburgh 22, Pa.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Rheem Mfg. Co., 570 Lexington Ave., N.Y.C. 22
Stover Steel Tank & Mfg. Co., 100 S. Hancock St., Freeport, Ill.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.

TANKS, ICE (See ICE TANKS)

TANKS, MILK HOLDING

Alloy Products Corp., 1045 Perkins Ave., Waukesha, Wis.
Beverage Engrg. & Equip. Co., 13301 Lakewood Hts. Blvd., Cleveland 7, O.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Fruehauf Trailer Co., 10940 Harper Ave., Detroit 32, Mich. (Transport)
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Gustav Glaser Co., Inc., 2 Wait St., Paterson 4, N.J.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Emil Steinhorst & Sons, Inc., 612 South St., Utica 3, N.Y.
Stover Steel Tank & Mfg. Co., 100 S. Hancock St., Freeport, Ill.

TANKS, PRESSURE (See PRESSURE VESSELS)

TANK HEADS

Ackermann Mfg. Co., Wheeling, W. Va.
Alloy Products Corp., 1045 Perkins Ave., Waukesha, Wis.
Bethlehem Steel Co., Bethlehem, Pa.
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Commercial Shearing & Stamping Co., 1775 Logan St., Youngstown, O.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
Lukens Steel Co., Coatesville, Pa.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
R. Munroe & Sons Mfg. Corp., 23rd & Smallman Sts., Pittsburgh 22, Pa.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
John Nooter Boiler Wks. Co., 1426 S. 2nd St., St. Louis 4, Mo.
Walsh Holyoke Boiler Wks., Div. of Continental-United Industries Co., Inc., 110 Appleton St., Holyoke, Mass.

TAP RODS, BEER

North Penn Co., 72-5th Ave., N.Y.C. 11

TAPE, FRICTION (See FRICTION TAPE)

TAPE, RUBBER

Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Van Cleef Bros., Inc., 7800 S. Woodlawn Ave., Chicago 19, Ill.

TEES, WELDING (See also FITTINGS, PIPE)

Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Tube-Turns, Inc., 224 E. Broadway, Louisville 1, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
J. A. Zurn Mfg. Co., Erie, Pa.

TERMINALS & TERMINAL BLOCKS

Burke Elec. Co., Erie, Pa.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Shakeproof, Inc., 2501 N. Keeler Ave., Chicago 39, Ill.
Thompson-Bremer & Co., 1640 W. Hubbard St., Chicago 22, Ill.

TERMINALS, HERMETIC

Fusite Corp., Carthage at Hannaford, Norwood, Cin'ti. 12, O.
Sperti, Inc., Norwood Sta., Cin'ti. 12, O.

TEST CABINETS (See REFRIGERATORS LABORATORY)

TESTERS, AMMONIA

La Motte Chemical Products Co., Towson, Baltimore 4, Md.

THERMOMETERS (See also particular type)

American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh 8, Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
James Berg Mfg. Co., 3707-12th Ave., Brooklyn 18, N.Y.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Electric Auto-Lite Co., Toledo 1, O.
GM Mfg. Co., 50 W. 3rd St., N.Y.C. 12
General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Henry J. Green, 1191 Bedford Ave., Brooklyn 16, N.Y.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Moore Products Co., H & Lycoming Sts., Phila. 24, Pa.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti. 12, O.
Partlow Corp., 2 Campion Rd., New Hartford, N.Y.

(Continued)

Precision Thermometer & Instrument Co.

Scientific Instruments

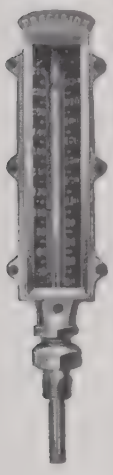
Main Office and Factory: 1442 Brandywine Street, Philadelphia 30, Pa.

Sales Offices

BALTIMORE 18, MD., 2301 North Charles Street
BOISE, IDAHO, P.O. Box 1746
BOSTON, MASS., 107 Massachusetts Avenue
CHARLESTON, W.VA., Day and Night Building
CHICAGO 11, ILL., 152 E. Superior St.
CINCINNATI 2, OHIO, 831 Temple Bar Building
CLEVELAND 3, OHIO, 6523 Euclid Avenue
DENVER 17, COLO., P.O. Box 5510
DETROIT 4, MICH., Box 517, Northwestern P.O. Station
DURHAM, N.C., P.O. Box 1367
EL PASO, TEX., 312 Mills Building
HONOLULU, H.I., P.O. Box 1721
JACKSONVILLE 4, FLA., 1054 W. Adams
KANSAS CITY 6, MO., 408 Railway Exchange Bldg.
LOS ANGELES 23, CALIF., 1430 Grande Vista Avenue
MILWAUKEE 3, WIS., 1022 North 4th Street
NEW ORLEANS 16, LA., 809 Royal Street
NEW YORK 7, N.Y., 119 Barclay Street
PITTSBURGH 22, PA., 807 Bessemer Building
ST. LOUIS 17, 7248 Richmond Street
SALT LAKE CITY, UTAH, P.O. Box 836
SAN FRANCISCO 7, CALIF., 25 Stillman Street
SEATTLE 4, WASH., 1915 First Avenue, South
CANADA, Peacock Brothers, Ltd., P.O. Box 6070, Montreal
SOUTH AMERICA, Leopoldo Sol & Cia., Reconquista 558, Buenos Aires, Argentina

Thermometers

Princo Engineers' Type Thermometers for all Refrigerating Engineering purposes are built with special attention to the proper temperature gradients for the specific applications; with special attention to the proper selection and heat-treatment of Thermometric Glasses; and with special attention to the details of mechanical construction. Machining to fine limit gauges assures interchangeability of Separable Sockets of the same overall length. The mercury used is refined by a special process in our own laboratory. Selection of materials of mechanical construction is controlled by metallurgical and chemical research.



Princo Special Low Temperature Thermometers employing Toluol, Pentane and other liquids suitably adapted to Thermometry at very low temperatures, are made for plant applications in quick freezing processes, and for laboratory research work.

Standardized Thermometers, with National Bureau of Standard Certificates, are carried regularly in stock.

Maximum and Minimum Self Registering Thermometers.

Princo Test Thermometers and Test Wells.

Princo Thermometers for All Purposes.

High Precision Temperature Control

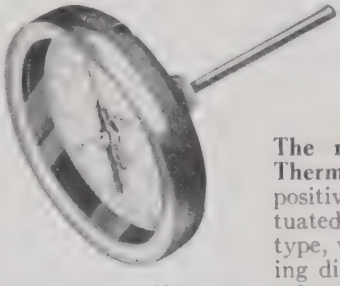
Princo Mercury-in-Glass Thermostats and Thermoregulators with Relays, for problems of accurate temperature control.

"Mercuthal MAGNA-SET," the Thermoregulator with the **micrometer** adjustment for temperatures down to minus 50°C..

Psychrometers

Princo Sling Psychrometers for Egg Storage Rooms are graduated in 0.1°F. divisions for the precise determinations of Egg Room Humidities. A.W.A. Relative Humidity Charts are supplied with each Instrument.

Dial Thermometers



The new Princo Dial Thermometers are of the positive, bimetallic-actuated, back-connected type, with 4″ easy-reading dial. Simplified dial and pointer adjustment and two types of "Easy-Access" cases (threaded frame or moisture-proof bolted frame) permit quick, easy zero-setting to any point from the front while the instrument is in service. Four stem lengths. Various types of separable sockets available. 5 scale ranges covering −100° to +850° F. or equivalent C.

Gauges

Princo Micrometer Draft Gauges.

Princo U Type Manometers, Water or Mercury, of special design, with demountable gauge glasses, for both high pressure differentials and low pressure applications.

Princo Barometric Type Mercurial Vacuum Gauges reading by vernier to 0.01 inch. The original gauge of this type.

Princo Mercurial Barometers of Fortin pattern.

Princo Special Tank Gauges. Factory Mutual approved type, for sprinkler tanks.

Hydrometers

Princo Hydrometers for Calcium Brine, Salt Brine, Aqua Ammonia.

Special and Experimental Instrument Making

THERMOMETERS (Continued)

Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Pyrometer Instrument Co., 103 Lafayette St., N.Y.C. 13
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Carl H. Wagner & Son, 1944 N. Albany Ave., Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

THERMOMETERS, ALARM

American Schaeffer & Budenberg Instrument Co., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave , Phila. 44, Pa.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Electric Auto-Lite Co., Toledo 1, O.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Charles O. Sherman Corp., 149 Broadway, N.Y.C. 6
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

THERMOMETERS, CONTROLLING (See CONTROLLERS, TEMPERATURE)

THERMOMETERS, DAIRY

American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
James Berg Mfg. Co., 3707-12th Ave., Brooklyn 18, N.Y.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
GM Mfg. Co., 50 W. 3rd St., N.Y.C. 12
General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti. 12, O.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16 Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Carl H. Wagner & Son, 1944 N. Albany Ave., Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

THERMOMETERS, DIAL

American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh 8, Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Electric Auto-Lite Co., Toledo 1, O.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
James P. Marsh Corp., 2073 Southport Ave., Chicago 14, Ill.
Marshalltown Mfg. Co., Marshalltown, Ia.
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti 12, O.
Partlow Corp., 2 Campion Rd., New Hartford, N.Y.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Pyrometer Instrument Co., 103 Lafayette St., N.Y.C. 13
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16. Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Carl H. Wagner & Son, 1944 N. Albany Ave. Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C 12
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

THERMOMETERS, DUCT

Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Johnson Service Co., 507 E. Michigan St., Milwaukee 22, Wis.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti. 12, O.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.J.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Carl H. Wagner & Son, 1944 N. Albany Ave., Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

THERMOMETERS, HOUSEHOLD REFRIGERATOR

Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
GM Mfg. Co., 50 W. 3rd St., N.Y.V. 12
General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.

C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Carl H. Wagner & Son, 1944 N. Albany Ave., Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

THERMOMETERS, INSERT

American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
James Berg Mfg. Co., 3707-12th Ave., Brooklyn 18, N.Y.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti. 12, O.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Pyrometer Instrument Co., 103 Lafayette St., N.Y.C. 13
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Carl H. Wagner & Son, 1944 N. Albany Ave., Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C 12

THERMOMETERS, MAXIMUM & MINIMUM REGISTERING

Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.
GM Mfg. Co., 50 W. 3rd St., N.Y.C. 12
General Scientific Equip. Co., 27th & Huntingdon St. Phila. 32, Pa.
Henry J. Green, 1191 Bedford Ave., Brooklyn 16, N.Y.
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
Carl H. Wagner & Son, 1944 N. Albany Ave., Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12
Weston Elec'l. Instrument Corp., 614 Frelinghuysen Ave., Newark 5, N.J.

THERMOMETERS, RECORDING

American Schaeffer & Budenberg Instrument Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Bacharach Industrial Instrument Co., 7000 Bennett St., Pittsburgh 8, Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Defender Instrument & Regulator Co., 815 Clark Ave., St. Louis 2, Mo.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Electric Auto-Lite Co., Toledo 1, O.
Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.
Gotham Instrument Co., Inc., 149 Wooster St., N.Y.C. 12
Henry J. Green, 1191 Bedford Ave., Brooklyn 16, N.Y.
Leeds & Northrup Co., 4970 Stenton Ave., Phila. 44, Pa.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Moore Products Co., H. & Lycoming Sts., Phila. 24, Pa.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti. 12, O.
Partlow Corp., 2 Campion Rd., New Hartford, N.Y.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Carl H. Wagner & Son, 1944 N. Albany Ave., Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

THERMOMETERS, WALL

GM Mfg. Co., 50 W. 3rd St., N.Y.C. 12
General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 134 Livingston St., Brooklyn 2, N.Y.
Palmer Thermometers, Inc., 2501 Norwood Ave., Cin'ti. 12, O.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Pyrometer Instrument Co., 103 Lafayette St., N.Y.C. 13
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Carl H. Wagner & Son, 1944 N. Albany Ave., Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

THERMOMETERS, WINDOW

Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
General Scientific Equip. Co., 27th & Huntingdon St., Phila. 32, Pa.
Moeller Instrument Co., Inc., 132nd St., & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. O. Trerice Co., 1420 W. Lafayette Blvd., Detroit 16, Mich.
U. S. Gauge, Div. of American Machine & Metals, Sellersville, Pa.
Carl H. Wagner & Son, 1944 N. Albany Ave., Chicago 47, Ill.
Weksler Thermometer Corp., 52 W. Houston St., N.Y.C. 12

THERMOREGULATORS

Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Carroll Glass Instrument Co., 6742 Lebanon Ave., Phila. 31, Pa.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Moeller Instrument Co., Inc., 132nd St. & 89th Ave., Richmond Hill 18, N.Y.
Nurnberg Thermometer Co., Inc., 124 Livingston St., Brooklyn 2, N.Y.

Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*

THERMOSTATIC BI-METALS

Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
W. M. Chace Co., 1601 Beard Ave., Detroit 9, Mich.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
General Plate Div., Metals & Controls Corp., Attleboro, Mass.
Red Spot Elec. Co., Tacoma, Wash.

THERMOSTATIC EXPANSION VALVES (See EXPANSION VALVES)

THERMOSTATS (See also CONTROLS, etc.)

Allen-Bradley Co., Milwaukee 4, Wis.
Barber-Colman Co., Rockford, Ill.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Burling Instrument Co., 253 Springfield Ave., Newark 3, N.J.
Detroit Lubricator Co., 5900 Trumbull Ave., Detroit 8, Mich. *(p. 100)*
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Thomas A. Edison, Inc., W. Orange, N.J.
Friez Instrument Div., Bendix Aviation Corp., Taylor Ave. at Loch Raven Blvd., Towson, Baltimore 4, Md.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
General Controls Co., 801 Allen Ave., Glendale 1, Cal. *(p. 199)*
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Hart Mfg. Co., 110 Bartholomew Ave., Hartford 1, Ct.
Johnson Service Co., 507 E. Michigan St., Milwaukee 2, Wis.
Mercoid Corp., 4201 Belmont Ave., Chicago 41, Ill. *(p. 74)*
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Penn Elec. Switch Co., Goshen, Ind. *(p. 73)*
Perfex Corp., 500 W. Oklahoma Ave., Milwaukee 7, Wis.
Precision Thermometer & Instrument Co., 1442 Brandywine St., Phila. 30, Pa. *(p. 209)*
Ranco, Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 75)*
Sampsel Time Control, Inc., 600 N. Strong Ave., Spring Valley, Ill.
Spencer Thermostat Co., Unit of Metals & Controls Corp., 34 Forest St., Attleboro, Mass.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
H. A. Thrush & Co., 21 E. Riverside Dr., Peru, Ind.
United Elec. Controls Co., 71 A St., S. Boston 27, Mass.
Wesix Elec. Heater Co., 390-1st St., San Francisco 5, Cal.
Westinghouse Elec. Corp., Meadville, Pa.
White-Rodgers Elec. Co., 1209 Cass Ave., St. Louis 6, Mo.

THERMOSTATS, HOUSEHOLD

Cutler-Hammer, Inc., 315 N. 12th St., Milwaukee 1, Wis.
Ranco, Inc., 601 W. 5th Ave., Columbus 1, O. *(p. 75)*

THREADED SPECIALTIES (See also SCREW MACHINE PARTS)

Eastern Machine Screw Corp., Truman & Barclay Sts., New Haven 6, Ct.
Ohio Nut & Bolt Co., 600 Front St., Berea, O.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Watson Flagg Machine Co., 845 E. 25th St., Paterson 3, N.J.

TIME SWITCHES

Automatic Control Co., 1005 University Ave., St. Paul 4, Minn.
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
R. W. Cramer Co., Centerbrook, Ct.
Thomas A. Edison, Inc., W. Orange, N.J.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Holcomb & Hoke Mfg. Co., Inc., 1545 Van Buren St., Indpls. 7, Ind.
International Register Co., 2620 W. Washington Blvd., Chicago 12, Ill.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
National Time & Signal Corp., 21800 Wyoming Ave., Detroit 20, Mich.
Paragon Elec. Co., 1600-12th St., Two Rivers, Wis.
Perfex Corp., 500 W. Oklahoma Ave., Milwaukee 7, Wis.
Photoswitch, Inc., 77 Broadway, Cambridge 42, Mass.
Potter & Brumfield Sales Co., 549 W. Washington Blvd., Chicago 6, Ill.
Reliance Automatic Lighting Co., 1927 Mead St., Racine, Wis.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
M. H. Rhodes, Inc., 20 Barholomew, Hartford, Ct.
Sampsel Time Control, Inc., 600 N. Strong Ave., Spring Valley, Ill.
Struthers-Dunn, Inc., 150 N. 13th St., Phila. 7, Pa.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Tork Clock Co., Inc., 1 Grove St., Mt. Vernon, N.Y.
Wesix Elec. Heater Co., 390-1st St., San Francisco 5, Cal.

TIN

Alpha Metals, Inc., 363 Hudson Ave., Brooklyn 1, N.Y.
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Galv-Weld Products, 324 E. 2nd St., Dayton 2, O.
National Lead Co., 111 Broadway, N.Y.C. 6

TIN PLATE

Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Bethlehem Steel Co., Bethlehem, Pa.
Carnegie-Illinois Steel Corp., U. S. Steel Corp. Subsidiary, Carnegie Bldg., Pittsburgh 30, Pa.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Weirton Steel Co., Weirton, W. Va.
Youngstown Sheet & Tube Co., Youngstown, O.

TRACERS (See LEAK INDICATORS)

TRACK, SLIDING DOOR

American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Knape & Vogt Mfg. Co., Grand Rapids 4, Mich.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

TRACKAGE, OVERHEAD FOR CRANES

American Monorail Co., 13107 Athens Ave., Cleveland 7, O.
Union Mfg. Co., New Britain, Ct.

TRANSFORMERS (SMALL)

American Transformer Co., 172 Emmet St., Newark 5, N.J.
Barber-Colman Co., Rockford, Ill.
Dean W. Davis & Co., Kentland, Ind.
Eisler Engrg. Co., Inc., 740 S. 13th St., Newark 3, N.J.
Forbes & Myers, 173 Union St., Worcester, Mass.
General Elec. Co., 1 River Rd., Schenectady 5, N.J.
Jefferson Elec. Co., 25 Ave. & Madison St., Bellwood, Ill.

TRAPS, REFRIGERATION

Armstrong Machine Wks., 846 Maple St., Three Rivers, Mich.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.

Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

TRAPS, REFRIGERATOR DRAIN
Kason Hardware Corp., 127 Wallabout St., Brooklyn 6, N.Y.

TRAPS, STEAM (See STEAM TRAPS)

TRAPS, WATER, FOR AIR LINES
Armstrong Machine Wks., 846 Maple St., Three Rivers, Mich.
Cryer Trap & Valve Co., 366 Madison Ave., N.Y.C.
Johnson Corp., 805 Wood St., Three Rivers, Mich.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Wright-Austin Co., 315 W. Woodbridge St., Detroit 26, Mich.

TRAYS, DEFROSTING
Mack Molding Co., Ryerson Ave., Wayne, N.J.
Sneath Glass Co., Hartford City, Ind.

TRAYS, ICE CUBE (See ICE CUBE TRAYS)

TRAYS, REFRIGERATOR
Gustav Glaser Co., Inc., 2 Wait St., Paterson 4, N.J
McCray Refrigerator Co., Kendallville, Ind.

TRAYS, WIRE (See also SHELVES, etc.)
Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
Standard Valve Mfg. Co., 817 Albany St., Boston 19, Mass.
United Steel & Wire Co., Battle Creek, Mich.
Wall Wire Products Co., 11333 General Dr., Plymouth, Mich.
L. A. Young Spring & Wire Corp., 9200 Russell St., Detroit 11, Mich.

TREATMENT, BRINE, WATER, etc. (See BRINE TREATMENT; also WATER TREATMENT; also INHIBITORS)

TRIM, BREAKER (See BREAKER STRIPS)

TRIM, CABINET (See also NAME PLATES)
American Emblem Co., Inc., 9 Genesee St., New Hartford, Utica 1, N.Y.
Brasco Mfg. Co., Harvey, Ill.
Croname, Inc., 3701 N. Ravenswood, Chicago 13, Ill.
Dahlstrom Metallic Door Co., 435 Buffalo St., Jamestown, N.Y.
Dearborn Glass Co., 2414 W. 21st St., Chicago 8, Ill.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Grand Rapids Brass Co., 60 Scribner Ave., N.W., Grand Rapids 1, Mich. *(p. 126)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Stanley Wks., 195 Lake St., New Britain, Ct.
U. S. Gypsum Co , 300 W. Adams St., Chicago 6, Ill.
R. D. Werner Co., Inc., 295-5th Ave., N.Y.C. 16

TRIM, SHELF
Brasco Mfg. Co., Harvey, Ill.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
R. D. Werner Co., Inc., 295-5th Ave., N.Y.C. 16

TROLLEYS, OVERHEAD CONVEYOR (See CONVEYORS, MONORAIL)

TRUCK & TRAILER BODIES, REFRIGERATED
Airtex Corp., 333 N. Michigan Ave., Chicago 1, Ill.
American Body & Equip. Co., Div. of Hyde Corp., P.O. Box 3107, Dallas 1, Tex.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Broquinda, Inc. of Florida, St. Petersburg, Fla.
Fortney Body Co., 4413 Train Ave., Cleveland 13, O.
Fruehauf Trailer Co., 10940 Harper Ave., Detroit 32, Mich.
Hackney Bros. Body Co., Wilson, N.C.
Highway Trailer Co., Edgerton, Wis.
Robbins & Burke, Inc., 29 Lansdowne, Cambridge, Mass.
Stokes Co., 131st & Morgan Sts., Blue Island, Ill.

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More detailed information will be sent you on request to:

THE AMERICAN SOCIETY OF REFRIGERATING ENGINEERS
40 West 40 Street
New York 18, N.Y.

TRUCK & TRAILER REFRIGERATION SYSTEMS

Advance Mfg. Co., Inc., 2700 Buchanan St., Detroit 8, Mich.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Broquinda, Inc. of Florida, St. Petersburg, Fla.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Cold Control, Inc., 111 Broadway, N.Y.C. 6
Dorsey Trailers, Elba, Ala.
Frick Co., Waynesboro, Pa. *(p. 47)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Fruehauf Trailer Co., 10940 Harper Ave., Detroit 32, Mich.
Industrial Mfg. & Engrg. Co., 3845 N. Ravenswood Ave., Chicago 13, Ill.
Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 215)*
Propane Development Corp., 41 Murray St., N.Y.C. 7
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Sterling Mfg. Co., 2523 Farnam, Omaha, Neb.
Stokes Co., 131st & Morgan Sts., Blue Island, Ill.
U. S. Thermo Control Co., 44 S. 12th St., Minneapolis 4, Minn.

TRUCK CASTERS

Fairbanks Co., 393 Lafayette, N.Y.C. 3

TRUCK PLATES

Dole Refrigerating Co., 5910 N. Pulaski Rd., Chicago 30, Ill. *(p. 167)*
Frosty Foods Equip. Co., 305 Benson Blvd., Sioux City 15, Ia.
Kold-Hold Mfg. Co., 603 E. Hazel St., Lansing 4, Mich. *(p. 215)*
Stangard Div., Noma Elec. Corp., 46 Oliver St., Newark, N.J. *(p. 166)*

TRUCKS, MATERIAL HANDLING

Fairbanks Co., 393 Lafayette, N.Y.C. 3
Falstrom Co., 13 Falstrom Court, Passaic, N.J.
W. J. Finnegan Co., 7402 Santa Monica Blvd., Los Angeles 46, Cal. *(p. 116)*
Gustav Glaser Co., Inc., 2 Wait St., Paterson 4, N.J.
Locker Engrg. Co., 521 N. La Cienga Blvd., Los Angeles 36, Cal.
Midwest Metal Stamping Co., Kellogg, Ia.

TUBE BENDING (See also PIPE BENDS & BENDING)

American Brass Co., Waterbury 88, Ct.
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
C. F. Moores Co., Inc., Ivy Hill Rd., Wyndmoor, Phila. 18, Pa.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Standard Valve Mfg. Co., 817 Albany St., Boston 19, Mass.
Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
Wallace Tube Co., Subsidiary of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.

TUBE ICE MACHINERY (See ICE MAKING MACHINES)

TUBES (See particular type; also TUBING)

TUBES, ALLOY

American Brass Co., Waterbury 88, Ct.
Babcock & Wilcox Co., 85 Liberty St. N.Y.C. 6
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
National Tube Co., U. S. Steel Corp., Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
epublic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Tube Co., 20 Germantown Ave., Norristown, Pa.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Wallace Tube Co., Subsidiary of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.

TUBES, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
American Brass Co., Waterbury 88, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Colonial Alloys Co., Ridge Ave. & Crawford St., Phila. 29, Pa.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

TUBES, BOILER & CONDENSER

American Brass Co., Waterbury 88, Ct.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Tube Co., U. S. Steel Corp. Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Tube Co., 20 Germantown Ave., Norristown, Pa.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Wallace Tube Co., Subsidiary of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Cihcago 14, Ill.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1411 Central Ave., Detroit 9, Mich.

TUBES, CAPILLARY, ASSEMBLIES (See also TUBING, CAPILLARY)

Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill.

TUBES, FIBRE (See FIBRE)

TUBES, STEEL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Bethlehem Steel Co., Bethlehem, Pa.
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. *(p. 219)*
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Jones & Laughlin Steel Co., Pittsburgh, Pa.
Michigan Seamless Tube Co., S. Lyon, Mich.
Michigan Steel Tube Products Co., 9450 Buffalo St., Detroit 12, Mich.

(Continued)

KOLD-HOLD LOW SIDES HAVE APPLICATIONS *Unlimited*

"HOLD-OVER" TRUCK PLATES

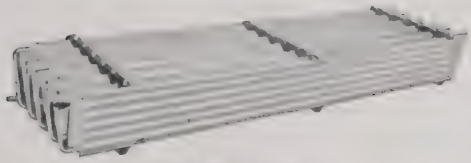
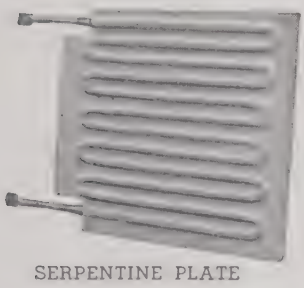
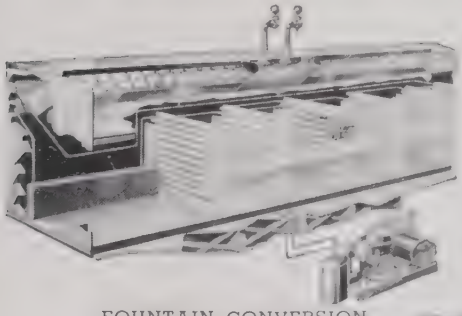


PLATE BANK



SERPENTINE PLATE



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For locker plant space cooling, for shelves and stands in sharp freezing, or as cabinet liners, Kold-Hold Quick-Action Serpentine Plates, either wall mounted or in ceiling banks, have no equal in efficiency and dependability. In truck refrigeration, Kold - Hold streamlined "Hold-Over" Plates maintain the temperature of delivery truck bodies at the uniform level necessary in the successful transportation of fresh meat, ice cream and frozen foods. New Kold-Hold "Pakaged" Truck Unit is the answer to many needs — no installation necessary. The Ice Bank is a new "accumulator" of cold for peak load air conditioning. Write today for complete data and engineering assistance.

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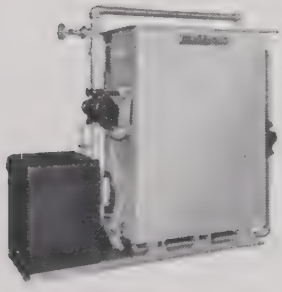
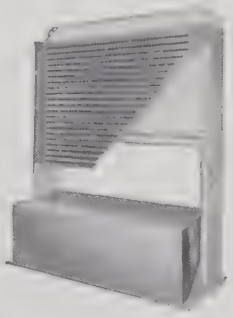


ICE BANK

PLATE STAND



"PAKAGED" TRUCK UNIT

TUBES, STEEL (Continued)

A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Tube Co., U. S. Steel Corp. Subsidiary, P. O. Box 266, Pittsburgh, Pa.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Tube Co., 20 Germantown Ave., Norristown, Pa.
Wallace Tube Co., Subsidiary of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.

TUBES, STAINLESS STEEL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Tube Co., U. S. Steel Corp. Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Pittsburgh Pipe Coil & Bending Co., 61 Bridge St., Etna P.O., Pittsburgh 23, Pa.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Tube Co., 20 Germantown Ave., Norristown, Pa.
Wallace Tube Co., Subsidiary of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.

TUBING, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.

American Brass Co., Waterbury 88, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Colonial Alloys Co., Ridge Ave. & Crawford St., Phila. 29, Pa.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Linderme Tube Co., 1500 E. 219th St., Cleveland 17, O.
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

TUBING, BEER

Aeroquip Corp., 300 S. East Ave., Jackson, Mich. *(p. 83)*
American Hard Rubber Co., 11 Mercer St., N.Y.C. 13
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. *(p. 219)*
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hudson Products Co., Inc., 4400 St. Aubin, Detroit 7, Mich.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17

TUBING, BLOCK TIN

Alpha Metals, Inc., 363 Hudson Ave., Brooklyn 1, N.Y
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
National Lead Co., 111 Broadway, N.Y.C. 6

TUBING, BRASS & BRONZE

American Brass Co., Waterbury 88, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Lewin Metals Div., Lewin-Mathes Co., 1111 Chouteau, St. Louis, Mo.
Linderme Tube Co., 1500 E. 219th St., Cleveland 17, O.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
Penn Brass & Copper Co., Powell Ave., Erie, Pa. *(p. 216)*
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
Shenango-Penn Mold Co., Dover, O.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co 1411 Central Ave., Detroit 9, Mich.

TUBING, CAPILLARY

American Brass Co., Waterbury 88, Ct.
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. *(p. 219)*
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
Penn Brass & Copper Co., Powell Ave., Erie, Pa. *(p. 216)*
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Superior Tube Co., 20 Germantown Ave., Norristown, Pa.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Wabash Mfg. Co., 2642 S. Michigan Ave., Chicago 16, Ill.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1411 Central Ave., Detroit 9, Mich.

TUBING, COPPER

American Brass Co., Waterbury 88, Ct.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91 Ct.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill

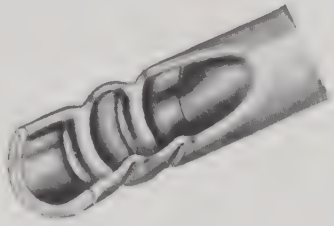
(Continued)

REVERE DRYSEAL COPPER REFRIGERATOR TUBE

NEW Package

Revere Dryseal Tube now comes in 50-foot double layer flat coils, packed two to an individual carton. These new red and blue cartons protect the tube, keep it clean, may be conveniently packed for storage, and are easier to unpack.

NEW Seal



Revere Dryseal Tube now has a double-groove mechanical seal. It is compact enough to pass through any opening large enough for the tube itself. It permanently keeps the interior of the tube clean and bone dry.

NEW Standards

Revere Dryseal Tube now comes in new, more economical dimensional standards as follows:

Size O.D. in inches	Wall Gauge in inches	Pounds per Linear Foot	Approx. Weight per 50-ft. Coil
1/8	.030	.0347	1.74
3/16	.030	.0575	2.88
1/4	.030	.0804	4.02
5/16	.032	.109	5.45
3/8	.032	.134	6.70
1/2	.032	.182	9.10
5/8	.035	.251	12.55
3/4	.035	.305	15.25

Revere Dryseal is the Standard of Quality

Revere Dryseal Tube is dead soft, so that you can bend it easily . . . and the ends won't split when flared. Exterior and interior surfaces are clean, smooth, and free from injurious defects. It is made of deoxidized copper (99.9% pure) and is kept oxide-free by special processing methods.

Revere Dryseal Copper Refrigeration Tube and Revere Aluminum Refrigeration Tube are available from leading distributors throughout the country.

REVERE COPPER AND BRASS INCORPORATED

Founded by Paul Revere in 1801

230 Park Avenue, New York 17, New York

Mills: Baltimore, Md.; Chicago, Ill.; Detroit, Mich.; New Bedford, Mass.; Rome, N.Y.—
Sales Offices in Principal Cities, Distributors Everywhere

TUBING, COPPER (Continued)

Lewin Metals Div., Lewin-Mathes Co., 1111 Chouteau, St. Louis, Mo.
Linderme Tube Co., 1500 E. 219th St., Cleveland 17, O.
Mueller Brass Co., Port Huron, Mich.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
Penn Brass & Copper Co., Powell Ave., Erie, Pa. *(p. 216)*
Reading Tube Corp., Empire State Bldg., N.Y.C. 1
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1411 Central Ave., Detroit 9, Mich.

TUBING, FINNED (See FINNED TUBING)

TUBING, FLEXIBLE METAL (See also CHARGING LINES; also FLEXIBLE CONNECTIONS)

Aeroquip Corp., 300 S. East Ave., Jackson, Mich. *(p. 83)*
American Brass Co., Waterbury 88, Ct.
Atlantic Metal Hose Co., Inc., 123 W. 64th St., N.Y.C. 23
Brockway Co., 361 Church St., Naugatuck, Ct.
Chicago Metal Hose Corp., Maywood, Ill.
Eclipse Aviation Metal Hose Dept., Div. of Bendix Aviation Corp., Phila. 44, Pa.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Packless Metal Products Corp., 31 Winthrop Ave., New Rochelle, N.Y.
Seamlex Co., Inc., 4123-24th St., Long Island City 1, N.Y. *(p. 218)*
Titeflex, Inc., 500 Frelinghuysen Ave., Newark 5, N.J.
Zallea Bros. & Johnson, 9th St. & P.R.R., Wilmington 99, Del.

TUBING, INCONEL, MONEL, NICKEL, etc.

Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. *(p. 219)*
Carpenter Steel Co., Reading, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Driver-Harris Co., Harrison, N.J.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C. 5
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Tube Co., U. S. Steel Corp., Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Shenango-Penn Mold Co., Dover, O.
Superior Tube Co., 20 Germantown Ave., Norristown, Pa.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

SEAMLEX
FLEXIBLE SEAMLESS METAL HOSE
ANY SIZE—ANY PURPOSE
Ask for Quotation
SEAMLEX CO., INC.
4123 24TH ST., LONG ISLAND CITY 1, N.Y.
—ESTABLISHED 1928—

TUBING, LEAD

Alpha Metals, Inc., 363 Hudson Ave., Brooklyn 1, N.Y.
Bridgeport Thermostat Co., Inc., 1225 Connecticut Ave., Bridgeport 1, Ct.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
National Lead Co., 111 Broadway, N.Y.C. 6

TUBING, LIGHT WALL

American Brass Co., Waterbury 88, Ct.
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. *(p. 219)*
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
Michigan Seamless Tube Co., S. Lyon, Mich.
Michigan Steel Tube Products Co., 9450 Buffalo St., Detroit 12, Mich.
National Copper & Smelting Co., 1862 E. 213rd St., Cleveland 6, O.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.

TUBING, LOCK SEAM

Brasco Mfg. Co., Harvey, Ill.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.

TUBING, MECHANICAL

Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Michigan Seamless Tube Co., S. Lyon, Mich.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Tube Co., U. S. Steel Corp. Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Pittsburgh Tube Co., 323-4th Ave., Pittsburgh 22, Pa.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Superior Tube Co., 20 Germantown Ave., Norristown, Pa.
Wallace Tube Co., Subsidiary of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.

TUBING, PREFABRICATED

Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. *(p. 219)*
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Kenmore Machine Products, Inc., 15 Depew Ave., Lyons, N.Y.
Joseph E. Lewis & Co., Inc., 1315 Carroll St., Baltimore 30, Md.
Mueller Brass Co., Port Huron, Mich.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Copper & Smelting Co., 1862 E. 123rd St., Cleveland 6, O.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Wallace Tube Co., Subsidiary of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1411 Central Ave., Detroit 9, Mich.

TUBING, RUBBER

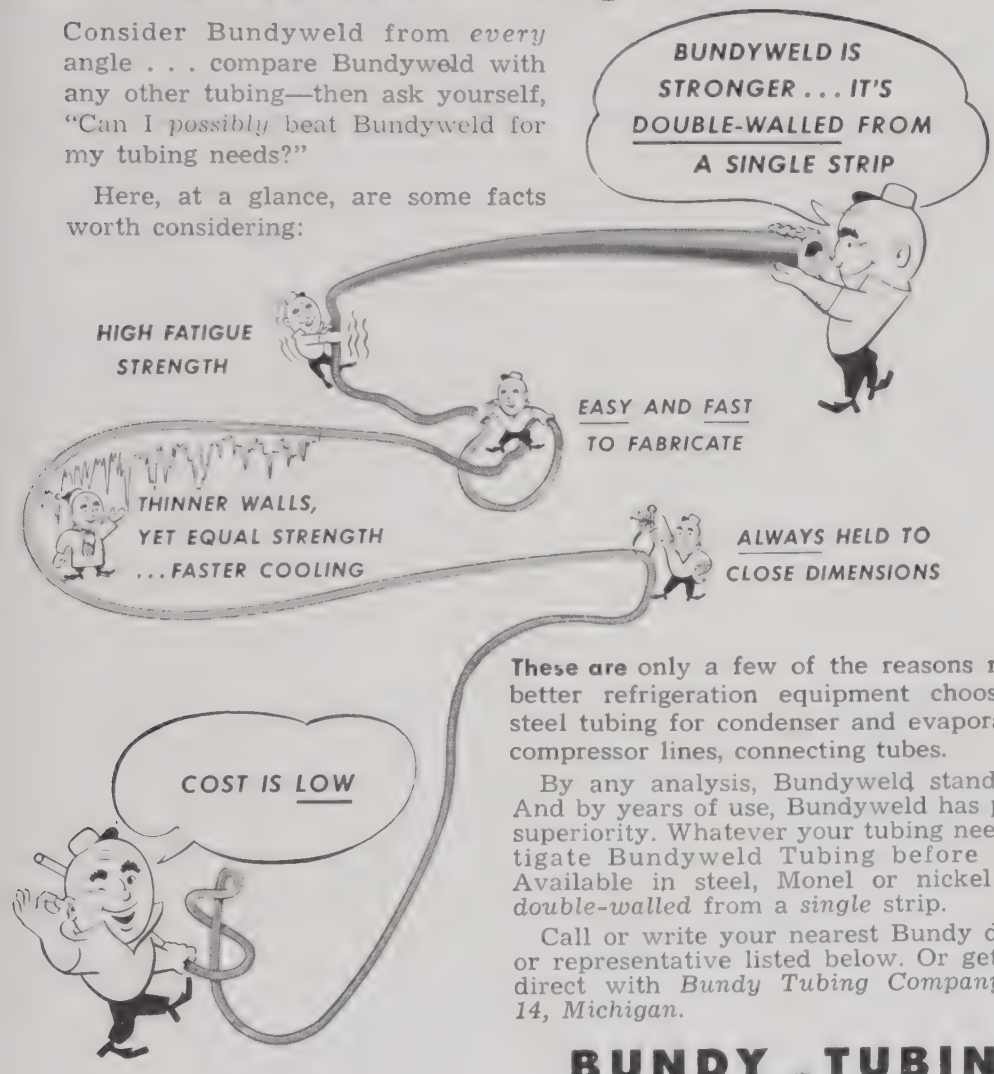
American Hard Rubber Co., 11 Mercer St., N.Y.C. 13 (Hard Rubber)
Buffalo Weaving & Belting Co., 262 Chandler St., Buffalo 7, N.Y.
Continental Rubber Wks., 2000 Liberty St., Erie, Pa.
Garlock Packing Co., 402 E. Main St., Palmyra, N.Y.
General Scientific Equip. Co., 27th & Huntingdon Sts., Phila. 32, Pa.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.

(Continued)

How can you possibly beat Bundyweld* for refrigeration tubing?

Consider Bundyweld from *every* angle . . . compare Bundyweld with any other tubing—then ask yourself, "Can I *possibly* beat Bundyweld for my tubing needs?"

Here, at a glance, are some facts worth considering:



These are only a few of the reasons makers of better refrigeration equipment choose Bundy steel tubing for condenser and evaporator coils, compressor lines, connecting tubes.

By any analysis, Bundyweld stands up top. And by years of use, Bundyweld has proved its superiority. Whatever your tubing needs, investigate Bundyweld Tubing before you buy. Available in steel, Monel or nickel . . . all *double-walled* from a *single* strip.

Call or write your nearest Bundy distributor or representative listed below. Or get in touch direct with *Bundy Tubing Company, Detroit 14, Michigan.*

BUNDY TUBING

ENGINEERED TO YOUR EXPECTATIONS

*REG. U. S. PAT. OFF. Ⓡ

WHY BUNDYWELD IS BETTER TUBING

1. Bundyweld Tubing is made by a patented process entirely different from that used in any other tubing. It starts as a single strip of basic metal, coated with a bonding metal.

2. This strip is continuously rolled twice laterally into tubular form. Walls of uniform thickness and concentricity are assured by use of close-tolerance, cold rolled strip.

3. Next, this double-rolled strip passes through a heating process, where the bonding metal fuses with the basic metal. When cooled, it becomes a solid tube free from scale, held to close dimensions.

4. Bundy Tubing comes in standard sizes, up to 5/8" O.D. Special sizes are also available. Available in steel, Monel or nickel. For tubing of other metals, call or write Bundy.

BUNDY TUBING DISTRIBUTORS AND REPRESENTATIVES:

Pacific Metals Co., Ltd., 3100 19th St., San Francisco 10, Calif.
Standard Tube Sales Corp., 76-01 Woodhaven Blvd., Brooklyn 27, N.Y.
Lapham-Hickey Co., 3333 W. 47th Place, Chicago 32, Ill.
Rutan & Co., 404 Architects Bldg., Philadelphia 3, Pa.
Eagle Metals Co., 3628 E. Marginal Way, Seattle 4, Wash.
Alloy Metal Sales, Ltd., 861 Bay St., Toronto 5, Canada
Peirson-Deakins Co., 823-824 Chattanooga Bank Bldg., Chattanooga 2, Tenn.

Bundyweld nickel and Monel tubing is sold by International Nickel Company distributors in all principal cities

TUBING RUBBER (Continued)

Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
John M. Watt's Sons, 112 Walnut St., Phila., Pa.

TUBING, SPONGE RUBBER

B. F. Goodrich Co., 500 S. Main St., Akron, O.
Jarrow Products, 420 N. La Salle St., Chidago 10, Ill. *(p. 89)*
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Sponge Rubber Products Co., 106 Derby Place, Shelton, Ct. *(p. 188)*
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

TUBING, STEEL

Babcock & Wilcox Co., 85 Liberty Sts., N.Y.C. 6
Bethlehem Steel Co., Bethlehem, Pa.
Bundy Tubing Co., E. Jefferson & Parker Ave., Detroit 4, Mich. *(p. 221)*
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Everhot Products Co., 2001 Carroll Ave., Chicago 12, Ill.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
Jones & Laughlin Steel Co., Pittsburgh, Pa.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
Michigan Seamless Tube Co., S. Lyon, Mich.
Michigan Steel Tube Products Co., 9450 Buffalo St., Detroit 12, Mich.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Tube Co., U. S. Steel Corp. Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Tube Co., 20 Germantown Ave., Norristown, Pa.
Wallace Tube Co., Subsidiary of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.

TUBING, STEEL, STAINLESS

Babcock & Wilcox Co., 85 Liberty St., N.Y.C. 6
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Globe Steel Tubes Co., 3839 W. Burnham, Milwaukee 4, Wis.
A. B. Murray Co., Inc., 604 Green Lane, Elizabeth, N.J.
National Tube Co., U. S. Steel Corp. Subsidiary, P.O. Box 266, Pittsburgh, Pa.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Superior Tube Co., 20 Germantown Ave., Norristown, Pa.
Wallace Tube Co., Subsidiary of Wallace Supplies Mfg. Co., 1300 Diversey Pkwy., Chicago 14, Ill.

TUBULAR PARTS

Fitzsimons Mfg. Co., 3775 E. Outer Dr., Detroit 12, Mich.
Michigan Steel Tube Products Co., 9450 Buffalo St., Detroit 12, Mich.
Mueller Brass Co., Port Huron, Mich.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Republic Steel Corp., Steel & Tubes Div., 224 E. 131st St., Cleveland, O.
Wolverine Tube Div., Calumet & Hecla Cons. Copper Co., 1411 Central Ave., Detroit 9, Mich.

TUNNEL FREEZERS (See also BLAST FREEZERS; also QUICK FREEZERS, etc.)

Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.

TURBINES (See STEAM TURBINES)

TURBO COMPRESSORS

Allis-Chalmers Mfg. Co., Milwaukee 1, Wis.
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Ingersoll-Rand Co., 11 Broadway, N.Y.C. 4
Roots-Connersville Blower Corp., P.O. Box 327, Connersville, Ind.
Spencer Turbine Co., Hartford 6, Ct.
Trane Co., La Crosse, Wis. *(p. 14)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

TURNBUCKLES

Bethlehem Steel Co., Bethlehem, Pa.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
U. S. Expansion Bolt Co., York, Pa.

TURNINGS (See SCREW MACHINE PARTS; also THREADED PARTS)

TWO TEMPERATURE VALVES, SNAP ACTION (See also SUCTION PRESSURE REGULATING VALVES)

Alco Valve Co., 855 Kingsland Ave., St. Louis 5, Mo. *(p. 98)*
Aminco Refrigeration Products Co., 14544-3rd Ave., Detroit 3, Mich. *(p. 159)*
Automatic Products Co., 2450 N. 32nd St., Milwaukee 10, Wis. *(p. 99)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 205)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.

ULTRA-VIOLET RADIATION (See also LAMPS, BACTERICIDAL)

Hanovia Chemical & Mfg. Co., Chestnut St. & N.J.R.R. Ave., Newark 5, N.J.
Payton Co., 546 W. Washington Blvd., Chicago 6, Ill.
Reynolds Elec. Co., 2650 W. Congress, Chicago 12, Ill.
Sperti, Inc., Norwood Sta., Cin'ti. 12, O.
Sun Ray Pasteurizer Industries, Inc., 119 W. Voorhees St., Cin'ti. 15, O.
Ultra-Violet Products, Inc., 5205 Santa Monica Blvd., Los Angeles 27, Cal

UNDERCOATERS (See COATINGS & COMPOUNDS; also FINISHES)

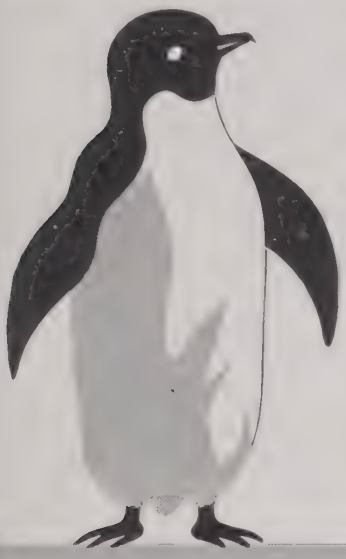
UNIONS, PIPE (See also FITTINGS, PIPE)

Cofley Co., Inc., 629 Grove St., Jersey City 2, N.J.
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Jefferson Union Co., Inc., 71 Gooding St., Lockport, N.Y.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)

UNIT COOLERS ABOVE 32°—CEILING MOUNTED
(A—Ammonia; B—Other refrigerants)

(A,B) Advanced Engrg. Co., 2646 W. Fond du Lac Ave., Milwaukee, Wis.
(B) American Coils Co., 25 Lexington St., Newark 5, N.J.
(A,B) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
(B) Betz Corp., 445 State St., Hammond, Ind.
(A,B) Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
(A,B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(A,B) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(B) Fedders-Quigan Corp., 57 Tonawanda St., Buffalo 7, N.Y.

(Continued)



FOR ECONOMY, DEPENDABILITY, EASE OF INSTALLATION . . . YOU CAN'T BEAT

BUSH HEAT TRANSFER PRODUCTS

The units illustrated here are only a few of the products that have won for Bush an outstanding reputation among users of heat transfer products. We shall be glad to furnish complete data and specifications on these and other Bush equipments.

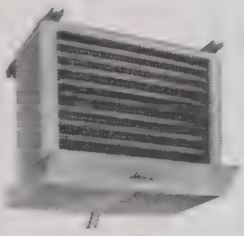
These include:

- Standard Unit Coolers
- Wall-Mounted Panel Coolers
- Plasti-Coolers
- Condensers
- Evaporators
- Evaporative Condensers
- Ice Makers
- Steam Coils
- Water Coils
- Ceiling Wall-Mounted Coolers
- Floor-and Suspended-Type Air-Conditioning Units

Sold by leading refrigeration wholesalers throughout the nation

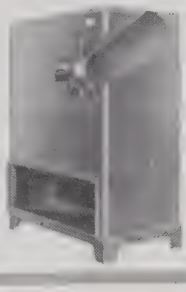
WATER DEFROST

A low-temperature unit designed to defrost in five minutes with tap water. Has no complicated parts and can be installed easily by any serviceman. Capacities range from 4500 to 18000 BTU at 10°T. D. and from 6750 to 27000 BTU at 15°T. D



FLOOR-TYPE UNIT

Designed to meet general requirements of both high and low temperature unit coolers where large capacities are required. Sturdily built, ample coil surface, easily assembled, attractive appearance

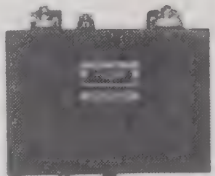


ROUND SUPREME

Designed for coolers with low headroom. Air is discharged away from the products, making it possible to maintain relatively high humidity. Bottom pan easily removed for access to evaporator.

AUTO DEFROST

A fully automatic, electronically controlled circuit that relieves you entirely of defrosting supervision. It defrosts only when necessary . . . can handle several coils by means of simple relay circuits. Easy to install



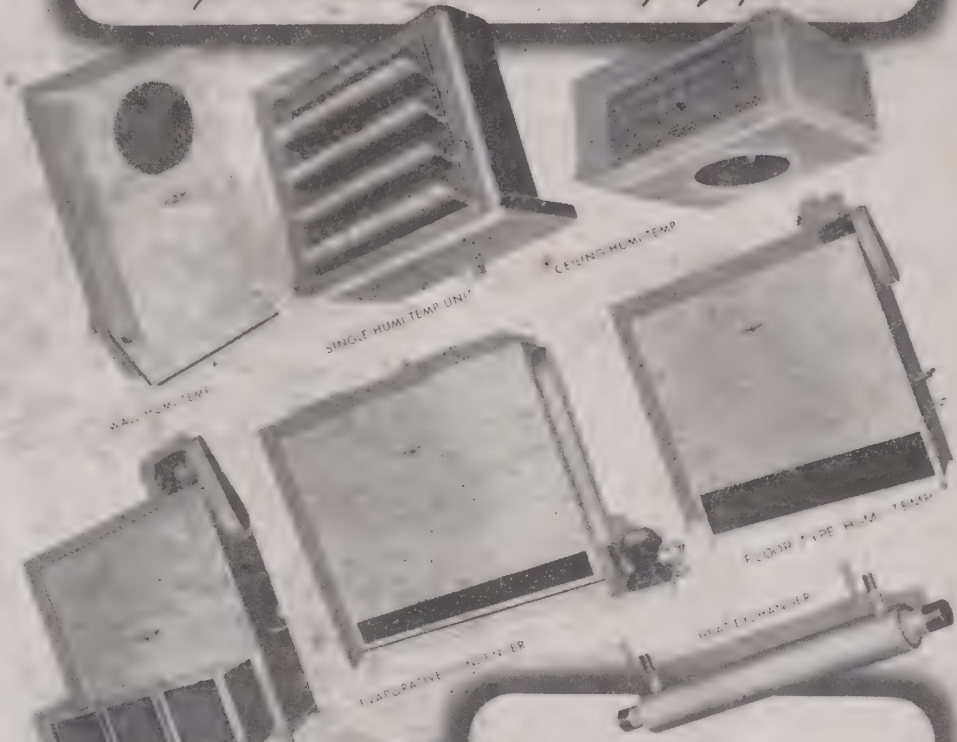
BUSH MANUFACTURING COMPANY

HARTFORD, CONN.

OUR FORTIETH YEAR

415 LEXINGTON AVE., NEW YORK • 549 W. WASHINGTON BOULEVARD, CHICAGO • EXPORT ADDRESS: 13 EAST 40th ST., NEW YORK • CABLE "ARLAB"

LARKIN . . . the leading manufacturer of complete commercial and industrial refrigeration and air conditioning equipment

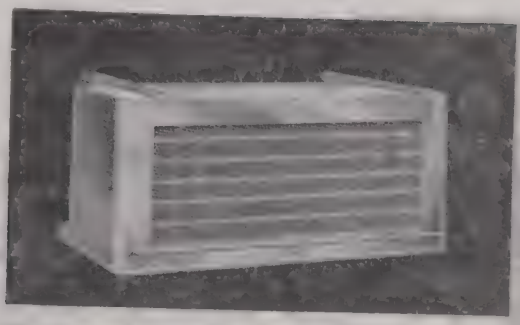


LONG and productive experience, keeping step with advanced engineering skill and conscience has made Larkin products recognized leaders in industrial refrigeration and air conditioning. Our manufacturing processes have explored the entire range of possibilities for improvement—and our present roster of products covers every phase of this growingly important field. Origination and patenting of the cross-fin coil is just one example of technical enterprise that has established the Larkin line. Bright and imaginative finishes on what might otherwise be drab and purely utilitarian equipment is another advance which illustrates our completeness in design and production For variety . . . versatility . . . and performance . . look to Larkin

LARKIN

519 Memorial Drive, S.E.
ATLANTA, GA.

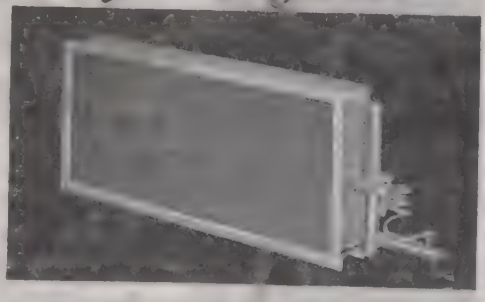
THE WATCHDOG OF THE NATION'S FOOD SUPPLY



"Recold" Water Defrost Coil For Temperatures Below 34°.

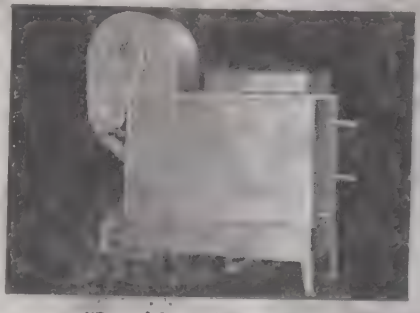
"RECOLD" PROVIDES DEPENDABLE REFRIGERATION EQUIPMENT *for every need*

Behind the "Recold" trade mark there is more than sixteen years of progressive growth and engineering. "Recold" is a pioneer in the development of blower coils in the West, and is the originator of Water Defrost Coils. Today "Recold" manufactures a complete line of refrigeration and air conditioning equipment to meet every commercial application.



"Recold" Air Conditioning Coil Designed for every application.

Again and again, "Recold" has taken the lead in the development of new engineering techniques in their field. Again and again it has been proven that "Recold" products have a dependable record of performance. Be sure to specify "Recold" on every job and you'll be sure the job is right



"Recold" Dri-Fan Evaporative Condenser From 5 to 100 ton capacities.

Write for complete "*RECOLD*" catalog!

OTHER FINE PRODUCTS WHICH ARE PRODUCED BY "RECOLD":

Tiny Mite Coils	Supreme Coils	Finned Pipe Coils
Snug Wall Coils	Humid Air Evaporators	Bare Pipe Coils
Hide-A-Way Coils	Floor Units	Commercial Ice Makers

RECOLD

REFRIGERATION ENGINEERING, INC.

7250 EAST SLAUSON AVE. LOS ANGELES 22, CALIF.

UNIT COOLERS ABOVE 32° (Continued)

(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(A,B) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(B) General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
(A,B) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 224)*
(A,B) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
(A,B) King Co., 902 N. Cedar St., Owatonna, Minn.
(A) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
(B) McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
(A,B) Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
(A,B) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
(A,B) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
(A,B) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(A,B) Rigidbilt, Inc., 2505 S. Pulaski Rd., Chicago 23, Ill. *(p. 42)*
(A,B) St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
(B) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) Trane Co., La Crosse, Wis. *(p. 14)*
(A,B) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 163)*

UNIT COOLERS ABOVE 32°—FLOOR MOUNTED
(A—Ammonia; B—Other refrigerants)

(A,B) Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
(A.B) Baker Ice Machine Co., Inc.S. Windham, Me. *(p. 191)*
(B) Betz Corp., 445 State St., Hammond, Ind.
(A,B) Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
(A,B) Bush Mfg. Co., 179 South St., W. Hartford 10, 10, Ct. *(p. 221)*
(A,B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(A,B) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(A,B) Frick Co., Waynesboro, Pa. *(p. 47)*
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(A,B) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(A,B) Howe Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 224)*
(A,B) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
(A,B) King Co., 902 N. Cedar St., Owatonna, Minn.
(A) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
(A,B) Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
(A,B) Peerless Of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(A,B) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
(A,B) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p 223)*
(A,B) Rigidbilt, Inc., 2505 S. Pulaski Rd., Chicago 23, Ill. *(p. 42)*
(A) St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A,B) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 163)*

UNIT COOLERS ABOVE 32°—WALL MOUNTED
(A—Ammonia; B—Other refrigerants)

(B) American Coils Co., 25 Lexington St., Newark 5, N.J
(A,B) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
(B) Betz Corp., 445 State St., Hammond, Ind.
(A,B) Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
(B) Bush Mfg. Co., 179 South St., W. Hartford 10, Ct. *(p. 221)*
(B) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(A,B) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.

(Continued)

CONDITIONAIRE UNIT COOLERS

Ceiling Type, for ALL Commercial Uses

• Use Ammonia, Methyl, Freon, Brine or Water with a Conditionaire. Its all-steel, welded, hot galvanized coil is suitable for any refrigerant. Other Conditionaire features: Sweat-proof, corrosion-free housing; silent motor; adjustable deflectors, any angle; thermo or float valve control. Sizes for every need. *Write for detailed literature!*

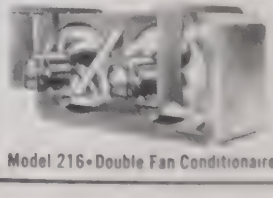
◀ Conditionaire installation • A. C. Hipp Co., Brownsville, Tex.

CONDITIONAIRE UNIT CO.

Sales Division — Howe Ice Machine Co.

2821 MONTROSE AVENUE • CHICAGO 18, ILLINOIS

Model 116 • Single Fan Conditionaire



Model 216 • Double Fan Conditionaire

CHILLATOR
Ceiling Mounted

CHILLATOR — CEILING MOUNTED — For storage rooms above 34°. Saves space. Ideal for low ceiling walk-ins. Quiet operating fans and excellent appearance.

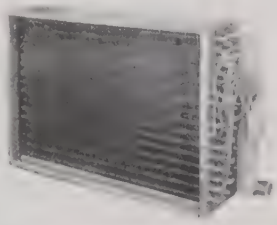
McQUAY UNIT COOLERS—For storage rooms from 0° to 50°. Rated according to latest standards. For freon and ammonia.

Unit Cooler

McQUAY ZER-O-PAK—Model X—An efficient locker plant cooling unit and sharp freezer combined in one unit. Eliminates need for a separate sharp freezing room.

ZEROPAK
Model X

McQUAY AIR CONDITIONING COILS—For central fan heating and cooling systems. Using direct expansion refrigerants or water for cooling; or steam or hot water for heating. Wide variety of sizes for any air conditioning application including industrial drying.



Air Conditioning Coil

CHILLATOR—WALL MOUNTED—For reach-in boxes and small refrigerators, beverage coolers and back bars where desired temperature is above 34°.

Chillator

ICE CUBE MAKERS—For reach-ins, back bars and special ice making cabinets. Available in both plain and quick release type.

Ice Cube Maker

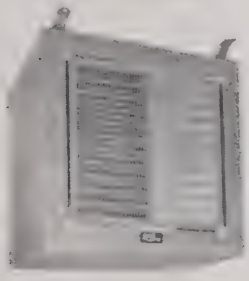
EVAPORATOR COIL—For all phases of commercial refrigeration work. A flexible line of special coils, tailor-made to fit all requirements. Inquiries from manufacturers of fixtures, cabinets and display cases invited.

Evaporator Coil

McQUAY ICY-FLO ACCUMULATOR — The new "storage-battery" for refrigeration effect is now available for handling heavy loads of short duration.

Icy-Flo Accumulator

McQUAY COMFORT COOLERS — Compact, low cost air conditioning units for the small job, i.e., beauty parlors, drug stores, etc.



Comfort Cooler

McQUAY SUSPENDED AND FLOOR TYPE AIR CONDITIONING UNITS—From 3 to 50 tons—Available with any combination of heating and cooling coils, filters, humidifiers, face and by-pass dampers, etc.

Air Conditioning Unit

Descriptive bulletins on all McQuay products available upon request

McQuay, Inc.

1600 Broadway, N.E., Minneapolis 13, Minn.

Manufacturers of Equipment for All Phases of Air Conditioning

UNIT COOLERS ABOVE 32° (Continued)

(B) Fedders-Quigan Corp., 57 Tonawanda St., Buffalo 7, N.Y.
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(B) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
(A) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
(A,B) Larkin Coils, 519 Memorial Dr., S.E., Atalnta 1, Ga. *(p. 222)*
(B) McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
(B) McCray Refrigerator Co., Kendallville, Ind.
(A,B) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(A,B) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
(A,B) D. J. Murray Mfg. Co., 1002-3rd St., Wausau, Wis.
(A,B) Niagara Blower Co., 6. E. 45th St., N.Y.C. 17 *(p. 96)*
(A,B) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(B) Reese & Long Refrigeration Products, Inc., 408 E 25th St., N.Y.C. 10
(A,B) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
(A,B) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
(A,B) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
(A,B) Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
(A,B) St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
(B) Sterling Mfg. Co., 2523 Farnam St., Omaha, Neb.
(A,B) Stewart Ice Machine Co., 1282 W. 1st St., Pomona, Cal.
(B) Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(A,B) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
(A,B) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A,B) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

UNIT COOLERS BELOW 32°—CEILING MOUNTED
(A—Ammonia; B—Other refrigerants)
(a—Reverse cycle; b—Electric; c—Brine spray; d—Water; e—Glycol; f—Warm air; g—Others)

(B:b) American Coils Co., 25 Lexington St., Newark 5, N.J.
(A,B:a,c,e) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
(B:d) Betz Corp., 445 State St., Hammond, Ind.
(f) Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
(A,B:b,d) Bush Mfg. Co., 179 South St., W. Hartford 10: Ct. *(p. 221)*
(A,B:d) Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(A,B:a,b,c,d) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(B) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(A,B:a,b,d,c) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(A,B:a,d) Howe Ice Machine Co., 2825 Montrose Ave., Chicago 18, Ill. *(p. 224)*
(A,B:d) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
(d,f) King Co., 902 N. Cedar St., Owatonna, Minn.
(a) Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
(A,B:d) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
(B:b) McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
A,B:b) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
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(A,B:a,b,d) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
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Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B:d,e,g) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(B:d) U. S. Air Conditioning Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
(A,B:a,c) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(c,e) Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
(A,B:a) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B) York Corp., York, Pa. *(p. 165)*

UNIT COOLERS BELOW 32°—FLOOR MOUNTED
(A—Ammonia; B—Other refrigerants)
(a—Reverse cycle; b—Electric; c—Brine spray; d—Water; e—Glycol; f—Warm air; g—Other)

(A,B:c) Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
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(A,B:d) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(A,B:c) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
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(A,B:a,b,c,d) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
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(A,B:d) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *p. (223)*
(c,d) Rigidbilt, Inc., 2505 S. Pulaski Rd., Chicago 23, Ill. *(p. 42)*
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(A,B:a,c) Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
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(A,B:a,c) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
(A,B:c) York Corp., York, Pa. *(p. 165)*

Only

THERMOBANK

by KRAMER

Keeps Coils Frost-Free

Automatically

at Freezer Temperatures

without . . LABOR
ATTENTION
ELECTRIC HEATERS
BRINE OR WATER SPRAYS

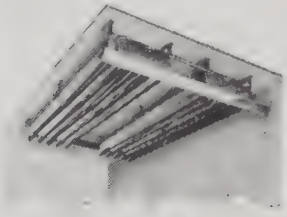
• *Write for Bulletin 16*

KRAMER TRENTON CO. *Trenton, N.J.*

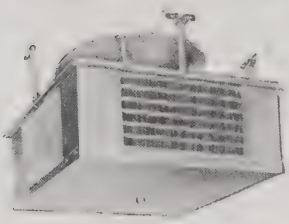
Refrigeration Economics Co., Inc.

1231 E. Tuscarawas Street, Canton 2, Ohio

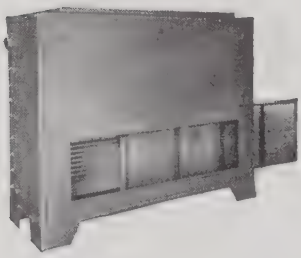
Recoy Products



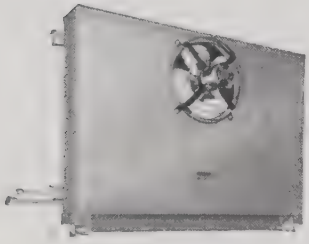
C. T. COILS. Continuous Tube down draft fin coils are still the most satisfactory for meat coolers. Other forms available for practically any application.



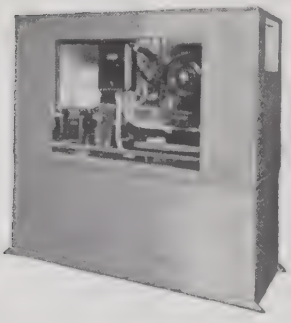
CEILING DIFFUSER. Ceiling diffusers distribute the cooled air across the ceiling, so the blast does not strike the products stored. Capacities: Refrig. 1 to 3 tons. Air Cond. 2½ to 7½ tons.



AIR CONDITIONING. Air conditioning units of ceiling or floor type in all capacities, for cooling, heating, or both. Capacities: Up to 40 tons.

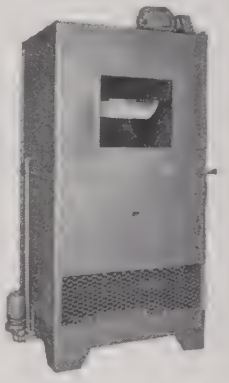


RECOY "ALL SEASONS" wall units provide means of deflecting the cold air down along the wall or out horizontally into the room, thus providing proper air circulation for "ALL SEASONS." Capacities: 1000 to 24000 B.T.U.

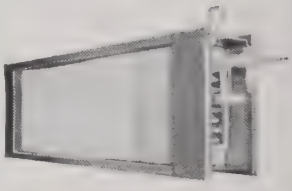


WATER COOLING. Self contained complete ice water and brine coolers complete with high and low sides, circulating pumps, controls, and insulation. Capacities: 1½ to 50 H.P.

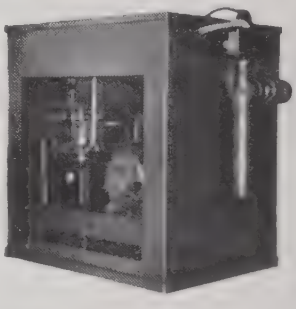
EVAPORATIVE CONDENSERS. Evaporative condensers from 2 to 100 tons. Brine spray cooling to 25 tons.



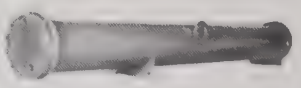
C. F. COILS. Continuous fin coils for unit coolers, blast heaters, air conditioning and condensers.



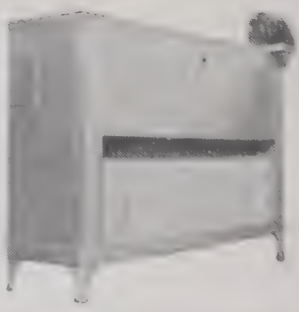
COOLANT COOLING. Coolant coolers from 1 to 20 tons are a necessity for modern production of accurate machine parts.



SHELL CONDENSER. Shell and tube, also shell and fin coil condensers. Both types have tubes arranged for cleaning with tube cleaner. Capacities: Up to 100 tons.



FLOOR UNITS. Floor units with cooling surface exposed to view have a definite advantage over those with coils hidden. Design permits water defrosting. Capacities: 3 to 12½ tons.



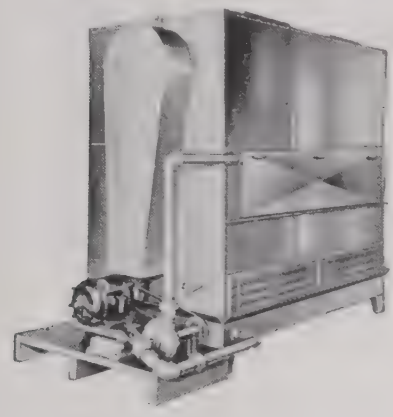
SPECIAL UNITS TO YOUR DESIGN

Marlo Coil Company

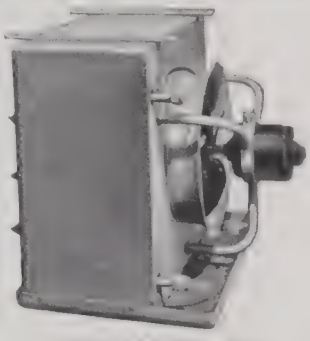
6135 Manchester Ave., St. Louis 10, Mo.

Manufacturers **MARLO = HEAT TRANSFER** Equipment
Since 1925

Industrial Coolers—Unit Coolers—Evaporative Condensers—Cooling Towers—Low Temperature Units—Air Conditioning Units—Heating and Cooling Coils—Diesel Engine and Oil Evaporative Coolers

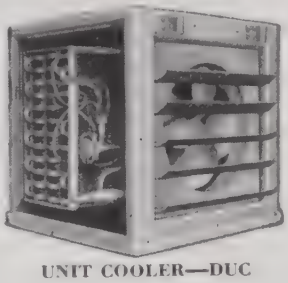


EVAPORATIVE CONDENSERS 3 to 100 tons—All refrigerants—All prime surface coils—No fins—Quiet—Motor Unidrive. Indoor or outdoor units—Durable construction. Write for *Bulletin* 404.

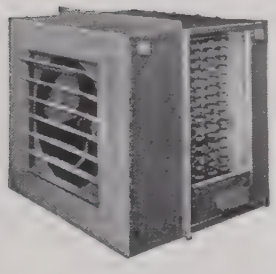


UNIT COOLER—UC

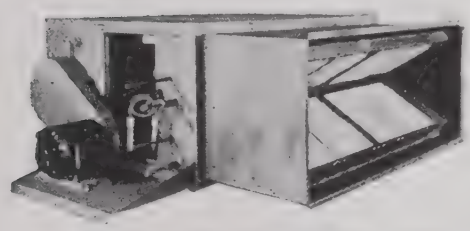
UNIT COOLERS UC and DUC Pull-through (DUC) and Blow-through (UC) types—for all refrigerants. 11 unit sizes—4 and 6 row coils—full range of capacities. 675 to 4160 cfm—Venturi fan ring—Deflector louvers. Sturdy—Economical—Quiet. Also Wall Panel Type. Write for *Bulletins* 412 and 392.



UNIT COOLER—DUC

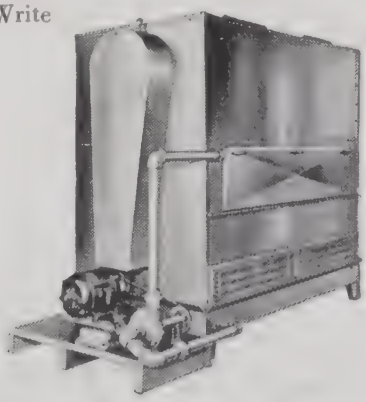


ELECTRIC DEFROST LT UNITS Compact ceiling type—High capacity—Low cost. 7 sizes—Ammonia or Freon—½ to 2½ tons at 12 deg. TD. Defrosted electrically. Quickly installed—Sectional doorway-sized. Write for *Bulletin* 408.



AIR CONDITIONING UNITS Cooling—Heating—Dehumidifying—Humidifying. 10 sizes—3 to 35 tons—900 to 13,000 cfm. Noise level ratings—Ceiling suspended or floor types. Write for *Bulletin* 409.

BLAST COILS Air conditioning—Industrial Refrigeration—Heating. Any material—All refrigerants—Every application. "BALL-BONDED"—mechanically expanded tubes to fins. Write for *Bulletin* 396.



INDUSTRIAL COOLERS 15 unit sizes—1000 to 24,000 cfm—Floor type. Galvanized frame and pans—Sectionally built. Variables: (1) Rows of coil and fin spacing. (2) Circulating brine spray or dry coil. (3) Defrost sprays optional. (4) All Refrigerants. Write for *Bulletin* 403.

Excellent service under severe conditions...

For flapper valves in compressors and elsewhere when

UNIFORM HARDNESS

HIGH FATIGUE LIFE

FINE SURFACE

ACCURATE SIZE

are called for,

SANDVIK QUALITY STEEL will meet the requirements.

SS-2

SANDVIK STEEL, INC.

111 Eighth Avenue · New York 11, N. Y. · WAtkins 9-7180

180 N. Michigan Ave., Chicago 1, Ill. 1738 Columbus Rd., Cleveland 13, Ohio

4415 S. Main St., Los Angeles 37, Cal.

UNIT COOLERS BELOW 32°—WALL MOUNTED

(A—Ammonia; B—Other refrigerants)
(a—Reverse cycle; b—Electric; c—Brine spray; d—Water; e—Glycol; f—Warm air; g—Other)

(B:b) American Coils Co., 25 Lexington St., Newark 5, N.J.
(A,B:a,c,e) Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
(B:d) Betz Corp., 445 State St., Hammond, Ind.
(A,B:a,b,c,d) Drayer-Hanson, Inc., 3301 Medford St., Los Angeles 33, Cal.
(A,B:a,b,c,d) Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
(B:d) Kennard Corp., 1819 S. Hanley Rd., St. Louis 17, Mo.
(A,B:d) Larkin Coils, 519 Memorial Dr., S.E., Atlanta 1, Ga. *(p. 222)*
(B:d) McCord Corp., 2587 E. Grand Blvd., Detroit 11, Mich. *(p. 107)*
(A,B:d) McQuay, Inc., 1600 Broadway, N.E., Minneapolis 13, Minn. *(p. 225)*
(A,B:b) Marlo Coil Co., 6135 Manchester Ave., St. Louis 10, Mo. *(p. 229)*
(A,B:e) Niagara Blower Co., 6 E. 45th St., N.Y.C. 17 *(p. 96)*
(A,B:b) Peerless of America, Inc., 2901 Lawrence Ave., Chicago 25, Ill.
(A,B:a,b,c,d) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
(A,B:a,b,d) Refrigeration Appliances, Inc., 917 W. Lake St., Chicago 7, Ill.
(A,B:b,d) Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
(A,B:d) Refrigeration Engrg., Inc., 7250 E. Slauson Ave., Los Angeles, Cal. *(p. 223)*
(A,B: a,b,c,d,e,f,g) St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
(A:d) Sterling Mfg. Co., 2523 Farnam St., Omaha, Neb.
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal
(d) Tenney Engrg., Inc., 26 Ave. B, Newark 5, N.J.
(B:d) U. S. Air Conditinoing Corp., Como Ave., S.E., at 33rd St., Minneapolis 14, Minn.
(A,B:a,b) Vilter Mfg. Co., 224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
(A,B:a,c) XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

UNIVERSAL JOINTS

Chicago Die Casting Mfg. Co., 2500 W. Monroe St., Chicago 12, Ill.
Spicer Mfg. Co., Div. of Dana Corp., 4100 Bennett Rd., Toledo 1, O.

V-BELTS (See BELTS & BELTING)

VALVE DISCS, FLAPPERS, REEDS, WAFERS, etc.

Acme Industrial Co., 205 N. Laflin St., Chicago 7, Ill.
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Chicago Seal Co., 232 S. Hoyne Ave., Chicago 20, Ill.
Cooper-Bessemer Corp., Mt. Vernon, O.
Detroit Stamping Co., 350 Midland Ave., Detroit 3, Mich.
Henry Disston & Sons, Inc., Tacony, Phila. 35, Pa.
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Leslie Co., Valley Brook & Grant Ave., Lyndhurst, N.J.
Sandvik Steel, Inc., 111-8th Ave., N.Y.C. 11 *(p. 230)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
J. H. H. Voss Co., 785 E. 144th St., N.Y.C. 54
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

VALVE PACKINGS (See PACKING)

VALVE PARTS (See also NEEDLES; also SEATS, etc.

Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Henry Disston & Sons, Inc., Tacony, Phila. 35, Pa. (Discs)
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.

VALVE POSITIONERS

Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Moore Products Co., H & Lycoming Sts., Phila. 24, Pa.
C. J. Tagliabue Mfg. Co., 550 Park Ave., Brooklyn 5, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Wheelco Instruments Co., Harrison & Peoria Sts., Chicago 7, Ill.

VALVES (See particular type under type name or following)

VALVES, AIR

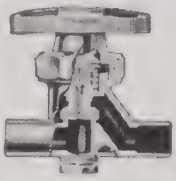
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Gas & Oil Industry Labs., Inc., 4 Paine Ave., Irvington 11, N.J.
Gorton Heating Corp., Cranford, N.J.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R. I.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Nordstrom Valve Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Walworth Co., 60 E. 42nd St., N.Y.C. 17

VALVES, ALLOY

Alloy Steel Products Co., Inc., 1508 W. Elizabeth Ave., Linden, N.J.
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Cherry-Burrell Corp., 427 W. Randolph St., Chicago 6, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.
Durabla Mfg. Co., 114 Liberty St., N.Y.C. 6
Duriron Co., Inc., Dayton 1, O.
Edward Valves, Inc., Subsidiary of Rockwell Mfg. Co., 1200 W. 145 St., E. Chicago, Ind.
Farris Engrg. Corp., 400 Commercial Ave., Palisades Park, N.J.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hasco Valve & Machinery Co., 1819 W. St. Paul Ave. Milwaukee 3, Wis.
Haveg Corp., Marshallton, Del.
Haynes Stellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
Henry Valve Co., Melrose Park, Ill. *(p. 232)*
Jenkins Bros., 80 White St., N.Y.C. 13
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Kitson Div., Welsbach Corp., 1500 Walnut St., Phila. 2, Pa.

(Continued)

HENRY



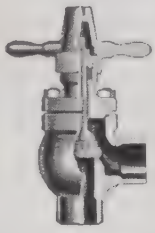
Balanced-Action Diaphragm Packless Valves

Type 626

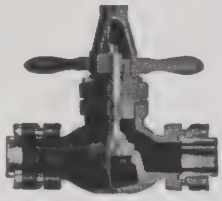
Twenty-four important features of design, construction and operation—many of them exclusive. Ports-in-line eliminate tube bending. Easier and quicker to install. Function regardless of pressure differential above or below valve seat. Leak-proof,—cannot "stick shut." Non-directional inlet and outlet—lines can be connected to either port. Solder, flare and threaded connections. Complete line shown in Catalog No. 98.

WING CAP PACKED VALVES

For Freon, Methyl Chloride and Other Refrigerants



Type 216 Non-Ferrous Alloy Angle Valve. Solder connections machined directly in valve body.



Type 223 Semi-Steel Flanged Globe Valve with Companion Flanges, Sleeves, Bolts and Gaskets.

Equipped with patented, self-aligning stem disc. Special resilient packing. Back seating thus permitting repacking under pressure. Wing cap can be inverted and its socket used to operate valve. No special wrench necessary. Added protection against leaks is provided by cap, forming tongue and groove, recessed gasketed joint with bonnet. Long stem travel between open and closed positions and full capacity passages insure unrestricted flow. Many other types shown in Catalog No. 98.

HENRY VALVE CO.

Melrose Park, Illinois, Suburb of Chicago.

APPROVED FOR USE BY ARMY • NAVY and MARITIME COMMISSION

VALVES, ALLOY (Continued)

Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
National Lead Co., 111 Broadway, N.Y.C. 6 (Lead)
Nordstrom Valve Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
U. S. Stoneware Co., 60 E. 42nd St., N.Y.C. 17
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
J. A. Zurn Mfg. Co., Erie, Pa.

VALVES, AMMONIA

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Creamery Package Mfg. Co., 1243 W. Washington Blvd., Chicago 7, Ill. *(p. 48)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 205)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Henry Valve Co., Melrose Park, Ill. *(p. 232)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.
York Corp., York, Pa. *(p. 163)*

VALVES, ANGLE, GATE, GLOBE, etc.

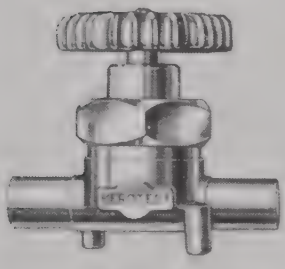
Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25, O.
Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Bailey Meter Co., 1050 Ivanhoe Rd., Cleveland 10, O.
Thomas Beckett & Co., Inc., 2118 Griffin St., Dallas 2, Tex.
L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
James B. Clow & Sons, 201 N. Talman Ave., Chicago 12, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.
Edward Valves, Inc., Subsidiary of Rockwell Mfg. Co., 1200 W. 145 St., E. Chicago, Ind.
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Flori Pipe Co., 601 E. Red Bud Ave., St. Louis, Mo.
Frick Co., Waynesboro, Pa. *(p. 47)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hancock Valve Div., Manning, Maxwell & Moore, Inc., Bridgeport 2, Ct.
Henry Valve Co., Melrose Park, Ill. *(p. 232)*
Jarecki Mfg. Co., 1345 W. 12th St., Erie, Pa.
Jenkins Bros., 80 White St., N.Y.C. 13
Jerguson Gage & Valve, 87 Fellsway, Somerville, Mass.
Kennedy Valve Mfg. Co., Elmira, N.Y.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Klingerit, Inc., 16 Hudson St., N.Y.C. 13
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
D. J. Murray Mfg. Co., 1002-3rd St., Wausau, Wis.
National Lead Co., 111 Broadway, N.Y.C. 6 (Lead & Lead-lined)

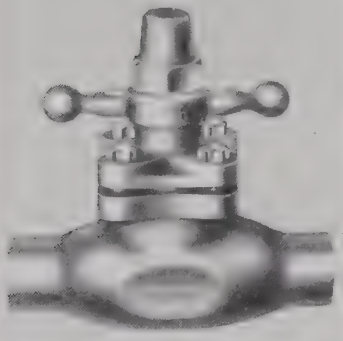
(Continued)

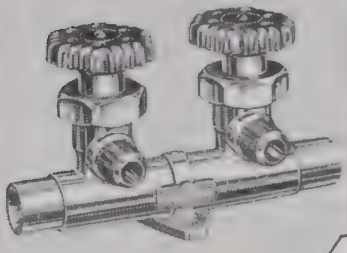
KEROTEST

America's Most Complete Line of Refrigeration and Air Conditioning

VALVES and FITTINGS



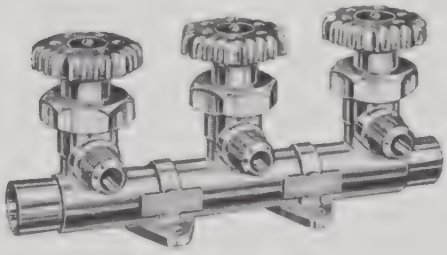


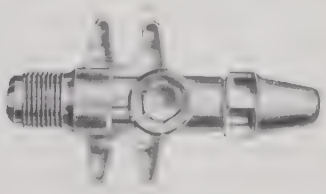


Now

SUPPLEMENTED WITH THESE NEW IMPROVED MODELS

With the addition of these newly designed in-line two-way and globe valves, compressor valves with variable bolt hole centers and forged integral valve manifolds, Kerotest Refrigeration and Air Conditioning Valves and Fittings become more than ever America's most *complete* line. Write for the new Kerotest Catalog 47-R.





KEROTEST

KEROTEST MANUFACTURING CO., PITTSBURGH 22, PA.

America's First Name in Quality Valves . . . .

VALVES, ANGLE, ETC. (Continued)

Nordstrom Valve Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12 O.
John Simmons Co., Inc., 50 Church St., N.Y.C. 7
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Trane Co., La Crosse, Wis. *(p. 14)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
White Flomatic Corp., P.O. Box 267, Hoosick Falls, N.Y.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

VALVES, CHECK, AIR, WATER (See CHECK VALVES)

VALVES, CHECK, REFRIGERANT (See CHECK VALVES, REFRIGERANT)

VALVES, COMPRESSOR SHUT-OFF

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Henry Valve Co., Melrose Park, Ill. *(p. 232)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Jenkins Bros., 80 White St., N.Y.C. 13
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Mueller Brass Co., Port Huron, Mich.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

VALVES, CARBON DIOXIDE

Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

VALVES, CYLINDER

Aluminum Industries, Inc., 2438 Beekman St., Cin'ti. 25 O.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Mueller Brass Co., Port Huron, Mich.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

VALVES, EXPANSION (See EXPANSION VALVES)

VALVES, FLOAT, REFRIGERANT (See HIGHSIDE FLOATS; also LOWSIDE FLOATS)

VALVES, FLOAT, WATER (See FLOAT VALVES, WATER)

VALVES, FOOT (See FOOT VALVES)

VALVES, MOTOR OPERATED

Automatic Switch Co., 41 E. 11th St., N.Y.C. 3
Barber-Colman Co., Rockford, Ill.
Bristol Co., Waterbury 91, Ct.
Brown Instrument Co., Div., Minneapolis-Honeywell Regulator Co., 4414 Wayne Ave., Phila. 44, Pa.
Crane Co., 836 Michigan Ave.., Chicago 5, Ill. *(p. 109)*
Cutler-Hammer, Inc., 315 N. 12 St., Milwaukee 1, Wis.
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.
Davis Regulator Co., 2511 S. Washtenaw Ave., Chicago 8, Ill.
Dickson Co., 7420 Woodlawn Ave., Chicago 19, Ill.
Edward Valves, Inc., Subsidiary of Rockwell Mfg. Co., 1200 W. 145 St., E. Chicago, Ind.
Fischer & Porter Col. Hatboro, Pa.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Jenkins Bros., 80 White St., N.Y.C. 13
Lunkenheimer Co., Beekman St., & Waverly Ave., Cint'i. 14, O.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Nordstrom Valve Div., Rockwell Mfg. Co., 400 N. Lexington Ave., Pittsburgh 8, Pa.
Philadelphia Gear Wks., Inc., G St. & Erie Ave., Phila. 20, Pa.
Sarco Co., Inc., 350-5th Ave., N.Y.C. 1 *(p. 182)*
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Walworth Co., 60 E. 42nd St., N.Y.C. 17

VALVES, NEEDLE (See also EXPANSION VALVES, HAND OR NEEDLE)

Ashton Valve Co., 161-1st St., Cambridge 42, Mass. *(p. 168)*
Thomas Beckett & Co., Inc., 2118 Griffin St., Dallas 2, Tex.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave. Chicago 39, Ill. *(p. 106)*
Edward Valves, Inc., Subsidiary of Rockwell Mfg. Co., 1200 W. 145 St., E. Chicago, Ind.
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. *(p. 232)*
Jarecki Mfg. Co., 1345 W. 12th St., Erie, Pa.
Jenkins Bros., 80 White St., N.Y.C. 13
Jerguson Gage & Valve, 87 Fellsway, Somerville, Mass.
Kerotest Mfg. Co., 2525 Liberty, Ave., Pittsburgh 22, Pa. *(p. 233)*
Lunkenheimer Co., Beekman St. & Waverly Ave., Cn'iti. 14, O.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
Wm. W. Nugent & Co., Inc., 410 N. Hermitage Ave., Chicago 22, Ill.
Oil Rite Corp., 3466 S. 13th St., Milwaukee 7, Wis.
Parker Appliance Co., 17325 Euclid Ave., Cleveland 12, O.
Swift Lubricator Co., Inc., 101 Home St., Elmira, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.
Walworth Co., 60 E. 42nd St., N.Y.C. 17
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

VALVES, PRESSURE RELIEF (See PRESSURE RELIEF VALVES)

VALVES, QUICK ACTING

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
L. J. Bordo Co., Inc., 115 New St., Glenside, Pa.
Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Dersch, Gesswein & Neuert, Inc., 4845 W. Grand Ave., Chicago 39, Ill. *(p. 106)*
Grove Regulator Co., 6529 Hollis St., Oakland, Cal.
Hubbell Corp., 319 N. Albany Ave., Chicago 12, Ill. *(p. 207)*
Jenkins Bros., 80 White St., N.Y.C. 13
Jerguson Gage & Valve, 87 Fellsway, Somerville, Mass.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
J. E. Lonergan Co., 2nd & Race Sts., Phila. 6, Pa.
Lunkenheimer Co., Beekman St. & Waverly Ave., Cin'ti. 14, O.
Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
D. J. Murray Mfg. Co., 1002-3rd St., Wausau, Wis.
Refrigerating Specialties Co., 728 S. Sacramento Blvd., Chicago 12, Ill.

VALVES, RECEIVER

Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
Mueller Brass Co., Port Huron, Mich.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.
XL Refrigerating Co., 1834 W. 59th St., Chicago 36, Ill.

VALVES, REFRIGERANT SHUT-OFF, NON-FERROUS

Automatic Switch Co., 41 E. 11th St., N.Y.C. 3
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Henry Valve Co., Melrose Park, Ill. *(p. 232)*
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsburgh 22, Pa. *(p. 233)*
Mueller Brass Co., Port Huron, Mich.
Superior Valve & Fittings Co., 1509 W. Liberty Ave., Pittsburgh 26, Pa. *(p. 108)*
Watson-Stillman Co., Roselle, N.J. *(p. 113)*
Weatherhead Co., 300 E. 131st St., Cleveland 8, O.

VALVES, REGULATING (See REGULATORS)

VALVES, SOLDER ENDS, WATER, etc.

Crane Co., 836 Michigan Ave., Chicago 5, Ill. *(p. 109)*
Darling Valve & Mfg. Co., Foot Walnut St., Williamsport, Pa.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Jenkins Bros., 80 White St., N.Y.C. 13
Northern Indiana Brass Co., 935 Plum St., Elkhart, Ind.

VALVES, SOLENOID (See SOLENOID VALVES)

VALVES, SPECIAL

Automatic Temperature Control Co., Inc., 34 E. Logan St., Phila., Pa.
Thomas Beckett & Co., Inc., 2118 Griffin St., Dallas 2, Tex.
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
Thomas A. Edison, Inc., W. Orange, N.J.
Fairbanks Co., 393 Lafayette, N.Y.C. 3
Hammel-Dahl Co., 243 Richmond St., Providence 3, R.I.
Hays Mfg. Co., 12th & Liberty Sts., Erie, Pa.
Imperial Brass Mfg. Co., 537 S. Racine Ave., Chicago 7, Ill. *(p. 111)*
Jarecki Mfg. Co., 1345 W. 12th St., Erie, Pa.
Kerotest Mfg. Co., 2525 Liberty Ave., Pittsbrugh 22, Pa. *(p. 233)*
McDonnell & Miller, Inc., 1316 Wrigley Bldg., Chicago 11, Ill.
Maid-O'-Mist, Inc., 3347 N. Pulaski Rd., Chicago 41, Ill.
Minneapolis-Honeywell Regulator Co., 2933-4th Ave., S., Minneapolis 8, Minn. *(p. 76)*
Oil Rite Corp., 3466 S. 13th St., Milwaukee 7, Wis.
Refrigerating Specialities Co., 728 S. Sacramento Blvd., Chicago 12, Ill.
Strong, Carlisle & Hammond Co., 1392 W. 3rd St., Cleveland 13, O.
Supreme Elec. Products Co., 194 Vassar St., Rochester 7, N.Y.
Taylor Instrument Cos., 95 Ames St., Rochester 1, N.Y. *(p. 183)*
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.
White Flomatic Corp., P.O. Box 267, Hoosick Falls, N.Y.

VAPOR SEALS

Alfol Div., Reflectal Corp., 155 E. 44th St., N.Y.C. 17
American Bitumuls Co., 200 Bush St., San Francisco 4, Cal.
Angier Corp., Framingham, Mass.
Benjamin Foster Co., 4635 W. Girard Ave., Phila. 31, Pa.
Munn and Steele, Inc., 130 Lister Ave., Newark 5, N.J.
Sisalkraft Co., 205 W. Wacker Dr., Chicago 6, Ill.

VARIABLE SPEED TRANSMISSIONS (See also SPEED CHANGERS; also SPEED REDUCERS)

Electric Machinery Mfg. Co., 1338 Tyler St., N.E., Minneapolis 13, Minn.
Gates Rubber Co., 999 S. Broadway, Denver 17, Colo.
Link-Belt Co., 2045 W. Hunting Park, Phila. 40, Pa.
Reliance Elec. & Engrg. Co., 1088 Ivanhoe Rd., Cleveland 10, O.
Stephens-Adamson Mfg. Co., Ridgeway Ave., Aurora, Ill.
Twin Disc Clutch Co., Racine, Wis.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*

VARNISHES, INSULATING

General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Grand Rapids Varnish Corp., 1350 Steele Ave., S.W., Grand Rapids 2, Mich.
A. C. Horn Co., Inc., 43-36-10th St., Long Island City, N.Y.
Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
Standard Varnish Wks., 2600 Richmond Terrace, Staten Island 3, N.Y.
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.

VARNISHES, PLASTIC (See also FINISHES)

Baer Bros., 438 W. 37th St., N.Y.C. 18
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Grand Rapids Varnish Corp., 1350 Steele Ave., S.W., Grand Rapids 2, Mich.
Haveg Corp., Marshallton, Del.
Interchemical Corp., 57 State St., Newark, N.J.
Maas & Waldstein Co., 438 Riverside Ave., Newark 4, N.J.
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Sherwin-Williams Co., 101 Prospect Ave., N.W., Cleveland, O.
Vita-Var Corp., 1180 Raymond Blvd., Newark 2, N.J.

VEGETABLE REFRIGERATORS (See DISPLAY CASES)

VENDING MACHINES, REFRIGERATED

Artkraft Mfg. Corp., Kibby St. & D.T.&I.R.R., Lima, O.
Bastian-Blessing Co., 4201 W. Peterson Ave., Chicago 40, Ill.
Ideal Mfg. Co., 509 S. McClun St., Bloomington, Ill.
Interstate Engrg. Corp., 2250 E. Imperial Highwy., El Segundo, Cal.
Kalva Venders, Inc., 605 W. Washington Blvd., Chicago, Ill.
Mills Industries, Inc., 4100 W. Fullerton Ave., Chicago 39, Ill. *(p. 62)*
Revco, Inc., Deerfield, Mich.
Spacarb, Inc., 311 E. 23rd St., N.Y.C. 10
Universal Refrigeration Co., 5601 W. Century Blvd., Inglewood, Cal.

VENEERS (See also PLYWOOD)

Aetna Plywood & Veneer Co., 1731 Elston Ave., Chicago 22, Ill.

VENTILATORS & VENTILATING SYSTEMS (See also particular type following)

E. H. Allen Co., 22 Dorrance St., Boston 29, Mass.
Burt Mfg. Co., 926 S. High St., Akron 11, O.
Chelsea Fan & Blower Co., Inc., 1206 Grove St., Irvington 11, N.J. *(p. 101)*
Electrovent Fan & Mfg. Co., 812 W. Lake St., Chicago 7, Ill.
Emerson Elec. Mfg. Co., 8100 Florissant Ave., St. Louis 21, Mo.
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
Hirschman-Pohle Co., Inc., 21 Lent Ave., LeRoy, N.Y.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
King Co., 902 N. Cedar St., Owatonna, Minn.
Knowles Mushroom Ventilator Co., 583 Upper Mountain Ave., Upper Montclair, N.J.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Herman Nelson Corp., 1824-3rd Ave., Moline, Ill.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
B. F. Sturtevant Div., Westinghouse Elec. Corp., Hyde Park, Boston 36, Mass. *(p. 148)*
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Trane Co., La Crosse, Wis. *(p. 14)*
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

VENTILATORS, CEILING & ROOF

Air Devices, Inc., 17 E. 42nd St., N.Y.C. 17
American Larson Ventilating Co., 1004 Keystone Bldg., Pittsburgh 22, Pa.
Arex Co., 333 N. Michigan Ave., Chicago 1, Ill.
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.
Burt Mfg. Co., 926 S. High St., Akron 11, O.
Chelsea Fan & Blower Co., Inc., 1206 Grove St., Irvington 11, N.J. *(p. 101)*
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
Hart & Cooley Mfg. Co., 500 E. 8th St., Holland, Mich.
Hartzell Propeller Fan Co., Div. of Castle Hills Corp., 910 S. Downing St., Piqua, O.
Hirschman-Pohle Co., Inc., 21 Lent Ave., LeRoy, N.Y.
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
Lockjoint Wood Products Co., 1721 Mildred Ave., Wichita 7, Kan.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Trane Co., La Crosse, Wis. *(p. 14)*
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

VENTILATORS, LOUVRE

Arex Co., 333 N. Michigan Ave., Chicago 1, Ill.
Buffalo Forge Co., P.O. Box 985, Buffalo 5, N.Y.
Burt Mfg. Co., 926 S. High St., Akron 11, O.
Chelsea Fan & Blower Co., Inc., 1206 Grove St., Irvington 11, N.J. *(p. 101)*
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
Lockjoint Wood Products Co., 1721 Mildred Ave., Wichita 7, Kan.
Maysteel Products, Inc., 135 W. Wells St., Milwaukee 3, Wis.
Milcor Steel Co., Milwaukee, Wis.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

VENTILATORS, WINDOW

Airgard Mfg. Co., 362 W. Erie St., Chicago 10, Ill.
Burt Mfg. Co., 926 S. High St., Akron 11, O.
Chelsea Fan & Blower Co., Inc., 1206 Grove St., Irvington 11, N.J. *(p. 101)*
"Frigid" Fans, Div. of Circulators & Devices Mfg. Corp., 22 Rose St., N.Y.C. 7
Ilg Elec. & Ventilating Co., 2850 N. Crawford Ave., Chicago 41, Ill.
Lockjoint Wood Products Co., 1721 Mildred Ave., Wichita 7, Kan.
Martin Fan & Blower Co., 4634 W. 21st Place, Chicago 50, Ill.
Monitor Equip. Corp., Riverdale, N.Y.C. 63
Pleasantaire Corp., 14th & K Sts., N.W., Washington 5, D.C.
H. J. Somers, Inc., 6063 Wabash Ave., Detroit 8, Mich.
O. A. Sutton Corp., KFH Bldg., Wichita, Kan.
Trane Co., La Crosse, Wis. *(p. 14)*
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
L. J. Wing Mfg. Co., 154 W. 14th St., N.Y.C. 11

VENTS, AIR

McAlear Mfg., Div. of Climax Industries, Inc., 15 N. Cincinnati, Tulsa, Okla.
Maid-O'-Mist, Inc., 3217 N. Pulaski Rd., Chicago 41, Ill.
St. Louis Blow Pipe & Heater Co., Inc., Div. of Skinner Heating & Ventilating Co., Inc., 1948 N. 9th St., St. Louis 6, Mo.
U. S. Gypsum Co., 300 W. Adams St., Chicago 6, Ill.
U. S. Register Co., 344 E. Burnham St., Battle Creek, Mich.
J. A. Zurn Mfg. Co., Erie, Pa.

VESSELS, PRESSURE (See PRESSURE VESSELS)

VIBRATION ABSORBERS, LINE

American Brass Co., Waterbury 88, Ct.
Apex Molded Products Co., 3574 Ruth St., Phila. 34, Pa.
Atlantic Metal Hose Co., Inc., 123 W. 64th St., N.Y.C. 23
Barco Mfg. Co., 1801 Winnemac Ave., Chicago 40, Ill.
Chicago Metal Hose Corp., Maywood, Ill.
Eclipse Aviation Metal Hose Dept., Div. of Bendix Aviation Corp., Phila. 44, Pa.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Packless Metal Products Corp., 31 Winthrop Ave., New Rochelle, N.Y.
Resistoflex Corp., 39 Plansoen St., Belleville 9, N.J.
Seamlex Co., Inc., 4123-24th St., Long Island City 1 N.Y. *(p. 218)*

VIBRATION ABSORBING BASES & MOUNTINGS

Apex Molded Products Co., 3574 Ruth St., Phila. 34, Pa.
Armstrong Cork Co., Lancaster, Pa. *(p. 136)*
Bishop & Babcock Mfg. Co., 4901 Hamilton Ave., N.E., Cleveland 14, O.

Bushings, Inc., 4358 Coolidge Highwy., Royal Oak, Mich.
Cork Import Corp., 39 Park Place, Englewood, N.J.
Cork Insulation Co., Inc., 155 E. 44th St., N.Y.C. 17 *(p. 142)*
Firestone Industrial Products Co., 1200 Firestone Pkwy., Akron 17, O.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
B. F. Goodrich Co., 500 S. Main St., Akron, O
Goodyear Tire & Rubber Co., 1144 E. Market St., Akron 16, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
Hamilton Kent Mfg. Co., Kent, O.
Johns-Manville, 22 E. 40th St., N.Y.C. 16 *(p. 145)*
Robt. A. Keasbey Co., 139 W. 19th St., N.Y.C. 11 *(p. 138)*
Korfund Co., Inc., 48-51-32nd Place, Long Island City 1, N.Y. *(p. 237)*
Lord Mfg. Co., 1635 W. 12th St., Erie, Pa.
MB Mfg. Co., Inc., 1060 State St., New Haven 11, Ct.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Sponge Rubber Products Co., 106 Derby Place, Shelton, Ct. *(p. 188)*
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J.
Vibration Eliminator Co., 10-28-47th Ave., Long Island City 1, N.Y.
Viking Air Conditioning Corp., 5600 Walworth Ave., Cleveland 2, O.

VOLTMETERS (See METERS, ELECTRIC)

WALK-IN COOLERS

(A—Self-contained; B—With coils but without condensing unit; C—No coils or condensing unit)

Amana Society, Amana, Ia.
(B,C) American Commercial Equip. Co., 4150 Holly Knoll, Los Angeles 27, Cal.
(A,B,C) American Refrigerator & Machine, Inc., 2700 University, N.E., Minneapolis, Minn.
Bally Case & Cooler Co., Bally, Pa.
(A) Bowser, Inc., Terryville, Ct. (Industrial)
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
(C) Cleveland Refrigerator Co., 2901 E. 65th St., Cleveland 4, O.
(C) Corbin Cabinet Lock Co., Div. of American Hardware Corp., New Britain, Ct.
(A,B) Cruse Refrigerator Co., Inc., 504 W. Main St., Louisville 2, Ky.
Duparquet-Dubois Refrigeration Div., Nathan Straus-Duparquet, Inc., 33 E. 17th St., N.Y.C. 3
(B) Econofreeze Mfg. Co., Inc., 300 Polson Bldg., Seattle 4, Wash.
Electro-Kold, Div. of E. S. Matthews, Inc., S. 151 Post St., Spokane 8, Wash.
(A,C) Esco Cabinet Co., West Chester, Pa.
(B,C) Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
(A,B,C) Fleetwood-Airflow, Inc., 421 N. Penna Ave., Wilkes-Barre, Pa.
(A,B,C) Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
(A,B) Ed Friedrich Sales Corp., 1117 E. Commerce St., San Antonio 6, Tex.
(A) Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
(A,B,C) Gem Refrigerator Co., 2539 Germantown Ave., Phila. 33, Pa.
(A,B,C) General Refrigerator & Store Fixtures Co., 856 N. Broad St., Phila., Pa.
(A,B,C) Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
(B) John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
(A,B,C) Herrick Refrigerator Co., 1019 Commercial St., Waterloo, Ia. *(p. 179)*
(A,P) C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
(B) Hussmann Refrigeration, Inc., 2401 N. Leffingwell, St. Louis 6, Mo. *(p. 87)*
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
(A,B,C) Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
(B) Koch Butchers' Supply Co., 600 E. 14th Ave., N. Kansas City 16, Mo.
(A,B,C) La Crosse Cooler Co., 2809 Losey Blvd., S., La Crosse, Wis.
(A,B,C) Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
(A,B) McCall Refrigerator Corp., Hudson, N.Y.
(B) McCray Refrigerator Co., Kendallville, Ind.
(A,B,C) Masterfreeze Corp., Sister Bay, Wis.
Matthews Refrigerator & Door Co., 5103 S.E. Powel Blvd., Portland 6, Ore.
(A,B,C) Minneapolis Show Case & Fixture Co., 1000 Washington Ave., S., Minneapolis, Minn.
(A,B,C) Modern Appliance Co., 111 S. Ellsworth, San Mateo, Cal.
(A,B,C) Modern Refrigerator Wks., 823 Milford St., Glendale 3, Cal.
(A,B,C) Nanticoke Refrigerator Manufacturers, Corner Hill & Slope Sts., Nanticoke, Pa.
(A,B,C) National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
(A,B,C) Nolin Mfg. Co., Inc., 1100 Madison Ave., Montgomery 2, Ala.
(A) Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.
R. Perlick Brass Co., 3110 W. Meinecke Ave., Milwaukee 10, Wis.
(A,B,C) J. P. Pfeiffer & Son, Inc., 200 N. Paca St., Baltimore 1, Md.
(B) Puffer-Hubbard Mfg. Co., Grand Haven, Mich.
(A,B,C) Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Refrigeration Corp. of America, Div. of Noma Elec. Corp. 55 W. 13th St., N.Y.C. 11
(B) W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
(B) Royal Store Fixture Co., 847 N. Broad St., Phila. 23, Pa.
(C) St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.

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(B) Schwenger-Klein, Inc., 720 Bolivar Rd., Cleveland, O.
(A,B) Seeger Refrigerator Co., 850 Arcade St., St. Paul 6, Minn.
(B,C) Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
(A,B,C) Charles Q. Sherman Corp., 149 Broadway, N.Y.C. 6
Super-Cold Corp., 1020 E. 59th St., Los Angeles 1, Cal.
(B) Tyler Fixture Corp., 1401 Lake St., Niles, Mich. *(p. 178)*
(A,C) Uniflow Mfg. Co., Erie, Pa.
(B) United Refrigerator Mfg. Co., Inc., 350 Robert St., St. Paul 1, Minn.
(A,B,C) Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
(B) Viking Refrigerators, Inc., 7500 Wilson Ave., Kansas City 3, Mo.
(B) Warren Co., Inc., P.O. Box 1436, Atlanta 1, Ga.
(A,B,C) Weber Showcase & Fixture Co., Inc., P.O. Box 2018, Los Angeles 54, Cal.
(B,C) Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.

WALK-IN FREEZERS (See also FARM & HOME FREEZERS)

Amana Society, Amana Ia.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Econofreze Mfg. Co., Inc., 300 Polson Bldg., Seattle 4, Wash.
Electro-Kold, Div. of E. S. Matthews, Inc., S. 151 Post St., Spokane 8, Wash.
Esco Cabinet Co., West Chester, Pa.
Federal Refrigerator Mfg. Co., 550 Elizabeth St., Waukesha, Wis.
Fogel Refrigerator Co., 5400 Eadom St., Phila. 37, Pa.
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Frosty Foods Equip. Co., 305 Benson Blvd., Sioux City 15, Ia.
Haverly Elec. Co., Inc., 1970 W. Fayette St., Syracuse, N.Y.
Henshaw Refrigeration & Fixture Co., 25 Oak Grove St., San Francisco 7, Cal.
John Herrel & Sons Co., 244 Lear St., Columbus 6, O.
C. V. Hill & Co., Inc., 360 Pennington Ave., Trenton 1, N.J.
Jewett Refrigerator Co., Inc., 2 Letchworth St., Buffalo 13, N.Y.
Jordon Refrigerator Co., 235 N. Broad St., Phila. 7, Pa.
Jack Langston Co., 3700 Elm St., Dallas 1, Tex.
Masterfreeze Corp., Sister Bay, Wis.
Modern Appliance Co., 111 S. Ellsworth, San Mateo, Cal.
National Refrigerators Co., 827 Koeln Ave., St. Louis 11, Mo.
Orley Freezers, Inc., 680 E. Fort St., Detroit 26, Mich.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Refrigeration Corp. of America, Div. of Noma Elec. Corp., 55 W. 13th St., N.Y.C. 11
W. Allen Rogers Industries, Inc., P.O. Box 272, Demopolis, Ala.
Royal Store Fixture Co., 847 N. Broad St., Phila. 23, Pa.
St. Louis Butchers' Supply Co., 1545 N. 15th St., St. Louis 6, Mo.
Sherer-Gillett Co., S. Kalamazoo Ave., Marshall, Mich.
Victor Products Corp., 901 Pope Ave., Hagerstown, Md.
Western Mineral Products Co., 1720 Madison St., N.E., Minneapolis 13, Minn.
York Corp., York, Pa. *(p. 163)*

WASHERS (See also particular type; also GROMMETS; also PACKING; also GASKETS)

Aetna Ball & Roller Bearing Co., 4600 Schubert Ave., Chicago 39, Ill.
Airtex Corp., 333 N. Michigan Ave., Chicago 1, Ill.
American Nut & Bolt Fastener Co., 2029 Doerr St., Pittsburgh 12, Pa.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Clark Bros. Bolt Co., Milldale, Ct.
Detroit Stamping Co., 350 Midland Ave., Detroit 3, Mich.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Wm. H. Haskell, Mfg. Co., 24 Commerce St., Pawtucket, R.I.
E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
Lakeside Malleable Castings Co., Racine, Wis.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Phoenix Specialty Mfg. Co., Inc., 80 Albany Ave., Freeport, Long Island, N.Y.
Rhopac, Inc., 168 N. Clinton St., Chicago 6, Ill.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Whitehead Stamping Co., 1661 W. Lafayette Blvd., Detroit 16, Mich.
Wrought Washer Mfg. Co., 2253 S. Bay St., Milwaukee 7, Wis

WASHERS, AIR (See AIR WASHERS)

WASHERS, LEATHER (See also LEATHER SPECIALTIES)

Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
Baldwin Belting, Inc., 85 Chambers St., N.Y.C. 7
H. N. Cook Belting Co., 401 Howard St., San Francisco 5, Cal.
Excelsior Leather Washer Mfg. Co., Inc., Rockford, Ill.
Fisher Leather Belting Co., Inc., 325 N. 3rd St., Phila. 6, Pa.
E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Phoenix Specialty Mfg. Co., Inc., 80 Albany Ave., Freeport, Long Island, N.Y.
Horace G. Preston Co., 2581 Beecher Ave., Detroit 16, Mich.
Rhopac, Inc., 168 N. Clinton St., Chicago 6, Ill.

WASHERS, LOCK

Aetna Ball & Roller Bearing Co., 4600 Schubert Ave., Chicago 39, Ill.
American Nut & Bolt Fastener Co., 2029 Doerr St., Pittsburgh 12, Pa.
Autoscrew Co., 216 W. 18th St., N.Y.C. 11
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
H. M. Harper Co., 2620 W. Fletcher St., Chicago 18, Ill.
Lamson & Sessions Co., 1971 W. 85th St., Cleveland 2, O.
Philadelphia Steel & Wire Corp., 5250 Belfield Ave., Phila. 44, Pa.
SKF Industries, Inc., Front St. & Erie Ave., Phila. 32, Pa.
St. Louis Screw & Bolt Co., 6900 N. Broadway, St. Louis 15, Mo.
Shakeproof, Inc., 2501 N. Keeler Ave., Chicago 39, Ill.
Stronghold Screw Products, Inc., 216 W. Hubbard St., Chicago 10, Ill.
Thompson-Bremer & Co., 1640 W. Hubbard St., Chicago 22, Ill.
Wrought Washer Mfg. Co., 2253 S. Bay St., Milwaukee 7, Wis.

WASHERS, RUBBER

American Nut & Bolt Fastener Co., 2029 Doerr St., Pittsburgh 12, Pa.
Anchor Packing Co., 401 N. Broad St., Phila. 8, Pa.
Apex Molded Products Co., 3574 Ruth St., Phila. 34, Pa.
Atlas Asbestos Co., Ltd., 110 McGill St., Montreal 1, Quebec, Canada
Ball Bros. Co., Muncie, Ind.
Ernst Water Column & Gage Co., 250 S. Livingston Ave., Livingston, N.J.
General Tire & Rubber Co., Garfield St., Wabash, Ind. *(p. 187)*
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Goshen Rubber & Mfg. Co., Box 517, Goshen, Ind. *(p. 186)*
E. F. Houghton & Co., 303 W. Lehigh Ave., Phila. 33, Pa.
Pacific States Felt & Mfg. Co., Inc., 843 Howard St., San Francisco 3, Cal.
Phoenix Specialty Mfg. Co., Inc., 80 Albany Ave., Freeport, Long Island, N.Y.
Quaker Rubber Corp., Tacony & Milnor Sts., Phila. 24, Pa.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
Rhopac, Inc., 168 N. Clinton St., Chicago 6, Ill.
Stokes Molded Products, Inc., Taylor at Webster St., Trenton 4, N.J. (Hard Rubber)
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

DOYLE & ROTH MFG. CO.

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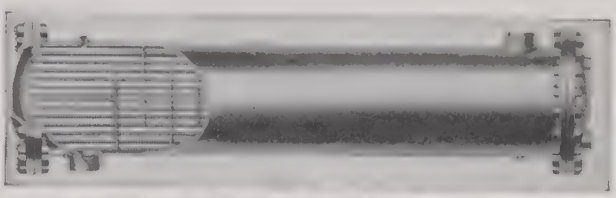
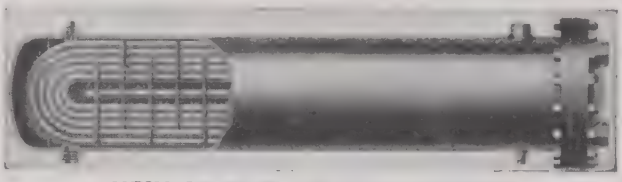
Newark 5, New Jersey

EXCHANGERS

- CONDENSERS
- PROCESS COOLERS
- OIL HEATERS
- WATER COOLERS
- FEED WATER HEATERS



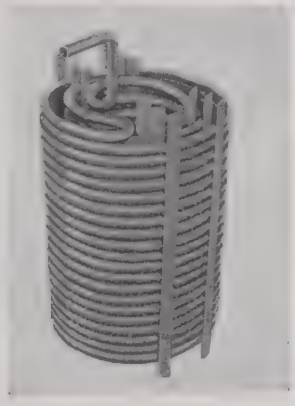
STRAIGHT TUBE EXCHANGERS



"U" TUBE EXCHANGERS

PIPE COILS

- BRINE TANK COILS
- REFRIGERANT COILS



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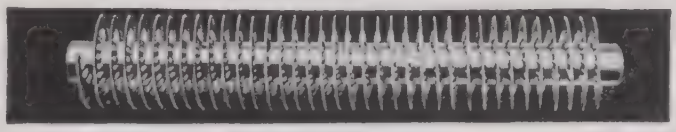
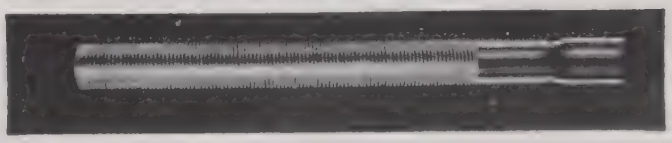
HIGH FINNED TUBING



LOW FINNED TUBING



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WASHERS, SPRING

American Nut & Bolt Fastener Co., 2029 Doerr St., Pittsburgh 12, Pa.
Philadelphia Steel & Wire Corp., 5250 Belfield Ave., Phila. 44, Pa.
Phoenix Specialty Mfg. Co., Inc., 80 Albany Ave., Freeport, Long Island, N.Y.
Shakeproof, Inc., 2501 N. Keeler Ave., Chicago 39, Ill.
Stronghold Screw Products Co., 216 W. Hubbard St., Chicago 10, Ill.
Wrought Washer Mfg. Co., 2253 S. Bay St., Milwaukee 7, Wis.

WASHERS, THRUST

Aetna Ball & Roller Bearing Co., 4600 Schubert Ave., Chicago 39, Ill.
Cleveland Graphite Bronze Co., 17000 St. Clair Ave., Cleveland 10, O. *(p. 27)*
Phoenix Specialty Mfg. Co., Inc., 80 Albany Ave., Freeport, Long Island, N.Y.
Shenango-Penn Mold Co., Dover, O.
Whitehead Stamping Co., 1661 W. Lafayette Blvd., Detroit 16, Mich.
Wrought Washer Mfg. Co., 2253 S. Bay St., Milwaukee 7, Wis.

WATER COILS (See AIR CONDITIONING COILS)

WATER COOLERS (See also particular type; also DRINKING WATER COOLERS)

Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Audiffren Refrigerating Co., Proctor, Vt.
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Crandal-Stone Div., Brewer-Titchener Corp., 336 Court St., Binghamton, N.Y. *(p. 167)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Ebco Mfg. Co., 401 W. Town St., Columbus 8, O. *(p. 240)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
General Elec. Co., Air Conditioning Dept., 5 Lawrence St., Bloomfield, N.J. *(p. 65)*
Heat-X-Changer Co., Inc., 415 Lexington Ave., N.Y.C. 17 *(p. 240)*
King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 240)*
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
W. Allen Rogers Industries, Inc, P.O. Box 272, Demopolis, Ala.
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

WATER COOLERS, ATMOSPHERIC (See COOLING TOWERS)

WATER COOLERS, BAUDELOT TYPE (See also AERATORS)

Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
Burge Ice Machine Co., 218 N. Jefferson St., Chicago 6, Ill.
Frick Co., Waynesboro, Pa. *(p. 47)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Jensen Machinery Co., Inc., Nelson St. at Locust Ave., Bloomfield, N.J. *(p. 241)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
York Corp., York, Pa. *(p. 163)*

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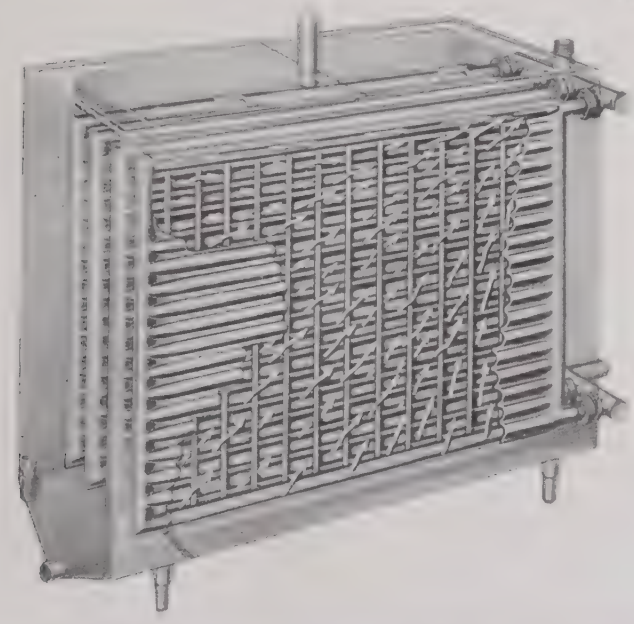

Cutaway view of a Jensen Industrial Chilled Water Unit showing placement of cooling sections in tank. Arrows indicate short refrigerant gas flow.

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WATER COOLERS, DRY EXPANSION

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Coolstream Corp., 240 Butler St., Brooklyn 17, N.Y.
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*
Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Interstate Engrg. Corp., 2250 E. Imperial Highway, El Segundo, Cal.
King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 240)*
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 19, Va.
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.

WATER COOLERS, SHELL & COIL

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Downingtown Iron Wks., Downingtown, Pa. *(p. 59)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Interstate Engrg. Corp., 2250 E. Imperial Highwy., El Segundo, Cal.
King-Zeero Co., 1447 Montrose Ave., Chicago 13, Ill. *(p. 240)*
Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Potter & Rayfield, Inc., P.O. Box 1042, Atlanta 1, Ga.
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Rempe Co., 340 N. Sacramento Blvd., Chicago 12, Ill.
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. *(p. 147)*
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

WATER COOLERS, SHELL & TUBE

Acme Industries, Inc., Mechanic & Ganson Sts., Jackson, Mich. *(p. 60)*
Richard M. Armstrong Co., Box 188, W. Chester, Pa. *(p. 125)*
Baker Ice Machine Co., Inc., S. Windham, Me. *(p. 191)*
California Steel Products Co., Barrett & "A" Sts., Richmond, Cal.
Carrier Corp., 302 S. Geddes St., Syracuse 1, N.Y. *(p. 51)*
Downingtown Iron Wks., Downingtown, Pa. *(p. 59)*
Doyle & Roth Mfg. Co., Foot Hawkins St. & C.R.R.N.J., Newark 5, N.J. *(p. 239)*
Frick Co., Waynesboro, Pa. *(p. 47)*
Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Kramer Trenton Co., Olden & Breuning Aves., Trenton 5, N.J. *(p. 227)*
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Patterson-Kelley Co., Inc., E. Stroudsburg, Pa. *(p. 58)*
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Refrigeration Economics Co., Inc., 1231 E. Tuscarawas St., Canton 4, O. *(p. 228)*
Richmond Engrg. Co., Inc., 7th & Hospital Sts., Richmond 19, Va.
Ross Heater & Mfg. Co., Div. of American Radiator & Standard Sanitary Corp., Buffalo 13, N.Y.
Schnacke, Inc., 1016 E. Columbia St., Evansville 7, Ind.
Southwestern Engrg. Co., 4800 Santa Fe Ave., Los Angeles 11, Cal.
Standard Heater & Oil Equip. Co., 245 Cornelison Ave., Jersey City 2, N.J.
Trane Co., La Crosse, Wis. *(p. 14)*
Typhoon Air Conditioning Co., Inc., Div. of Ice Air Conditioning Co., Inc., 794 Union St., Brooklyn 51, N.Y.
Vilter Mfg. Co., 2224 S. 1st St., Milwaukee 7, Wis. *(p. 49)*
Whitlock Mfg. Co., Drawer 390, Hartford 1, Ct.
Wittenmeier Machinery Co., 850 N. Spaulding Ave., Chicago 51, Ill.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

WATER GAUGE COLUMNS (See GAUGE GLASSES)

WATER HEATERS

Frigidaire Div., Gen'l. Motors Corp., Dayton 1, O. *(p. 6)*
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Leonard Div., Nash-Kelvinator Corp., 14250 Plymouth Rd., Detroit 32, Mich.
Patterson-Kelley Co., E. Stroudsburg, Pa. *(p. 58)*
Red Spot. Elec. Co., Tacoma, Wash.
Rheem Mfg. Co., 570 Lexington Ave., N.Y.C. 22
Servel, Inc., Evansville 20, Ind. *(p. 69)*
David Stout & Sons, 7016 Manchester Blvd., St. Louis 17, Mo.
Wilson Refrigeration, Inc., Div. of Wilson Cabinet Co., Inc., Smyrna, Del.

WATER PURIFIERS (See WATER TREATING)

WATER STRAINERS (See STRAINERS)

WATER TREATING & TESTING EQUIPMENT & MATERIALS

American Sand-Banum Co., Inc., 9 Rockefeller Plaza, N.Y.C. 20 *(p. 36)*
American Water Softener Co., S.E. Corner 4th & Lehigh Ave., Phila. 33, Pa.
Automatic Pump & Softener Corp., 2412 Grant Ave., Rockford, Ill.
E. W. Bacharach & Co., 800 Rialto Bldg., Kansas City 6, Mo.
Bello Industrial Equip. Div., Bogue Elec. Co., 37 Kentucky Ave., Paterson, N.J.
W. H. & L. D. Betz, Gillingham & Worth Sts., Phila. 24, Pa.
Buhring Water Purifying Co., P.O. Box 155, Atlantic Highlands, N.J.
Chemical Solvent Co., Box 487, Birmingham, Ala.
Cochrane Corp., 17th St. below Allegheny Ave., Phila. 32, Pa.
Dearborn Chemical Co., 310 S. Michigan Ave., Chicago, Ill.
Dow Chemical Co., Midland, Mich.
Elgin Softener Corp., Elgin, Ill.
Feedwaters, Inc., 140 Cedar St., N.Y.C. 6
Ferro-Nil Corp., 381-4th Ave., N.Y.C. 16
Filter Paper Co., 2450 S. Michigan Ave., Chicago 16, Ill.
Filtrine Mfg. Co., 53 Lexington Ave., Brooklyn 5, N.Y. *(p. 243)*
General Ozone Corp., 1455 W. Congress St., Chicago 7, Ill.
Graver Tank & Mfg. Co., Inc., 4809 Tod Ave., E. Chicago 1, Ind.
D. W. Haering & Co., 205 W. Wacker Dr., Chicago 6, Ill.
Inertol Co., Inc., 470 Frelinghuysen Ave., Newark 5, N. J.
Infilco, Inc., 325 W. 25th St., Chicago 16, Ill.
Lehigh Fan & Blower Co., Div. of Heilman Boiler Wks., Inc., 128 Linden St., Allentown, Pa.
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Mueller Steam Specialty Co., Inc., 40-20-22nd St., Long Island City 1, N.Y.
Mutual Chemical Co. of America, 270 Madison Ave., N.Y.C. 16
National Aluminate Co., 6216 W. 66th Place, Chicago 38, Ill. *(p. 244)*
North American Fibre Products Co., Standard Bldg., Cleveland 13, O.
Oakite Products, Inc., 22 Thames St., N.Y.C. 6
Oshkosh Filter & Softener Co., 51 Ceape St., Oshkosh, Wis.
Perolin Co., Inc., 10 E. 40th St., N.Y.C. 16
Pittsburgh Plate Glass Co., 632 Duquesne Way, Pittsburgh 22, Pa.
Puro Filter Corp. of America, 440 Lafayette St., N.Y.C. 3
Reco Products Div., Refrigeration Engrg. Corp., 2020 Naudain St., Phila. 46, Pa. *(p. 130)*
Milton Roy Co., 1401 E. Mermaid Ave., Phila. 18, Pa.
Tempo Chemical Co., Inc., 47-02-5th St., Long Island City 1, N.Y.
Water Service Labs., Inc., 423 W. 126th St., N.Y.C. 27
Waters Filter & Cooler Co., 10 W. 29th St., N.Y.C. 1
Western Chemical Co., 713 Washington St., Kansas City 6, Mo.
Wilbur-Williams Co., 43 Leon St., Boston, Mass.
Worthington Pump & Machinery Corp., Harrison, N.J. *(p. 66)*
York Corp., York, Pa. *(p. 163)*

WATTMETERS (See METERS, ELECTRIC)

WEATHER STRIP (See also DOOR GASKETS)

Brasco Mfg. Co., Harvey, Ill.
Chamberlin Co. of America, 1254 La Brosse St., Detroit 26, Mich.
W. J. Dennis & Co., 1732 N. Kolmar Ave., Chicago 39, Ill.
B. F. Goodrich Co., 500 S. Main St., Akron, O.
Raybestos-Manhattan, Inc., 61 Willett St., Passaic, N.J.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

WELDING, CONTRACT

E. H. Allen Co., 22 Dorrance St., Boston 29, Mass.
Behringer Metal Wks., Inc., 108 Jabez St., Newark 5, N.J.
Eisler Engrg. Co., Inc., 740 S. 13th St., Newark 3, N.J. (Spot)
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Frostrode Products, Inc., 19929 Exeter Rd., Detroit 3, Mich.
Heintz Mfg. Co., Front St. & Olney Ave., Phila. 20, Pa.
Swan Engrg. Co., Inc., 22 Nelson St., Bloomfield, N.J.
Vibrashock Div., Robinson Aviation, Inc., Teterboro, N.J. (Aluminum Spot)
Henry Vogt Machine Co., 10th & Ormsby St., Louisville 10, Ky.

WELDING ELECTRODES, RODS & WIRE

American Brass Co., Waterbury 88, Ct.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Arcos Corp., 1515 Locust St., Phila. 2, Pa. (Stainless)
Bristol Brass Corp., Bristol, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Driver-Harris Co., Harrison, N.J.
Eutectic Welding Alloys Corp., 40 Worth St., N.Y.C. 13
Frostrode Products, Inc., 19929 Exeter Rd., Detroit 3, Mich.
Galv-Weld Products, 324 E. 2nd St., Dayton 2, O.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Handy & Harman, 82 Fulton St., N.Y.C. 7
International Nickel Co., 67 Wall St., N.Y.C. 5 (Monel, Nickel, etc.)
Linde Air Products Co., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17 *(p. 197)*
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind. (Electrodes only)
Marquette Mfg. Co., Inc., 307 E. Hennepin, Minneapolis 13, Minn.
National Carbon Co., Inc., Unit of Union Carbide & Carbon Corp., 30 E. 42nd St., N.Y.C. 17
Revere Copper & Brass, Inc., 230 Park Ave., N.Y.C. 17 *(p. 217)*
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.
Westinghouse Elec. Corp., 4454 Genesee St., Buffalo 5, N.Y.

WELDING ELECTRODE COOLERS (See ELECTRODE COOLING SYSTEMS)

WHEELS, BLOWER (See BLOWER WHEELS)

WINDOWS, COLD STORAGE (See COLD STORAGE DOORS & WINDOWS)

WIRE (See also particular type)

American Rolling Mill Co., Middletown, O.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Bethlehem Steel Co., Bethlehem, Pa.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Youngstown Sheet & Tube Co., Youngstown, O.

WIRE, ALUMINUM

Aluminum Co. of America, Pittsburgh 19, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Metallizing Engrg. Co., Inc., 38-14-30th St., Long Island City 2, N.Y.
Reynolds Metals Co., 2500 S. 3rd St., Louisville 1, Ky.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

WIRE, BRASS, BRONZE, etc.

American Brass Co., Waterbury 88, Ct.
Bristol Brass Corp., Bristol, Ct.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
P. R. Mallory & Co., Inc., 3029 E. Washington St., Indpls., Ind.
Metallizing Engrg. Co., Inc., 38-14-30th St., Long Island City 2, N.Y.
Scovill Mfg. Co., 99 Mill St., Waterbury 91, Ct.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

WIRE, CABLE & ROPE

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
Macwhyte Wire Rope Co., 2923-14th Ave., Kenosha, Wis.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Il.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

WIRE, ELECTRIC (See also CABLE, ELECTRIC; CORDS ELECTRIC)

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
General Elec. Co., 1 River Rd., Schenectady 5, N.Y.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
U. S. Rubber Co., 1230 Ave. of the Americas, N.Y.C. 20

WIRE, ELECTRIC RESISTANCE

Driver-Harris Co., Harrison, N.J.

WIRE, GALVANIZED

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

WIRE, INCONEL, MONEL & NICKEL

Audubon Wire Cloth Corp., Richmond St. & Castor Ave., Phila. 34, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Driver-Harris Co., Harrison, N.J.
Haynes Steellite Co., Unit of Union Carbide & Carbon Corp., Kokomo, Ind.
International Nickel Co., 67 Wall St., N.Y.C. 5
Metallizing Engrg. Co., Inc., 38-14-30th St., Long Island City 2, N.Y.
United Wire & Supply Corp., 1497 Elmwood Ave., Providence 7, R.I.

WIRE, STAINLESS STEEL

Allegheny Ludlum Steel Corp., Oliver Bldg., Pittsburgh 22, Pa.
American Rolling Mill Co., Middletown, O.
American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Arcos Corp., 1515 Locust St., Phila. 2, Pa.
Audubon Wire Cloth Corp., Richmond St. & Castor Ave., Phila. 34, Pa.
Carpenter Steel Co., Reading, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Driver-Harris Co., Harrison, N.J.
Metallizing Engrg. Co., Inc., 38-14-30th St., Long Island City 2, N.Y.
Republic Steel Corp., Republic Bldg., Cleveland 1, O.
Tennessee Coal, Iron & Railroad Co., U. S. Steel Corp. Subsidiary, Brown-Marx Bldg., Birmingham, Ala.

WIRE, TINNED

American Steel & Wire Co., Rockefeller Bldg., Cleveland 13, O.
Bethlehem Steel Co., Bethlehem, Pa.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Laclede Steel Co., Arcade Bldg., St. Louis 1, Mo.

WIRE, WELDING **(See WELDING ELECTRODES)**

WIRE CLOTH & MESH **(See also MESH, METAL)**

Audubon Wire Cloth Corp., Richmond St. & Castor Ave., Phila. 34, Pa.
Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
Central Steel & Wire Co., 3000 W. 51st St., Chicago 32, Ill.
Chase Brass & Copper Co., 236 Grand St., Waterbury 91, Ct.
Cyclone Fence Div., American Steel & Wire Co., U. S. Steel Corp. Subsidiary, P.O. Box 260, Waukegan 1, Ill.
Kentucky Metal Products Co., Preston St. & Audubon Park, Louisville 4, Ky.
Michigan Wire Cloth Co., 2098 Howard St., Detroit 16, Mich.
Newark Wire Cloth, 351 Verona Ave., Newark 4, N.J.
Joseph T. Ryerson & Son, Inc., 16th & Rockwell Sts., Chicago, Ill.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18

WIRE FORMING, WIRE SPECIALTIES & WIREWORK

American Spring & Wire Specialty Co., 816 N. Spaulding Ave., Chicago 51, Ill.
Audubon Wire Cloth Corp., Richmond St. & Castor Ave., Phila. 34, Pa.
Bethlehem Steel Co., Bethlehem, Pa.
Buffalo Wire Wks., 450 Terrace, Buffalo 2, N.Y.
L. F. Grammes & Sons, Inc., 365 Union St., Allentown, Pa.
Hindley Mfg. Co., 53 John St., Valley Falls, R.I.
Hunter Pressed Steel Co., 801 Maple St., Lansdale, Pa.
Kentucky Metal Products Co., Preston St. & Audubon Park, Louisville 4, Ky.
McIntire Connector Co., 252 Jefferson St., Newark 5, N.J. *(p. 189)*
Union Steel Products Co., 448 Pine St., Albion, Mich.
United Steel & Wire Co., Battle Creek, Mich.
Wall Wire Products Co., 11333 General Dr., Plymouth, Mich.
Wickwire Spencer Steel Div., Colorado Fuel and Iron Corp., 500-5th Ave., N.Y.C. 18
L. A. Young Spring & Wire Corp., 9200 Russell St., Detroit 11, Mich.

WIRING DEVICES **(See also HARNESSES, ELECTRICAL)**

General Elec. Co., 1285 Boston Ave., Bridgeport 2, Ct.
Graybar Elec. Co., Inc., Graybar Bldg., N.Y.C. 17
Interstate Mfg. Corp., 133 Sussex Ave., Newark 4, N.J. *(p. 246)*
Riverside Mfg. Co., 200 S. River St., Ypsilanti, Mich. *(p. 24)*

WORT COOLERS, BREWERIES

Jensen Machinery Co., Inc., Nelson St. at Locust Ave., Bloomfield, N.J. *(p. 241)*

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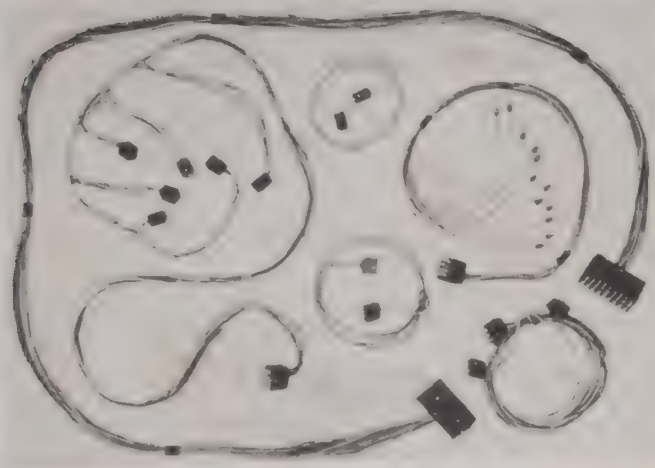

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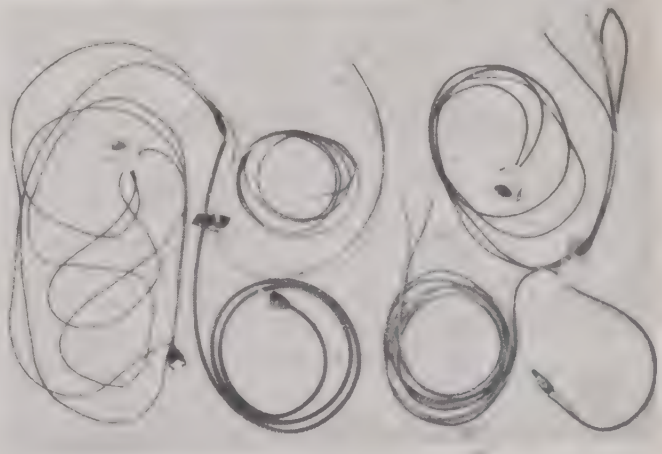
RECORD PLAYERS



X-RAY COOLING EQUIPMENT (See also WATER COOLERS)

Gay Engrg. Co., 2730 E. 11th St., Los Angeles 23, Cal.
Temprite Products Corp., 47 Piquette Ave., Detroit 2, Mich. (*p. 147*)

ZEOLITES

Liquid Conditioning Corp., 114 E. Price St., Linden, N.J.

ZINC

Belmont Smelting & Refining Wks., Inc., 330 Belmont Ave., Brooklyn 7, N.Y.
Eagle-Picher Sales Co., American Bldg., Cin'ti. 1, O.
Galv-Weld Products, 324 E. 2nd St., Dayton 2, O.
Illinois Zinc Co., 2959 W. 47th St., Chicago 32, Ill.
Metallizing Engrg. Co., Inc., 38-14-30th St., Long Island City 2, N.Y.
National Lead Co., 111 Broadway, N.Y.C. 6
New Jersey Zinc Co., 160 Front St., N.Y.C. 7

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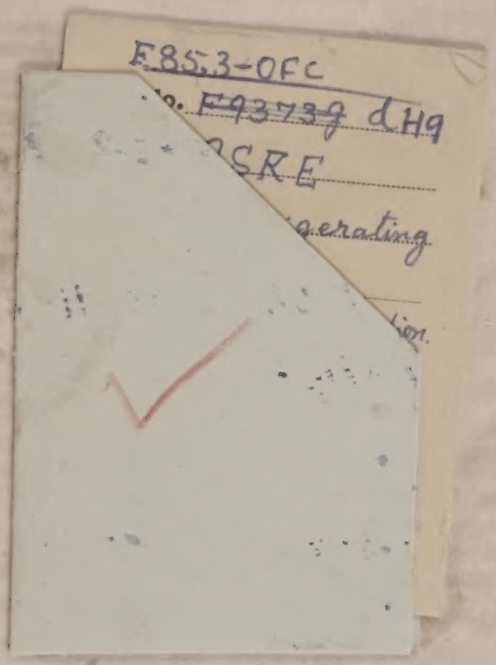
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